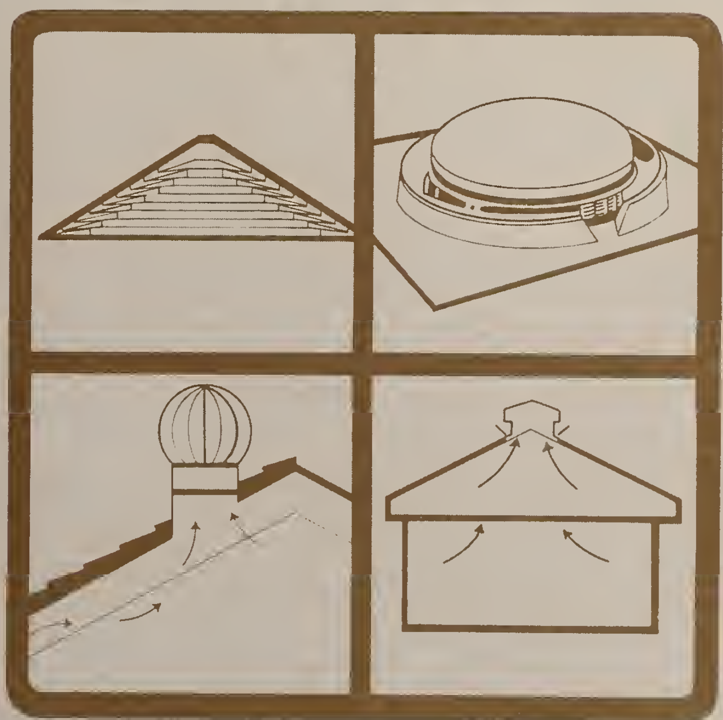
DEPARTMENT OF COMMERCE ★ NATIONAL BUREAU OF STANDARDS

NBS SPECIAL PUBLICATION 548

U.S. DEPARTMENT OF COMMERCE / National Bureau of Standards

Summer Attic and Whole-House Ventilation



NATIONAL BUREAU OF STANDARDS

The National Bureau of Standards[1] was established by an act of Congress March 3, 1901. The Bureau's overall goal is to strengthen and advance the Nation's science and technology and facilitate their effective application for public benefit. To this end, the Bureau conducts research and provides: (1) a basis for the Nation's physical measurement system, (2) scientific and technological services for industry and government, (3) a technical basis for equity in trade, and (4) technical services to promote public safety. The Bureau's technical work is performed by the National Measurement Laboratory, the National Engineering Laboratory, and the Institute for Computer Sciences and Technology.

THE NATIONAL MEASUREMENT LABORATORY provides the national system of physical and chemical and materials measurement; coordinates the system with measurement systems of other nations and furnishes essential services leading to accurate and uniform physical and chemical measurement throughout the Nation's scientific community, industry, and commerce; conducts materials research leading to improved methods of measurement, standards, and data on the properties of materials needed by industry, commerce, educational institutions, and Government; provides advisory and research services to other Government Agencies; develops, produces, and distributes Standard Reference Materials; and provides calibration services. The Laboratory consists of the following centers:

Absolute Physical Quantities[2] — Radiation Research — Thermodynamics and Molecular Science — Analytical Chemistry — Materials Science.

THE NATIONAL ENGINEERING LABORATORY provides technology and technical services to users in the public and private sectors to address national needs and to solve national problems in the public interest; conducts research in engineering and applied science in support of objectives in these efforts; builds and maintains competence in the necessary disciplines required to carry out this research and technical service; develops engineering data and measurement capabilities; provides engineering measurement traceability services; develops test methods and proposes engineering standards and code changes; develops and proposes new engineering practices; and develops and improves mechanisms to transfer results of its research to the utlimate user. The Laboratory consists of the following centers:

Applied Mathematics — Electronics and Electrical Engineering[2] — Mechanical Engineering and Process Technology[2] — Building Technology — Fire Research — Consumer Product Technology — Field Methods.

THE INSTITUTE FOR COMPUTER SCIENCES AND TECHNOLOGY conducts research and provides scientific and technical services to aid Federal Agencies in the selection, acquisition, application, and use of computer technology to improve effectiveness and economy in Government operations in accordance with Public Law 89-306 (40 U.S.C. 759), relevant Executive Orders, and other directives; carries out this mission by managing the Federal Information Processing Standards Program, developing Federal ADP standards guidelines, and managing Federal participation in ADP voluntary standardization activities; provides scientific and technological advisory services and assistance to Federal Agencies; and provides the technical foundation for computer-related policies of the Federal Government. The Institute consists of the following divisions:

Systems and Software — Computer Systems Engineering — Information Technology.

[1]Headquarters and Laboratories at Gaithersburg, Maryland, unless otherwise noted; mailing address Washington,D.C. 20234.
[2]Some divisions within the center are located at Boulder, Colorado, 80303.

The National Bureau of Standards was reorganized, effective April 9, 1978.

SUMMER ATTIC AND WHOLE-HOUSE VENTILATION

Proceedings of a Workshop Held on July 13, 1978
At the National Bureau of Standards
Gaithersburg, Maryland

Edited by:

Mary H. Reppert

Center for Building Technology
National Engineering Laboratory
National Bureau of Standards
Washington, D. C. 20234

Co-Sponsored by the

Department of Energy
Washington, D. C. 20585

and the Attic Ventilation Industry

U.S. DEPARTMENT OF COMMERCE, Juanita M. Kreps, Secretary

Jordan J. Baruch, Assistant Secretary for Science and Technology

NATIONAL BUREAU OF STANDARDS, Ernest Ambler, Director

Issued July 1979

Library of Congress Catalog Card Number: 79-600086

National Bureau of Standards Special Publication 548
Nat. Bur. Stand. (U.S.) Spec. Publ. 548, 153 pages (July 1979)
CODEN: XNBSAV

U.S. GOVERNMENT PRINTING OFFICE
WASHINGTON: 1979

For sale by the Superintendent of Documents, U.S. Government Printing Office, Washington, D.C. 20402
Stock No. 003-003-02089-3 Price $4.75
(Add 25 percent additional for other than U.S. mailing).

Li

SUMMER ATTIC AND WHOLE-HOUSE VENTILATION

ABSTRACT

These are the proceedings of the Summer Attic and Whole-house Ventilation Workshop sponsored by the National Bureau of Standards in collaboration with the Department of Energy and the attic ventilation industry. The purpose of the Workshop was to provide a forum for technical discussion to assess summer energy savings that might be achieved from the use of static and powered attic ventilation and whole-house ventilation equipment. Papers on experimental and mathematical model studies relating to attic and whole-house ventilation were presented. In addition, a paper on roof solar absorptance and its effect on the cooling requirement of a residence was presented.

After each paper was presented, participants of the Workshop were given an opportunity to question the speaker concerning his paper. The resulting questions and responses are included at the end of each of the papers.

Key Words: Attic ventilation; energy conservation; solar absorptance of roofing materials; whole-house ventilation.

TABLE OF CONTENTS

PREFACE

With the advent of increasing energy costs and concern about diminishing natural resources, much interest has been directed toward energy conservation strategies which can be used to reduce summer air conditioning costs. One item of consideration toward this goal has been the effect of attic ventilation on air conditioning and indoor comfort.

In order to provide a forum for technical discussion to assess the summer energy savings that might result from the use of powered and non-powered attic ventilating and whole-house ventilating equipment, the National Bureau of Standards sponsored the Summer Attic and Whole-House Ventilation Workshop, which was held at the NBS Gaithersburg site on July 13, 1978. Papers on experimental and mathematical modeling studies relating to attic and whole-house ventilation were presented, followed by brief discussion and invitation for questions.

Dr. Preston E. McNall, Chief of the Building Thermal and Service Systems Division, NBS, was Chairman of the Workshop. Douglas M. Burch was the Coordinator, and Mary H. Reppert is Editor of the Proceedings. The Proceedings reflect, in chronological sequence, the main presentations by the speakers. Every effort has been made to minimize the editing and to reflect each author's original material as submitted prior to the Workshop.

SOLAR REFLECTIVITY OF COMMON ROOFING MATERIALS AND ITS INFLUENCE ON THE ROOF HEAT GAIN OF TYPICAL SOUTHWESTERN RESIDENCES

J. A. Reagan
The University of Arizona, Tucson 85721

D. M. Acklam
U.S. Air Force, SAMSO, USAYPG, Yuma, Arizona 85364

The fraction of incident solar energy reflected by the exterior elements of a structure significantly affects the overall heat gain or loss of the structure. This is particularly true for regions that receive an abundance of solar insolation, as occurs in the southwestern part of the United States. However, the available data on the solar reflectivity of common building materials are apparently rather sparse. In order to augment these data, the authors developed a probe for measuring hemispheric solar flux and employed it to measure the solar reflectance of a wide variety of building materials. Results of these measurements will be presented and reviewed. The measured reflectivity values have also been used in heat gain/loss computations for typical southwestern residences. Results of some of these computations will be presented to demonstrate the change in structure heat gain/loss that can realistically be achieved by altering roof reflectivity.

Key words: Solar reflectivity; solar absorptivity; sol-air temperature; TETD; heat-transfer calculations, building heat gain/loss computations.

1. Introduction

The amount of incident sun light reflected and absorbed by the outside surfaces of a structure can significantly affect the overall heat gain or loss of the structure. It is well known that a simple degree day analysis approach is inadequate for computing summer heat gains of structures located in regions that receive an abundance of solar insolation. For the purpose of making heat-transfer calculations, the effect of incident sunlight being partially absorbed by a building element is typically included through the use of the sol-air temperature which requires specifying the solar absorptivity of the element. The available data on the solar absorptivity and reflectivity of common building materials are apparently rather sparse. More importantly, only a very limited selection of solar absorptivity values (such as Light or Dark values) is typically available in standard design information for making heat-transfer calculations in buildings. This paper presents solar reflectivity measurements made for a variety of building materials used in residential construction in the southwestern part of the United States. Calculations are also presented which display the influence of solar absorptivity on heat transfer through opaque building elements, particularly with regard to summer heat gains through roofs of southwestern residences.

2. Solar Reflectivity

2.1 Theoretical Development

The radiant flux incident on a surface element is in general partially absorbed, partially reflected, and partially transmitted. If the surface element is opaque, as is the case for most non-glass wall and roof elements of a structure, the portion of incident radiant flux which is not absorbed is reflected. Thus, if the reflectance of an opaque surface element is known, one can determine the fraction of incident radiant flux which will be absorbed by the surface, or vice versa. The reflectance of interest in this case is the total or hemispheric reflectance, R_t, which is defined in general as the ratio of total radiant flux density (flux per unit area) which is reflected to the total radiant flux density incident on the surface element [1]. Analytically, R_t may be expressed by

$$R_t = \frac{\int_{\lambda_{r1}}^{\lambda_{r2}} \int_0^{2\pi} \int_0^{\pi/2} \ell_r(\theta,\phi,\lambda_r)\cos\theta \, d\Omega(\theta,\phi) \, d\lambda_r}{\int_{\lambda_{i1}}^{\lambda_{i2}} \int_0^{2\pi} \int_0^{\pi/2} \ell_i(\theta,\phi,\lambda_i)\cos\theta \, d\Omega(\theta,\phi) \, d\lambda_i} \quad (1)$$

where

θ,ϕ = zenith and azimuth variables, respectively, of a spherical coordinate system which defines a hemisphere over the surface element.

$d\Omega(\theta,\phi)$ = $\sin\theta d\theta d\phi$, the differential solid angle along the direction θ,ϕ.

$\ell_i(\theta,\phi,\lambda_i)$ = monochromatic radiant intensity (radiant energy/time-area-solid angle-wavelength) at wavelength λ_i incident on surface element along direction θ,ϕ.

$\ell_r(\theta,\phi,\lambda_r)$ = monochromatic radiant intensity at wavelength λ_r reflected from surface element along direction θ,ϕ.

$\lambda_{i1} \rightarrow \lambda_{i2}$ = wavelength range of incident radiant intensity.

$\lambda_{r1} \rightarrow \lambda_{r2}$ = wavelength range of reflected radiant intensity.

If R_t is applied to describe the solar reflectivity of a surface, which is the case of interest here, then $\lambda_i \cong \lambda_r = \lambda$ and the wavelength range $\lambda_1 \rightarrow \lambda_2$ extends from about 0.35 to 2.0 microns for solar radiation reaching the earth's surface [2,3]. For clear sky conditions, $\ell_i(\theta,\phi,\lambda)$ is composed primarily of the directly transmitted solar component if the surface is exposed to direct sunlight, but $\ell_r(\theta,\phi,\lambda)$ is effectively diffused in all directions for reflection from optically rough surfaces characteristic of common building materials. The wavelength dependence of $\ell_r(\theta,\phi,\lambda)$ may also be significantly altered from that of $\ell_i(\theta,\phi,\lambda)$ depending on the color of the surface.

2.2 Measurement Technique and Instrumentation

The denominator of equation 1 is simply an expression for the hemispheric flux density incident on the surface in question, while the numerator term describes the hemispheric flux density reflected from the surface. Thus, to experimentally determine R_t, one only has to measure the hemispheric flux incident on and reflected from the surface and divide the reflected flux measurement by the incident flux measurement. This is easier said than done because to accurately measure hemispheric flux, as can be seen from the denominator or numerator term of equation 1, the measurement probe must have 1) a uniform spectral response over the wavelength range of interest, and 2) a Lambertian or $\cos\theta$ weighted response to intensity incident on the probe from any direction θ,ϕ.

The probe that was developed for the purpose of making the reflectivity measurements reported here consists basically of a color corrected photodetector mounted behind a *WHITE LIGHT* diffusive receiver. Specifically, a United Detector PIN-10 DF silicon photodiode was used for the photodetector. The photodiode was operated in the photovoltaic mode such that the output voltage was proportional to the flux incident on the photodiode. The spectral response of the PIN-10 DF is essentially flat and extends over the wavelength range from about 0.44 to 0.96 microns. While this wavelength range does not extend over the complete spectral range of sunlight received at the earth's surface (~0.35 to 2.0 microns), it does cover a range which includes about 65 → 70% of the incident solar energy. Moreover, most of the energy which is not detected falls within the longwave 0.96 to 2.0 micron wavelength region, and the reflectivities of materials and coatings commonly used in buildings typically do not deviate greatly in this spectral region from the visible wavelength region [1]. The diffusive receiving element of the probe was made from a plastic diffuser material, Plexiglass W-2247. This material has excellent diffusing properties [4], but simply using a flat piece of it over the photodetector will not provide a Lambertian response. The material was geometrically corrected according to the design determined by Huttenhow [5] to form a diffusing head which did follow the Lambertian response extremely well for incidence angles out to about 85°.

Extensive tests were performed with the probe to determine how to best use it for measuring the solar reflectivity of building materials. To accurately measure the reflectivity of a wall or roof element, it was determined that an unobstructed target area about 6 feet in diameter was required. To make reflectivity measurement, the probe was first positioned 6 inches (0.152m) away from and aimed along a line normally incident on the surface in question. This defined the alignment for measuring the flux reflected from the surface. The probe was then rotated 180° to point directly away from the surface which defined the alignment for measuring the flux incident on the surface. The ratio of the reflected flux measurement divided by the incident flux measurement was then taken as the reflectivity of the surface. Measurements of this type were made on both vertical wall elements and flat or nearly flat roof elements for many different sun angles. It was observed that the solar reflectivity of a roof or wall element typically remained constant over a wide range of sun angles, which indicated that the probe's angular response was indeed nearly Lambertian. Probe measurements of the direct normal, total horizontal, and diffuse horizontal (by occulting the direct solar rays) solar flux also served to confirm the Lambertian response of the probe. Additional details about the probe assembly and these test measurements are given by Acklam [6].

2.3 Measurement Results

Results of solar reflectivity measurements made on a variety of wall and roof materials used in structures in Tucson, Arizona are summarized in Tables 1 through 5. The corresponding solar absorptivity value for any of these materials is simply one

TABLE 1. SOLAR REFLECTIVITIES OF WALLS MADE OF CONCRETE AND ADOBE BLOCKS.

Description	Reflectance value
Burnt adobe block, running bond, tooled light grey mortar joint	36%
Same with raked joint	34%
Colored slump block, running bond, concave low contrast mortar joint	
Tan (San Xavier[a] SX-15)	43%
Plain (San Xavier SX-16)	44%
Buff (Columbia Block[b])	39%
Santa Rosa (Columbia Block)	36%
Palo Verde (San Xavier SX-17)	33%
Coral(San Xavier SX-14)	38%
Adòbe Red (Columbia Block) with raked joint	21%
Colored CMU (concrete masonry unit) running bond, concave low contrast mortar joint	
Coral (San Xavier SX-14)	34%
Adobe Red (San Xavier SX-26)	32%
Buff (Columbia Block)	31%
Plain or grey	39%
Same with plain joint	45%

[a]San Xavier Rock & Materials/Tucson color index.

[b]Columbia Building Materials/Tucson color index.

TABLE 2. SOLAR REFLECTIVITIES OF WALLS MADE OF BRICKS

Description	Reflectance value
Brown (PBY[a] color #19) scratch brick, common bond, concave medium grey mortar joint	28%
Same color ruffled brick, basket weave bond, same color and type joint	36%
Same with herringbone bond	33%
Light red (PBY color #16) scratch brick, common bond, concave medium grey mortar joint	38%
Orange (PBY color #06) ruffled brick, plain medium grey mortar joint	41%
Buff (PBY color #94) plain brick, stack bond stretchers, raked medium grey mortar joint	51%
Same color ruffled brick, English cross bond, concave medium grey mortar joint	43%
Same color scratch brick, running bond, plain medium grey mortar joint	41%
Red (PBY color #04) ruffled brick, third bond oversize brick, raked medium grey mortar joint	35%
Same color and type brick, English cross bond, concave medium grey mortar joint	34%

[a]Phoenix Brick Yard/Tucson Division color index.

TABLE 3. SOLAR REFLECTIVITIES OF PAINTED AND COATED WALLS.

Description	Reflectance value
Painted slump block, running bond, concave joint	
Pearl White (Pioneer Paints[a])	74%
Navaho White (Pioneer Paints)	70%
White (Pioneer Paints)	71%
Spanish White (Pioneer Paints)	68%
Egg Shell White (Pioneer Paints)	65%
Mortar washed, solid grey coverage on slump block, same bond and joint	49%
Painted CMU (concrete masonry unit), same bond and joint	
Bone White (Southwestern Paints[b])	73%
Navaho White (Pioneer Paints)	72%
Sea Shell Beige (Pioneer Paints)	55%
Pearl White (Pioneer Paints)	69%
Desert Sand (Sears Roebuck & Co.)	42%
Painted stucco, Bone White (Southwestern Paints)	65%
Painted wood paneling	
Avocado Green (Pioneer Paints)	15%
Sand Dune (Pioneer Paints)	26%
Beige (brand unknown)	40%
Stained wood paneling	
Weathered Brown (2310 Southwestern's wood stain)	10%
Dark Brown (2302 Southwestern's wood stain)	13%

[a]Pioneer Paint & Varnish Co./Tucson color index.

[b]Southwestern Paint & Varnish Co./Tucson color index.

TABLE 4. SOLAR REFLECTIVITIES OF SHINGLED ROOFS.

Description	Reflectance value
Asphalt tab shingles, common lay	
Woodblend (GAF)	17%
Russet Blend (GAF)	9%
Autumn (Flintkote)	10%
Frosted Red (Flintkote)	20%
Canyon Red (Flintkote)	13%
Snow White (Flintkote)	24%
Dark Mahogany (GAF)	8%
Pastel Green (GAF)	16%
Earthtone Brown (GAF)	9%
Blizzard (Fire King)	34%
White (JM)	33%
Red (JM)	14%
Clover Green (Flintkote)	11%
Shake cedar wood shingles, new, unoiled	32%
Same but oiled	28%
Red clay mission tile	26%

TABLE 5. SOLAR REFLECTIVITIES OF COATED AND BUILT-UP ROOFS.

Description	Reflectance value
Pea gravel covered	
Dark blend	12%
Medium blend	24%
Light blend	34%
White coated	65%
Crushed used brick, red, covered	34%
White marble chips covered	49%
Flexstone or mineral chip roof type, white	26%
Polyurethane foam, white coated	70%
Same with tan coating	41%
Silver, aluminum painted tar paper	51%
White coated, smooth, Kool Kote (Corbett Roofing Co./Tucson)	75%
Tarpaper, "weathered"	41%

minus the reflectivity value. The reflectivities of a few miscellaneous ground surfaces are also given in Table 6. While many of the products cited in the tables are made in Tucson, they are, nevertheless, similar in type and color to products available in many other cities in the southwestern United States. Where comparison is possible, the reflectivities given here appear to be in good agreement with values given elsewhere for similar materials and coatings [1,7].

From the results given in Tables 1 through 5, it appears, for the purpose of heat gain/loss calculations, that the different materials and coatings may be conveniently classified in the different color-reflectivity groups given in Table 7. The Light, Medium, and Dark classifications are the same as proposed earlier by Reagan [8], and most southwestern residences appear to be fairly well described by these three choices. The Very Light category is essentially limited to the case of a stark white paint or coating applied to a fairly smooth surface, while the Very Dark category is limited to a few dark paints, dark shingles, or a black roof coating.

TABLE 6. SOLAR REFLECTIVITIES OF MISCELLANEOUS GROUND SURFACES.

Description	Reflectance value
Grass, mowed	25%
Desert soil, natural	29%
Weathered asphalt driveway	19%
Redwood decorative chips, weathered	19%
Concrete slab, smooth, light grey	36%
Crushed used brick, red, decorative landscape	30%

TABLE 7. SUGGESTED COLOR-REFLECTIVITY CLASSIFICATION FOR OPAQUE BUILDING MATERIALS.

Color Code	Solar Reflectivity R_t	Solar Absorptivity $(1 - R_t)$
Very Light	0.75	0.25
Light	0.65	0.35
Medium	0.45	0.55
Dark	0.25	0.75
Very Dark	0.10	0.90

Very Light: Smooth building material surfaces covered with a fresh or clean stark white paint or coating.

Light: Masonry, textured, rough wood, or gravel(roof) surfaces covered with a white paint or coating.

Medium: Off-white, cream, buff or other light colored brick, concrete block, or painted surfaces and white-chip marble covered roofs.

Dark: Brown, red or other dark colored brick, concrete block, painted, or natural wood walls and roofs with gravel, red tile, stone, or tan to brown shingles.

Very Dark: Dark brown, dark green or other very dark colored painted, coated, or shingled surfaces.

3. Influence of Solar Reflectivity on Heat Transfer through Opaque Building Elements

3.1 Sol-Air and TETD Temperatures

The heat transferred per unit area per unit time, q, through an opaque building element may be expressed by [9]

$$q = U \cdot TD \tag{2}$$

where

- q = thermal flux density transmitted through building element, Btu/hr-ft^2 (W/m^2).
- U = thermal transmission coefficient, air to air, of building element, Btu/hr-ft^2-°F (W/m^2-°C).
- TD = effective air temperature difference between outside and inside faces of building element, °F (°C).

To accurately determine q, the effects of outside temperature variations and solar absorption, longwave absorption or emission, and thermal storage by the building element must be properly accounted for. These effects can be handled by using the Total Equivalent Temperature Differential [9], TETD, in place of TD in equation 2. While the TETD approach is not as accurate as other methods such as the Response Factor method [10] for making transient heat transfer calculations on an hourly basis, it does yield daily average values that are in good agreement with averages determined by the more involved transient techniques [11,12]. Thus, average TETD values may be used with some success in energy conservation and analysis studies requiring computation of daily, monthly, or seasonal heat gains and losses.

The daily average TETD for fairly repetitive external temperature and radiation conditions is given by

$$TETD_a = T_{ea} - T_i \tag{3}$$

where T_i is inside air temperature (assumed to be constant) and T_{ea} is the average sol-air temperature determined by averaging the instantaneous sol-air temperature [9], $T_e(t)$, over 24 hours,

$$T_{ea} = \frac{1}{24}\int_0^{24} T_e(t)\,dt \approx \frac{1}{24}\sum_{k=1}^{24} T_e(t_k). \tag{4}$$

The instantaneous sol-air temperature is derived from the heat balance equation for a surface element where heat is transferred into and away from the surface by conduction, convection, and radiation [9]. For the exterior surface of an opaque building element exposed to sunlight, $T_e(t)$ is given by

$$T_e(t) = T_o(t) + \frac{\alpha\ I(t)}{h_o(t)} - \frac{\varepsilon\ \Delta R(t)}{h_o(t)} \tag{5}$$

where

$T_e(t)$ = sol-air temperature in °F (°C) at time t (hr).

$T_o(t)$ = outside air temperature at time t in °F (°C).

α = solar hemispheric absorptivity of surface element (equal to $1 - R_t$).

$h_o(t)$ = surface film heat transfer coefficient from surface element to outside air at time t, Btu/hr-ft^2-°F (W/m^2-°C).

$I(t)$ = hemispheric solar flux density incident on surface element at time t, Btu/hr-ft^2(W/m^2).

ε = longwave hemispheric emittance of surface element.

$\Delta R(t)$ = net longwave radiant flux density loss from surface element to atmosphere at time t if surface were a black body, Btu/hr-ft^2(/m^2).

The incident solar flux density, I(t), in equation 5 varies with latitude, time of year, orientation of the surface element with respect to the sun, time of day, and the particular turbidity and cloud cover conditions on the day in question. Tables of I(t), representative of clear days, are given for various latitudes, months of the year, and hours of the day in the 1972 ASHRAE Handbook of Fundamentals [9], and values of I(t) for conditions other than those covered in the tables may be obtained using the computational methods given in the NBSLD computer program [12]. The surface film heat-transfer coefficient, $h_o(t)$, varies with wind velocity, surface roughness, and surface temperature. Methods for calculating $h_o(t)$ are also given in NBSLD. The longwave radiation factor, $\Delta R(t)$, depends upon the effective sky temperature [13], the temperature of the surface element, and the surface to sky view factor [13].

From the above, it is evident that $T_e(t)$ cannot readily be determined without having certain supplemental information and/or making certain simplifying assumptions. The approach outlined in the 1972 ASHRAE Handbook of Fundamentals [9] is to assume $\varepsilon \approx 1$, $h_o \approx 3$ Btu/hr-ft^2-°F (~ 17 W/m^2-°C), and ΔR equal to either 20 Btu/hr-ft^2 (~ 63 W/m^2) for horizontal surfaces or 0 Btu/hr-ft^2 (0 W/m^2) for vertical surfaces. In addition, the solar absorptivity, α, is set either to 0.9 for *DARK* surfaces or 0.45 for *LIGHT* surfaces. These approximations are really more gross than they have to be. For example, the results given in the previous section indicate that the solar absorptivity can be specified with greater accuracy using a classification such as that given in Table 7. Calculations of $h_o(t)$ using the method given in NBSLD [12] also indicate that the daily average value of $h_o(t)$ is better approximated by a value between 4.0 and 5.0 Btu/hr-ft^2-°F (~ 22.7 and 28.4 W/m^2-K) for typical Tucson weather conditions. Finally, rather than simply setting $\Delta R(t)$ equal to some assumed constant value, it can be solved for along with $T_e(t)$ using constrained iterative techniques. These modifications have been incorporated in a modified TETD determination method developed by one of the authors [14], and tables of modified TETD values have been computed for use in residential energy conservation analyses [8,15]. A comparison of daily average sol-air temperatures, T_{ea}, computed for Tucson weather conditions using the method given in the 1972 ASHRAE Handbook of Fundamentals and the abovementioned modified method are given in Table 8. The results are for north, south, east, and west wall and horizontal roof orientations, and apply for the months of July and January. The same values of incident solar flux density were used in each method (I(t) for ~ 32° north latitude and 21st of July and January as determined by subroutines Sun and Solad in NBSLD [12]). The required temperature and wind speed data were ten-year averages for July and January determined from National Weather Service measurements made at Tucson International Airport. Values of T_{ea} computed by the ASHRAE method are given for solar absorptivities of $\alpha = 0.9$, which is the value used for Dark in the ASHRAE method, and for $\alpha = 0.75$, which is the value recommended for Dark given earlier in Table 7. Values of T_{ea} computed by the modified method are given for only $\alpha = 0.75$. The results obtained by the ASHRAE method for $\alpha = 0.9$ and 0.75 show that the change in absorptivity does cause a noticeable change in T_{ea} for sunlit orientations. However, comparison of the results obtained by the two methods for $\alpha = 0.75$ shows that T_{ea} is altered by even a greater amount by improving on the values of $h_o(t)$ and $\Delta R(t)$ used in determining T_{ea}. The T_{ea} values obtained by the modified method are consistently lower than those obtained by the ASHRAE method. This means that average TETD values obtained by the modified method (recall that $TETD_a = T_{ea} - T_i$ from equation 3)

TABLE 8. COMPARISON OF DAILY AVERAGE TUCSON SOL-AIR TEMPERATURES COMPUTED BY DIFFERENT METHODS.

Month & Orientation*		Daily Average Sol-Air Temperature, T_{ea} °F (°C)		
		ASHRAE Method		Modified (Reagan) Method
		α = 0.9	α = 0.75	α = 0.75
July	N	91.8 (33.2)	90.7 (32.6)	85.9 (29.9)
	E	101.7 (38.7)	98.9 (37.2)	92.6 (33.7)
	S	92.3 (33.5)	91.2 (32.9)	88.9 (31.6)
	W	101.7 (38.7)	98.9 (37.2)	90.8 (32.7)
	H	109.9 (43.3)	104.7 (40.4)	94.3 (34.6)
January	N	51.6 (10.9)	51.2 (10.7)	47.9 (8.8)
	E	58.0 (14.4)	56.5 (13.7)	51.8 (11)
	S	73.4 (23)	69.4 (20.8)	59.8 (15.4)
	W	58.0 (14.4)	56.6 (13.7)	51.4 (10.8)
	H	57.4 (14.1)	54.9 (12.7)	50.0 (10)

*N,E,S,W,&H for north, east, south, west, and horizontal orientations.

T_{oa} = 85.3 °F (29.6°C) in July

T_{oa} = 49.4 °F (9.67°C) in January

will predict lower summer heat gain and higher winter heat loss values than predicted by the ASHRAE method.

The heat gain/loss effect of significantly altering the absorptivity is shown in Table 9 where daily average TETD values for the horizontal (flat roof) orientation are given for α = 0.75 (Dark), 0.55 (Medium), and 0.35 (Light) as determined by both the ASHRAE and the modified methods. The values apply for Tucson for the months of July and January, and the calculations were made for the same I(t) and weather conditions used to compute the results given in Table 8. The inside temperature, T_i, was set at 78°F (25.6°C) for July and 72°F (22.2°C) for January. It can be seen that varying α does not greatly affect $TETD_{aH}$ for January, because the sun angle is low for this month, but $TETD_{aH}$ is greatly affected by changing α in July. The results obtained by the two methods also differ the most for July. However, the results for either method indicate that making a roof light in color will significantly reduce the summer roof heat gain but not greatly increase winter roof heat loss (for flat or very low-pitched roofs).

3.2 Example Heat Gain/Loss Computations for a Typical Tucson Residence

Heat gain/loss calculations have been made by one of the authors [15] for several types of Tucson residences using daily average TETD values, $TETD_a$, computed by the modified method mentioned in the preceding subsection. The solar absorptivity values used for these calculations were based on the color classifications given in Table 7. The floor plan for one of these residences is shown in Figure 1, and the specifications for the structure are given in Table 10. The house is oriented with the broader side and most window area facing north-south, as is common practice in the southwest to reduce solar heat gain effects. The specifications for Case A are representative of poorly insulated homes, many of which were constructed in Tucson prior to about 1974, while the specifications for Case B are representive of better insulated homes now being built in Tucson. The predicted daily heat gain for an average July day in Tucson for Cases A and B, for both light and dark colored roofs, are given in Table 11. The results show that while changing the roof color from dark (α = 0.75) to light (α = 0.25) does greatly reduce the roof heat gain, the reduction in total house heat gain is nevertheless small (~6% or less) because the roof heat gain is only a small fraction (<~13%) of the total house heat gain. Calculations for a number of other houses in Tucson have revealed similar results. The roof component of summer heat gain (for dark roofs) was found to be less than 20% of the total house heat gain in all cases that were analyzed. By comparing the results for cases A and B, it is apparent that adding insulation, particularly wall insulation, is a much more effective means of reducing summer heat gain if the house is rather poorly insulated to begin with. Insulation also has the added benefit of reducing winter

TABLE 9. COMPARISON OF DAILY AVERAGE TUCSON HORIZONTAL (ROOF) TETD VALUES COMPUTED FOR DIFFERENT ABSORPTIVITIES AND BY DIFFERENT METHODS.

Month & Absorptivity α		Daily Average Horizontal TETD, $TETD_{aH}$ °F (°C)	
		ASHRAE Method	Modified (Reagan) Method
	0.35	12.8 (7.1)	8.1 (4.5)
July	0.55	19.8 (11)	12.2 (6.8)
	0.75	26.7 (14.8)	16.3 (9.1)
	0.35	-23.6 (-13.1)	-25.8 (-14.3)
January	0.55	-20.3 (-11.3)	-23.9 (-13.3)
	0.75	-17.3 (-9.6)	-22.0 (-12.2)

T_i = 72°F (22.2°C) for July

T_i = 78°F (25.6°C) for January

Computations for same I(t) and weather data conditions used to obtain results in Table 8.

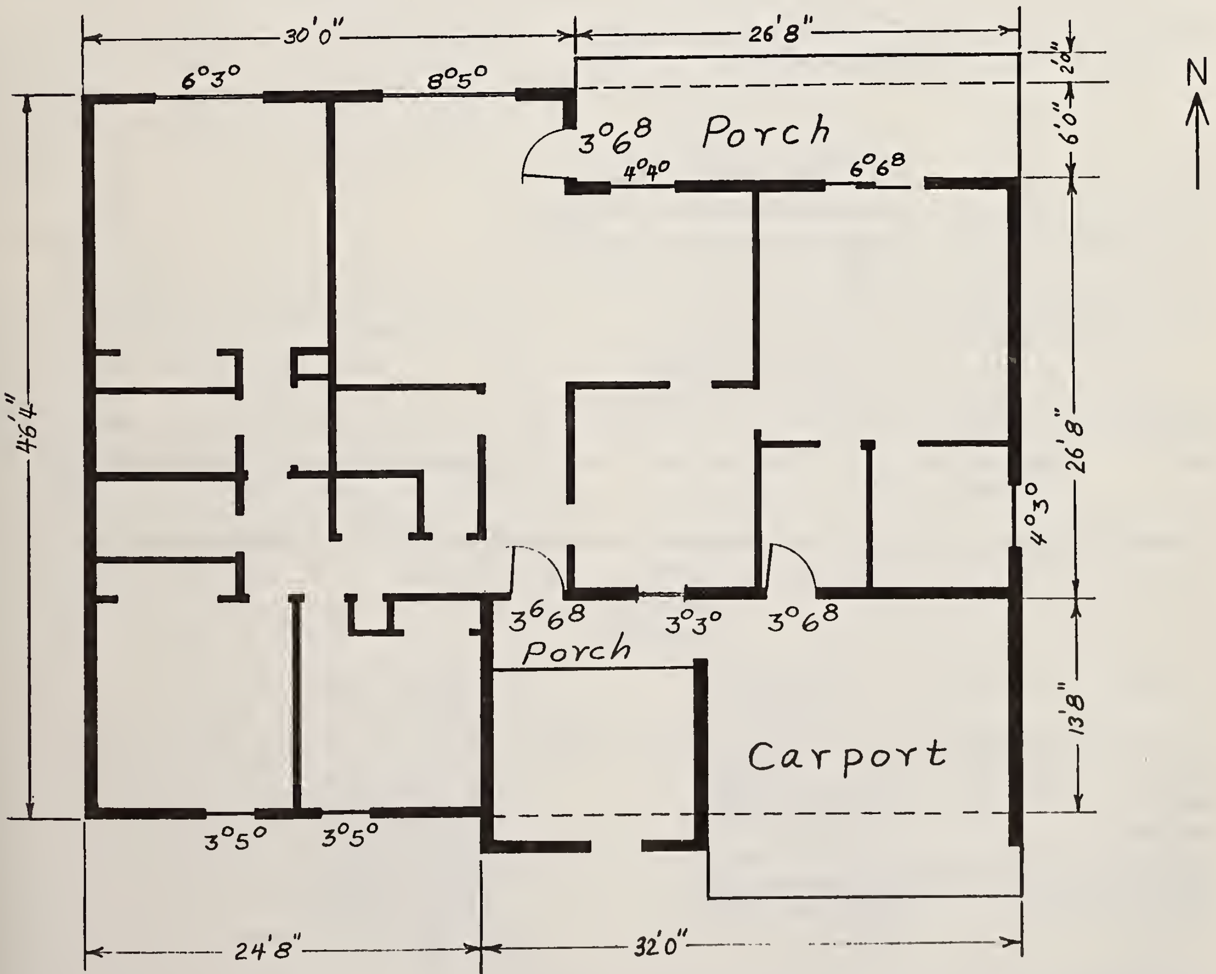


Figure 1. Floor plan for Slump-Block Territorial House (2028 square feet of floor area).

TABLE 10. SPECIFICATIONS FOR SLUMP-BLOCK TERRITORIAL HOUSE.

Item	Case A	Case B
Wall Construction	slump block, medium in color (α = 0.55), U_{wall} = 0.52 Btu/hr-ft^2-°F (2.9 W/m^2-°C)	same with R-8 additional Insulation, U_{wall} = 0.10 Btu/hr-ft^2-°F (0.57 W/m^2-°C)
Roof Construction	flat roof, built-up gravel, light or dark in color (α = 0.35 or 0.75), R-11 batt insulation, U_{roof} = 0.07 Btu/hr-ft^2-°F (0.40 W/m^2-°C)	same with R-19 additional insulation, U_{roof} = 0.03 Btu/hr-ft^2-°F (0.17 W/m^2-°C)
Windows	single pane, U_{window} = 1.06 Btu/hr-ft^2-°F (6.01 W/m^2-°C) Shading coefficient of 0.75	double pane, U_{window} = 0.54 Btu/hr-ft^2-°F (3.06 W/m^2-°C) Shading coefficient of 0.75
Doors	solid wood, dark color (α = 0.75), U_{door} = 0.53 Btu/hr-ft^2-°F (3.0 W/m^2-°C)	same
Floor	Slab on grade, summer heat gain/loss assumed negligible	same
Infiltration	1/2 air change in house per hour	same
Internal Heat Load	86,000 Btu/day (9.07 x 10^7 J /day)	same

TABLE 11. PREDICTED DAILY AVERAGE HEAT GAIN FOR SLUMP-BLOCK TERRITORIAL HOUSE IN TUCSON FOR MONTH OF JULY.

	Case A Heat Gain (in Btu)	
Item	Dark roof (α = 0.75)	Light roof (α = 0.35)
Roof	55,809 (12.8%)	27,733 (6.8%)
Walls	169,422 (38.8%)	169,422 (41.4%)
Windows	93,098 (21.3%)	93,098 (22.8%)
Doors	6,792 (1.5%)	6,792 (1.7%)
Internal Loads	86,000 (19.7%)	86,000 (21.0%)
Infiltration	25,708 (5.9%)	25,708 (6.3%)
Totals	436,820	408,744
Reduction by making Roof Light Colored	28,076 (6.4% reduction)	

(TABLE 11 continued)

Case B

Item	Heat Gain (in Btu) Dark roof (α = 0.75)		Light roof (α = 0.35)	
Roof	23,918	(9.6%)	11,886	(5.0%)
Walls	32,581	(13.1%)	32,581	(13.8%)
Windows	73,408	(29.6%)	73,408	(31.0%)
Doors	6,792	(2.7%)	6,792	(2.9%)
Internal Loads	86,000	(34.6%)	86,000	(36.4%)
Infiltration	25,708	(10.4%)	25,708	(10.9%)
Totals	248,407		236,375	
Reduction by making Roof Light Colored	12,032 (4.8% reduction)			

To convert the Btu values to joules, multiply by 1055.1.

loss as well. Beyond this, about the next most effective thing that can be done is to reduce the window solar radiation gain by additional window shading (no additional shading was added to windows between Cases A and B).

4. Conclusions

Measurement results have been presented which show that the solar reflectivity of materials commonly used in walls and roofs of southwestern residences varies over a rather large percentage range ($\sim 10\% < R_t < \sim 75\%$). On the basis of these measurements, a color classification scheme has been recommended for identifying the solar reflectivity and absorptivity of opaque building elements for the purpose of making heat-transfer calculations. Calculations of daily average sol-air temperatures and TETD's have been presented which show that varying the solar absorptivity can greatly change the heat gain or loss through a given building element. Thus, it is important to specify the solar absorptivity as closely as possible. Results have also been given which indicate that the surface film heat-transfer coefficient, $h_o(t)$, and the longwave radiation factor, $\Delta R(t)$, must be more accurately determined than is often done in simple analysis procedures if the predicted TETD values are to yield fairly accurate heat gain/loss determinations. Finally, roof heat-gain calculations have been presented which indicate that changing the roof color from dark to light does greatly reduce the summer roof heat gain of a typical southwestern house, but such a reduction has little effect on the summer total house heat gain because the roof heat gain is typically small to begin with compared to the total house heat gain.

5. Acknowledgements

The work reported here was not supported by any particular grant. The support of the University of Arizona, Department of Electrical Engineering, in assisting with the fabrication of the solar flux sensing probe and providing computer time for some of the required calculations is gratefully acknowledged. The authors also thank Miss Joann Main for her efforts in typing the manuscript.

References

1. Kreith, F., Principles of Heat Transfer, 3rd ed. (New York , N.Y.: Intext Educational Publishers, 1973.)

2. Moon, P., "Solar Radiation Curves for Engineering Use," J. Franklin Inst., Vol. 230 (1940), pp. 583-618.

3. Thekaekara, M. P., "Quantitative Data on Solar Energy," Energy Primer, pp. 25-26. (Freemont, Calif.: Fricke-Parks Press, Inc., 1974.)

4. Stimson, A., Photometry and Radiometry for Engineers. (New York, N.Y.: John Wiley & Sons, Inc., 1974.)

5. Huttenhow, J. D., "Design and Analysis of a Spectrally Narrow-Band Radiometer," Master's Thesis, Dept. of Electrical Engineering, University of Arizona, 1976.

6. Acklam, D. M. "A Measurement of Solar Reflectivity of Building Materials, Tucson, Arizona," Master's Thesis, Dept. of Electrical Engineering, University of Arizona, 1977.

7. Gubareff, G. G., J. E. Janssen and R. H. Torborg, Thermal Radiation Properties Survey. (Minneapolis, Minnesota: Honeywell Research Center, 1960.)

8. Reagan, J. A., "Energy Conservation for the Home or How to Lower Your Utility Bill," Engineering Experiment Station Report No. 45, University of Arizona, 1975.

9. "Heating Load" and "Air-Conditioning Cooling Load," Chapters 21 and 22, ASHRAE Handbook of Fundamentals, pp. 375-445. (New York, N.Y.: American Society of Heating, Refrigeration and Air-Conditioning Engineers, 1972.)

10. Stephenson, D. G. and G. P. Mitalas, "Cooling Load Calculations by Thermal Response Factor Method," ASHRAE Semiannual Meeting, Detroit, Michigan, Paper No. 2018, January 20 - February 2, 1967.

11. Peavy, B. A., F. J. Powell and D. M. Burch, "Dynamic Thermal Performance of an Experimental Masonry Building," Building Science Series 45, National Bureau of Standards, July 1973.

12. Kusuda, T., "NBSLD, the Computer Program for Heating and Cooling Loads in Buildings," Building Science Series 69, National Bureau of Standards, July 1976.

13. Duffie, J. A. and W. A. Beckman, Solar Energy Thermal Processes. (New York, N.Y.: John Wiley & Sons, Inc., 1974.)

14. Reagan, J. A., "Revised form of Computer Program Heat for Computing Equivalent Temperature Differentials," Department of Electrical Engineering, University of Arizona, all rights to program assigned to Efficient Energy Management, Inc., Tucson, Arizona, 1976.

15. Reagan, J. A., "Residential Energy Conservation, an Analysis of Potential Building Code Changes," Rept. to City of Tucson Planning Dept., City of Tucson, January 1978.

Questions and Answers

Elmer R. Streed, National Bureau of Standards, Washington, D.C.:

Have you performed any direct comparison between the solar reflectance obtained with your probe and an instrument using an integrating sphere such as used with ASTM Standard E424-71, on the same material? Since there is ∽ 40% of the incident solar radiation beyond 1.0 μm, and light-colored paints change optical properties in this wavelength region, comparison would be of interest.

J.A. Reagan: We have made a comparison between the solar reflectance measured by our probe and an integrating sphere system which covered a wider spectral range. These comparison tests were made for several dark-colored paints applied to smooth metal plates, and the reflectances measured by the two devices typically agreed within 3%. As noted in the paper and discussed in depth in reference 5, measurements have been made to verify the Lambertian response of our probe. The probe's response is sufficiently close to being Lambertian so that the measurement error in determining hemispheric flux, for the wavelength range covered by the probe, is less than ∽1%. With regard to the spectral response, the probe covers a wavelength range between about 0.44 to 0.96 microns, and as noted in the paper, this includes about 65→70% of the incident solar energy (at ground level). Most of the energy which is not detected falls in the longwave (0.96 to 2.0 μm) range. If the spectral reflectivity of a material is greatly different in this longwave range compared to the shorter wavelength (0.44 to 0.96 μm) range, then the reflectivity measured by our probe will be somewhat in error compared to the true or complete wavelength range solar reflectivity. For example, if the spectral reflectivity of a material changes abruptly by 25% between the shorter (0.44 to 0.96 μm) and longwave (0.96 to 2.0 μm) spectral ranges, then the reflectivity measured by our probe would be in error by about 8%. As noted in the paper, the reflectance of most materials and coatings used in buildings does not apparently deviate too greatly in this longwave region compared to the shorter wavelength region. Certainly, some paints and oxides which have "selective" wavelength characteristics could present problems. However, in most cases, we would estimate that the reflectance measured by our probe would agree within 5% or less with the complete wavelength range solar reflectance.

MEASUREMENT OF ATTIC TEMPERATURES IN FLORIDA

by

T. I. Wetherington, Jr.
Energy Conservation Research Engineer
Florida Power Corporation
St. Petersburg, Florida

Measurements of maximum temperature were made in 30 Florida attics. In some of these attics temperature profiles and humidity profiles were taken in an attempt to differentiate between the effects of roof materials, roof color, and ventilation methods. This paper reports on these measurements and the deviation from accepted theory that was observed, and suggests reasons for these deviations.

Our company's interest in attic ventilation and attic temperatures has been largely due to their impact on summer comfort. In 1973 we were approached by a manufacturer of attic ventilating equipment to assist in promoting his product on the basis of its value as an energy conservation device. We received a demonstrator unit and his literature, but upon inspecting the literature we found that he was talking in terms of attic temperatures on the order of 130°F (55°C). Having been personally involved in some work in my own attic a short while before, I was quite certain this manufacturer underestimated Florida attic temperatures.

We reasoned that the effect on air conditioning load would be indicated by the temperature at the top of the ceiling insulation. We obtained thermometers that would measure and retain the highest temperature attained and used available recorders from our instrument stock and started out testing attic temperatures. We mounted our sensors for the recorders and our thermometers at the top of the insulation or on the attic floor if the attic was floored above the ceiling insulation.

Our first reaction when we inspected our recorders at the end of the first test period was one of amazement, followed by a certainty that the instrument had failed. In our first week of measurement in several houses, the highest temperature that we found was under 108 degrees (Appendix I). Our recording charts had a 50-250°F (10-121°C) scale, with the result that virtually no deviation was visible because of the compression at the low end of the scale. When we checked the mercury thermometers, we found the maximum readings confirmed the results of the tape. As a double check, we actually took the thermometers into the attic and stayed with them while we observed the readings, and sure enough the attic was just as hot as we remembered it being, but the thermometer showed that human bodies make poor temperature measurements.

We expanded our data collection to homes which had different types of ventilation systems and different types of roofs and found that there was very little difference between hip roofs with only eave ventilation and homes with rather elaborate ridge vents and wind-turbine ventilation systems (Appendix II).

At this time we were only concerned with maximum temperatures. We typically would place the thermometer on a joist or on the insulation somewhere close to the center of the house, usually through the attic access door. On two of these houses we arranged to cover the ventilators on alternate days so that we could get a picture of the differences that occurred with ventilation and without augmentation of the ventilation. On these houses we found mixed results--in one case the surface temperature of the insulation was materially reduced and in the other it actually increased. The average for all of the days with or without ventilation for the two houses we tested showed a six-degree difference in surface temperature of the insulation in favor of ventilating with wind turbines.

After inspection of the data we obtained from our first series of tests, we noted the differences in attic temperatures seemed to be less dependent on roof color than on composition. All tile roofs, regardless of color, operated with lower attic temperatures than did asphalt shingle roofs.

In order to check out the differences between light and dark roofs, we selected three pairs of houses that were as nearly identical as we could locate and placed recorders in the attics. At the end of the week the charts were removed and a comparison was made of the attic temperatures in each of the houses. Figures 1 and 2 display the results of group 1,where there was a slight difference in favor of the dark-colored roof. In group 2 (Figures 3 and 4) there was a difference in favor of the light roof. In the third pair of houses (Figures 5 and 6) there was a difference in favor of the dark roof. Since there was little difference in the structures we concluded that factors other than roof color were probably more significant in determining the heat gain of a house.

We also tried to extend the number of houses sampled to see if we could detect any big difference on the basis of total energy consumption. At the time, we were making a study of meter reading, and the industrial engineer who was observing the process made some notes on the color of the roofs of the houses along his meter reading route,along with the direction the houses were facing, and whether any ventilation systems such as turbine ventilators or power roof vents were visible.

In this study we did find a lower average use of energy for the homes that were observed to have light-colored roofs than for the homes with dark-colored roofs. The homes observed to have attic ventilation had slightly higher energy usages than the ones that were not observed to have ventilation equipment. Homes facing north had lower energy use than did homes facing south or east. The highest energy use was in homes facing west. No correlations were made between direction, color, and ventilation (Appendix III).

Except for the north-facing homes, the differences between energy uses did not appear to be large enough to warrant any conclusions except possibly that the home would be better off facing south and possibly that where there is a choice, the roof color should be light.

In evaluating the tests to date,we reached a general conclusion that insulation in the attic was probably more valuable than ventilation, roof color, or roof texture, because the measured differences in temperature did not appear to warrant predicting a major difference in energy use due to any of these factors.

Since we have not seen any major differences in temperature due to forced and natural ventilation, we decided to go one step further and completely seal an attic and observe what happened in the house and in the attic. Figure 7 is a picture of the house we used and Figure 8 shows the results. There were moderate increases in the maximum temperatures reached when the attic was sealed as compared to when it was ventilated, but the average temperature was lower when the attic was sealed. Another thing we observed was that the temperature rose and fell more sharply in the sealed attic than in the open attic.

This house is typical of most homes in the Florida area in that they are constructed without a vapor barrier in the ceiling. When we sealed the attic the dew point in the attic dropped down to approximately the dew point of the conditioned air inside the house. The reduced moisture loading of the house due to reduced attic vapor pressure resulted in a decrease in the relative humidity inside the house (Figure 9). This change in humidity may well explain the steeper slope that we see on the temperature changes in the attic. A typical attic will have 2,000 to 3,000 lbs. (907-1,360 kg) of wood exposed to the attic air. In midsummer the dew point of the outdoor air will go as high as 70°F (21°C) and along towards the end of the summer it will go as high as 75°F (24°C). When this is contrasted to a dew point of approximately 60°F (16°C) for the inside conditioned air, there is a big difference in relative humidity seen by the wood structure of the attic. These changes could easily result in a change of 10 percent in moisture content of the wood with 200-300 lbs. (91-136 kg) of moisture being absorbed and evaporated from the wood of a ventilated attic. This would of course result in a stabilizing effect on the sensible temperature of the attic as well as changing the specific heat of the wood.

These results suggest an automatic shutter for attic vents that will open in winter and close in summer.

Our work in the field of attic temperatures has been rough by laboratory standards; however, we have made enough measurements in a variety of houses to feel reasonably comfortable with our conclusions, as follows:

1. Attic ventilation makes a slight reduction in the top surface temperature of the ceiling insulation.

2. Attic temperatures are not as high as we thought they were and may not be high enough to be a significant problem as long as insulation thicknesses are proper.

3. Where air conditioning ducts are in the attic, any heat gains to the ducts or air losses from the ducts will have a tendency to reduce insulation surface temperatures above the ceiling and the less air that moves through the attic, the more effect these gains have in holding attic floor temperatures down.

4. Roof color does not appear to be as significant as the texture of the roof in determining attic temperatures.

5. Sealing the attic in summer did produce a significant drop in indoor relative humidity and a reduction in average attic temperature for the house where this experiment was conducted.

As we have started to emphasize energy use considerations rather than peak load considerations in our studies, we are observing different aspects of attic temperatures than we did in the past. Where we were concerned with a perhaps transient high temperature, we are now more concerned with the prolonged and typical conditions imposed during the summer.

We see that there is some very good work going on in pinpointing the actual heat flows and the conditions that cause them so that we can design greater energy efficiency in our roof and attic construction. Accurate information will allow us to put our construction dollars where they will do us the most good in the energy conservation picture.

Recorder located on joist nearest hallway- attic access.

FIGURE 1

8/5/77 Friday	Black	120	80
	White	122	82
	Difference(2)	- 2	- 2
8/6/77 Saturday	Black	123	78
	White	123	81
	Difference	0	- 3
8/7/77 Sunday	Black	123	77
	White	123	80
	Difference	0	- 3
8/8/77 Monday	Black	125	80
	White	126	82
	Difference	- 1	- 2

(1) Outdoor conditions: high in low to mid-90's, low in mid-70's.

(2) Difference = Black minus white.

FIGURE 2

FIGURE 3

Date		°F Maximum	°F Minimum	°F St. Pete Hi/Lo
8/12/77 Friday	Black	127	80	92/73
	White	121	78	
	Difference	6	2	
8/13/77 Saturday	Black	107	79	83/72
	White	100	76	
	Difference	7	3	
8/14/77 Sunday	Black	115	80	89/75
	White	110	79	
	Difference	5	1	
8/15/77 Monday	Black	126	80	91/75
	White	120	79	
	Difference	6	1	
8/16/77 Tuesday	Black	128	80	92/73
	White	122	80	
	Difference	6	0	
8/17/77 Wednesday	Black	125	79	91/76
	White	117	78	
	Difference	8	1	

FIGURE 4

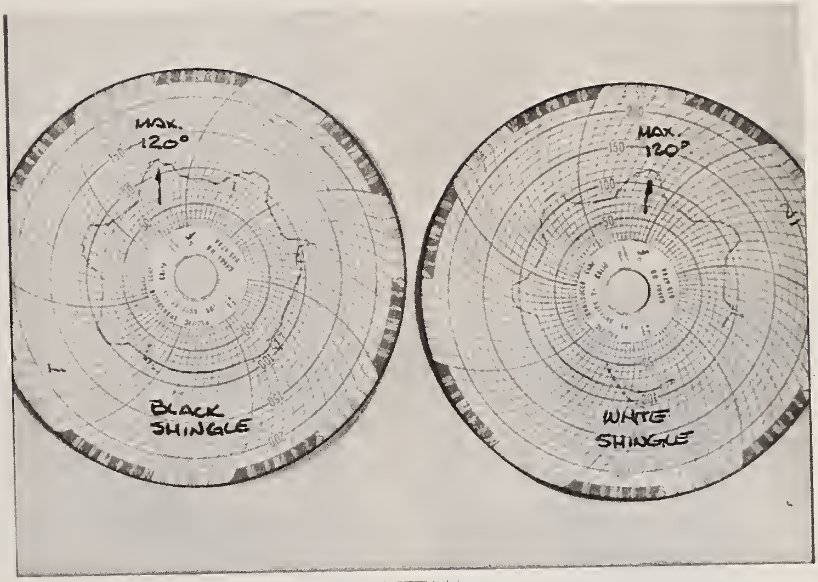


FIGURE 5

Date		°F Max.	°F Min.	°F St. Pete Hi/Lo
8/21/77	Black	90	N/A	89/79
Sunday	White	90	N/A	
	Difference	0		
8/22/77	Black	85	80	85/74
Monday	White	85	80	
	Difference	0	0	
8/23/77	Black	101	79	85/73
Tuesday	White	110	70	
	Difference	-9	0	
8/24/77	Black	112	79	91/75
Wednesday	White	119	78	
	Difference	-7	1	
8/25/77	Black	120	80	91/71
Thursday	White	120	80	
	Difference	0	0	
8/26/77	Black	116	80	92/77
Friday	White	119	80	
	Difference	-3	0	

FIGURE 6

FIGURE 7

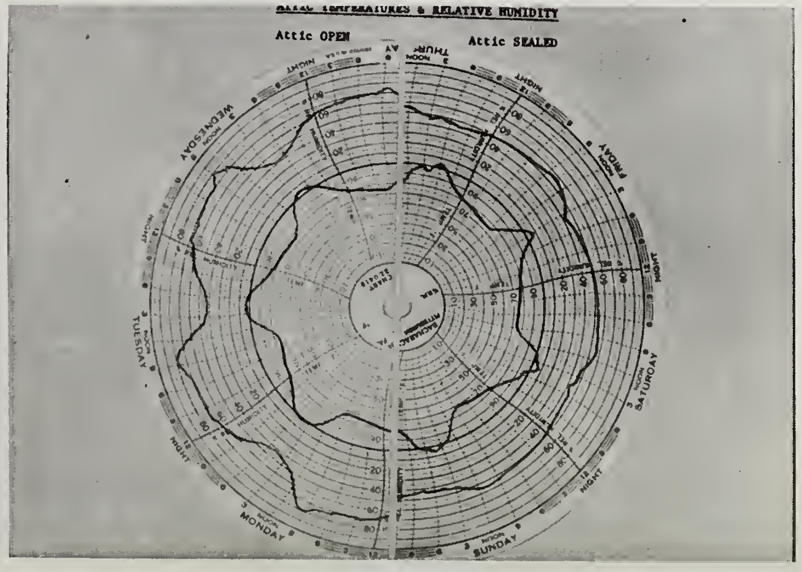


FIGURE 8

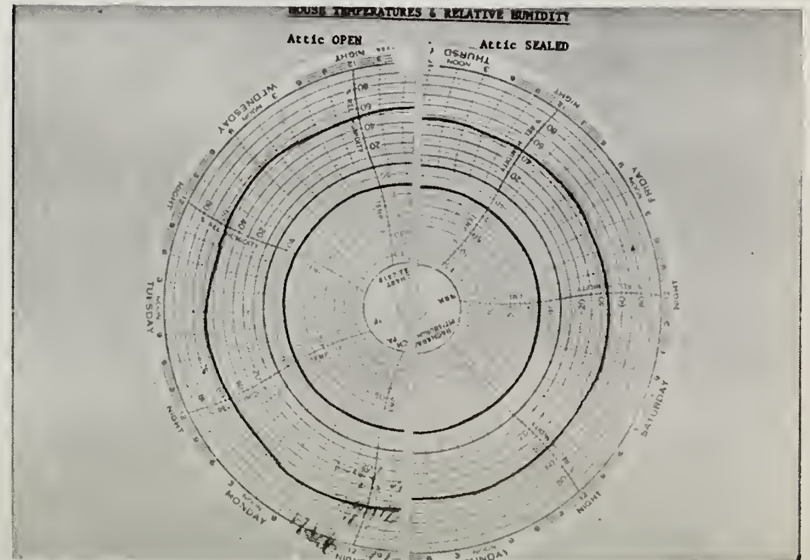


FIGURE 9

APPENDIX I

HOW HOT DOES IT GET IN THE ATTIC SPACE???

Investigation was started in the latter part of the summer to determine attic temperatures on hot summer days. The findings were to be used in determining the need for electrically powered attic exhaust fans or additional ventilation.

All of the attics checked were for approximately one week with a maximum reading thermometer located at floor level and when outdoor ambient temperatures were 90°F or above. In most cases it was attempted to check temperatures where the owner claimed an extremely hot attic space. Most of the attics were of the hip roof type with no gable vents.

House No.	Maximum Temperature
1	100°F
2	89
3	108
4	80
5	92
6	106
7	96
Average	96°F

It is anticipated that this investigation will continue to obtain more conclusive data but from the above data it would appear that a power exhauster is not required and may even increase the energy consumption.

Certainly to date the attic temperatures are much lower than the 130 to 150°F temperatures often quoted.

***

APPENDIX II

House Number	Season	Test Length	Ambient	Roof Type	Vent Type	Turbine Operating Top	Turbine Operating Insulation	Turbine Not Operating or No Turbine Top	Turbine Not Operating or No Turbine Insulation
1	1973	7 Days	90 +	White Tile Gable	Gable Eave	-	-	-	100
2	1973	7 Days	90 +	White Tile Hip	Eaves	-	-	-	89
3	1973	7 Days	90 +	Tan Asphalt Shingle-Gable	Eaves	-	-	-	108
4	1973	10 Days	90 +	White Tile Hip	Eave	-	-	-	80
5	1973	3 Days	90 +	Grey Tile Gable	Gable	-	-	-	92
6	1973	30 Days	90 +	Gable Asphalt Green	Gable Eave	-	-	-	106
7	1973	10 Days	90 +	White Tile Hip	Eave	-	-	-	96
8	1974	6 Days	90-93	Dirty-White Asphalt-Gable	2 Turbine Eave & Gable	124	114	129	119
9	1974	7 Days	90-91	Charcoal-Grey Asphalt-Gable	Eave & Gable	-	-	131	112
10	1974	4 Days	88 +	Charcoal-Grey Asphalt-Gable	Eave & Gable	-	-	115	104
11	1974	14 Days	90-93	Brown-Red Asphalt-Hip	4 Turbine Eave	134	117	133	110
12	1974	3 Days	90-92	Tan Built-Up Gravel-Gable	Eave & Gable	-	-	-	112
13	1974	21 Days	90 +	Gable Grey Asphalt	Gable & Eave	-	-	-	106
14	1974	10 Days	90 +	White Asphalt-Gable	Eave	-	-	120	108

APPENDIX III

AVERAGE MONTHLY POWER CONSUMPTION

Parameter	KWH	Avg.No.of Customers
Dark Roof	1,610	12.6
Light Roof	1,508	71.5
Unknown Roof color	1,530	36.2
North facing	1,387	28.8
South facing	1,535	33.2
East facing	1,541	22.5
West facing	1,596	29.5
Turbine Vents	1,549	29.9
Unknown ventilation	1,516	87.7

Questions and Answers

John Felter, AVA, Houston, Texas:

Why does not the computer at the Florida Power Company use actual electric bills of well-ventilated houses compared to non- or poorly ventilated houses, as test results? --Could be for six months--100 houses--definite results. Could not be as costly as what you did.

T.I. Wetherington, Jr.: Quality of ventilation is a subjective evaluation without extensive study and measurement of each house. The problem of making such a computer study as you suggest is in identifying and categorizing the houses. Our use of visible ventilation devices as shown in Appendix II was one attempt to categorize houses for computer studies.

Home Ventilating Institute (HVI). Two questions with responses by the author.

1. The paper draws attention to the high temperature of 108°F and average of 96°F at attic floor level for seven houses in 1973 (Appendix I). However, for another seven houses tested the next year, the high was 119° and average maximum 110° (Appendix II) -- an average 14° hotter than that of the first group, and 32° above a 78° air conditioner setting. The hotter seven readings seem at least as important as the other seven in generalizing about attic temperatures.

Wetherington: Most of the second group had asphalt shingle roofs and the average maximum was only slightly above the two houses in the first group that had asphalt shingles. As previously noted, houses with tile roofs had cooler attics than those with asphalt-shingled roofs.

2. No data on air conditioner energy use are presented on the experiment of blocking off all ventilation from the attic. Questioners suggested that heat gains through attic cooling ducts and cold air leaks from the ducts may be the price, with an abnormally high air conditioner load. How can the sealed-attic theory be evaluated for energy- and cost-effectiveness without measured comparative data on air conditioner energy use?

Wetherington: Air conditioner energy use was not measured but it can be reasonably inferred that air conditioning load decreased when the attic was sealed because the indoor relative humidity dropped, indicating a reduction of latent load. The air conditioning duct system performance appeared normal when compared through attic temperature profiles of other attics of similar homes. The reduced average temperature of the sealed attic indicates that the net duct losses were reduced when the attic was sealed.

FORCED VENTILATION FOR COOLING ATTICS IN SUMMER

by

Gautam S. Dutt
David T. Harrje
Center for Environmental Studies
Princeton University

The potential for air conditioning energy savings using exhaust fans to cool attics was investigated in six occupied townhouses at Twin Rivers, N.J. These houses were compared with similar houses without attic fans. The houses had various levels of instrumentation. Data collected for two summer months in 1977 was the basis for this study. The principal quantities measured were attic and living space temperatures, air conditioner and attic fan usage, together with outside air temperature and solar flux. The attics with fans were substantially cooler. However, the corresponding reduction in heat flux into the living space through the attic insulation is a very small part of the house air conditioning load. Any difference between the air conditioner energy use between houses with and without attic fans is not discernible from other factors which lead to house-to-house variation in air conditioner use.

Key words: Attic fans; air conditioner energy; ventilation.

1. INTRODUCTION

There has been some controversy on how much energy savings, if any, may be realized from the use of fans to ventilate attic spaces [1,2,3]. Both theoretical studies and experimental data have shown that the use of attic fans led to no net energy savings. [4,5] On the other hand, some have claimed large energy savings from fan use [3]. The study reported here was carried out in occupied townhouses at Twin Rivers, New Jersey. Comparisons of air conditioner usage in houses with and without attic fans, made for the same period in summer 77, show no difference which may be ascribed to the use of attic fans. The attic fans were installed late in 1976. No change in the air conditioner energy use pattern between the summers of 1976 and 1977 is discernible either. Simple theoretical calculations indicate that no significant reduction in air conditioner energy use was to be expected from the use of attic fans. The data indicate that the attic fans operated continuously for many hours on hot days and consumed a significant amount of energy. When the energy used by the fan is combined with the air conditioner energy use, it is apparent that the total energy consumption increases when attic fans are used. Moreover, the peak electrical demand also increases. Thus, there is little justification for the use of attic fans at Twin Rivers, either for reducing energy consumption or reducing peak electrical demand.

The method of analysis and the results are presented in Section 2, while a discussion of the results and relevance to other houses is contained in Section 3.

2. THE EXPERIMENTS IN CONTEXT

Since 1972, members of Princeton University's Center for Environmental Studies have been examining residential energy use in a number of townhouses at Twin Rivers [6]. Of these townhouses, twenty six three-bedroom units were instrumented using the 'OMNIBUS' package. In this instrument package, data are collected hourly on 12 channels. Data included living space and attic temperatures, electrical energy use, water heater and air conditioner on-times. In addition, two other houses, instrumented by the National Bureau of Standards, were available for study. In these houses, data from about fifty channels could be recorded as often as every five minutes. This has been named the 'RAPIDSCAN' data acquisition system. One of the Rapidscan houses was also part of the Omnibus set so that this house had two data acquisition systems operating simultaneously. The principal weather variables -- air temperature, wind velocity and direction--and solar flux on a horizontal surface were recorded every twenty minutes at a local weather station.

The study reported here shows data from five townhouses. All five had Omnibus instrumentation and are identified as TR9, TR13, TR16, TR18, and TR27, according to the instrument serial number. Of these, TR9 was also part of the Rapidscan data acquisition system. TR13 did not have an attic fan - the others did. All the houses had received various energy-saving retrofits prior to the start of the attic fan study. The differences between the houses and their instrumentation packages are set out in Table 1. The common data channels

TABLE 1. HOUSE DESCRIPTION AND INSTRUMENTATION

	TR 9	TR 13	TR 16	TR 18	TR 27
Attic insulation	R-30	R-30	R-11	R-11	R-30
Furnace shaft in attic	sealed	sealed	sealed	sealed	sealed
Gaps on attic floor near party walls	sealed	sealed	as built-not sealed	as built-not sealed	sealed
Appliances: water heater, range, clothes dryer	elec.	elec.	elec.	gas	elec.
Attic fan Capacity (cfm) Power (watts)	1000 (2.7A)	No	709 214	713 209	697 214
Air conditioner	All houses had 24000 Btu/hour units which consumed 3.2kW				
Omnibus instrumentation active from:	4/22/75	6/24/75	7/2/75	12/10/75	8/13/76
No. of attic temp. measurements	19	1	5	5	5
Attic fan-on measurement	June, Aug 77	-	Aug 77	Aug 77	Aug 77

for these houses were:

Basement temperature
Downstairs temperature
Upstairs temperature
Air conditioner on-time
Water heater on-time
Thermostat setting.

The standard 'Omnibus' house is equipped with only one attic thermistor, located at mid-height halfway between the trapdoor and the closer party wall. In TR16, TR18, and TR27, there were four additional thermistors, under insulation, on floor (2), and below roof.

The Rapidscan measurement in TR9 included two temperatures on the underside of the roof, one for the East-slanted roof and one for the West-slanted roof. There were also two infrared radiometers (sometimes known as pyrgeometers) installed in the attic - one pointing up and the other down - to measure the infrared radiation emitted by the roof and the floor of the attic. Major appliances were also monitored.

3. DOES THE ATTIC FAN SAVE ANYTHING?

A) One house - one summer

In this section, some of the results of our study and analysis are presented. There are a number of different ways of determining the influence of the attic fans on the heat balance of the house. One way is by comparing days of similar weather and household occupancy with the attic fan on and with the fan off. During the summer of 77, the attic fan switch was turned off for two weeks. (Note that the fan is thermostatically controlled so that the fan does not run continuously even when the switch is turned on.) It was difficult to watch the weather between days with the fan on and fan off even when the outside temperature and solar flux were the only two weather variables considered. The reason, apparently, is that the summer period in New Jersey is short and extremely variable in temperature and cloudiness. This method, which has been used by other researchers [5], could not be used for our data set.

B) Two houses - one summer

The second approach was to examine the changes in a number of variables with the time of day, averaged over long periods, - one month or longer. Twenty four data points are created for each variable, one for each hour of the day. In this data reduction, the day-to-day "random" weather patterns would be eliminated but the aggregated variables would still retain the average variation with time of day. One period considered consists of about four weeks during August 1977. The average outside temperature and horizontal solar flux at Twin Rivers for this period are shown in Fig. 1. The corresponding variation of air conditioner use, attic fan use (where applicable) and attic-upstairs temperature differences were calculated from data for a number of townhouses, six of which had attic fans installed. Two houses were chosen that had attic retrofits A and D*, i.e. they had R-30 ($5.29m^2$ °C/W) insulation on the floor, and openings around the furnace flue and along the party walls had been sealed. Moreover, both houses were oriented the same way - the windows and doors face roughly east and west. The principal physical difference between the two houses is that one of them (TR27) was equipped with a thermostatically controlled attic fan while the other (TR13) was not.

Fig. 2 shows the average time-of-day variation of T_A - T_U (attic temperature** minus

*For a description of Twin Rivers retrofits see Ref. 7.

**Unless otherwise stated, attic temperature means the mid-attic air temperature and is denoted by T_A. For consistency all the computations are carried out using this value, which was measured by a thermistor in the same location in all attics. The errors involved with this simplification are discussed in the Appendix.

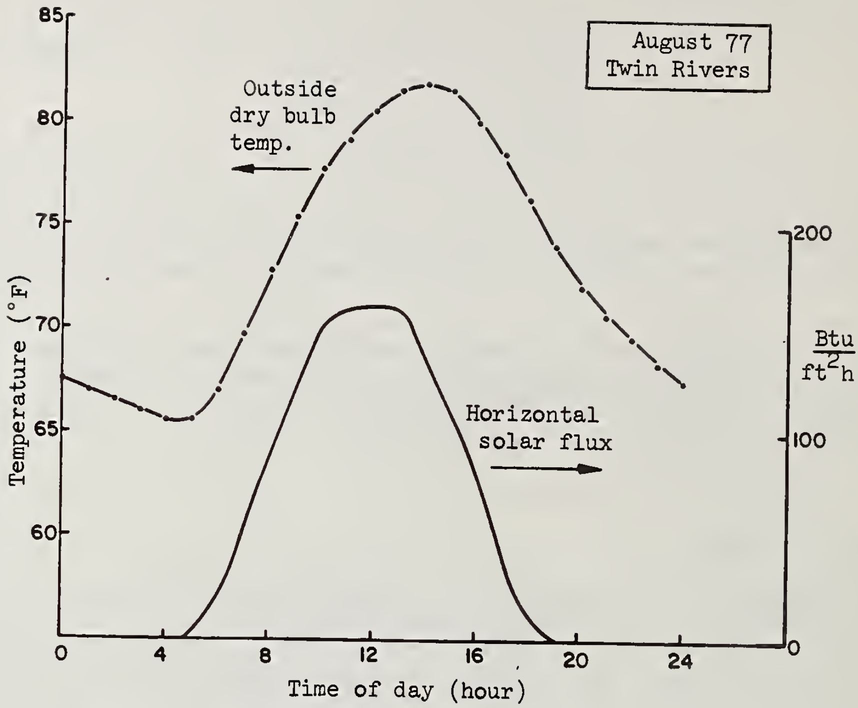
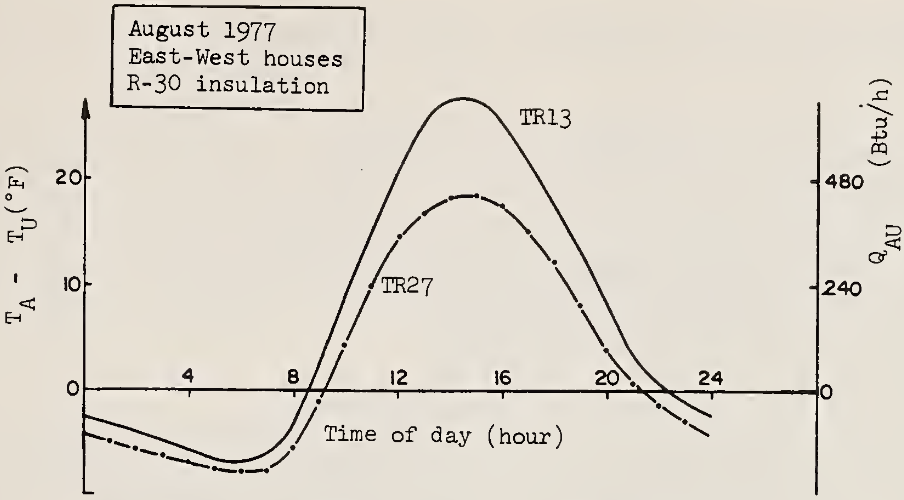


Figure 1. Average hourly variation of solar flux and outside air temperature.

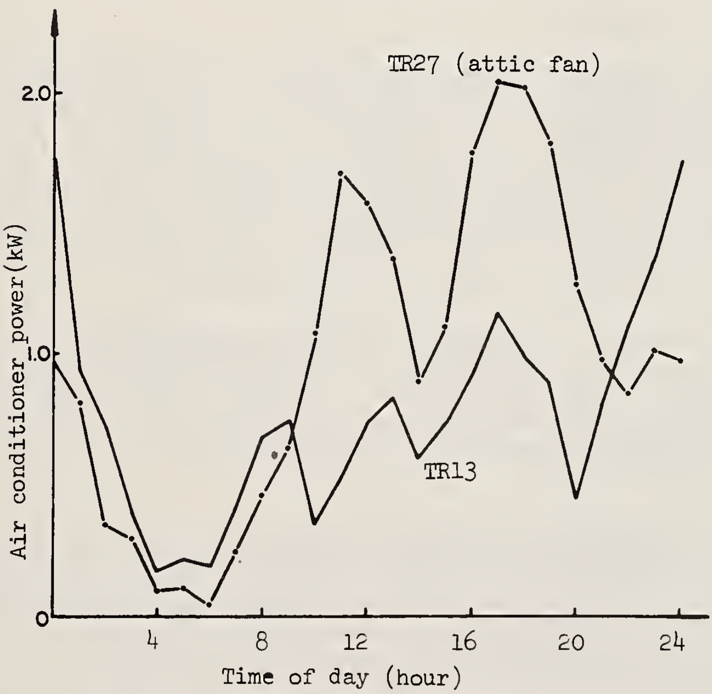
upstairs temperature), air conditioner input power, and attic fan on-time for August 1977. Several features of Fig. 2 stand out. The daytime attic-upstairs temperature differences in the "fan" attic are substantially lower, the maximum reduction in temperature difference being 8.9°F (4.9°C). The graph for T_A - T_U also shows the heat transfer rate from attic to upstairs corresponding to the temperature difference, obtained with an R-value for the attic floor of 26 (4.58 m^2 °C/W).* The reduction in heat flow for an 8.9°F (4.9°C) drop in T_A-T_U amounts to 240 Btu/h (70 watt). Since the 24000 Btu/h (7.0kW) air conditioner requires 3.2kW to operate, the corresponding reduction in air conditioner power is 32 watts, a small part of the total air conditioner power in the middle of the day - typically 1000 to 2000 watts. The average power consumption by the attic fan at this time is 160 watts and far exceeds the reduction in air conditioner power.

At other hours of the day, the reduction in ceiling heat flux and corresponding savings in air conditioner power are smaller, but the attic fan power remains large for most of the day (Fig. 2). The daily averages of ceiling heat flux, air conditioner and attic fan energy use for August 77 may be obtained from the area under the curves in Fig. 2. For instance, the area under the attic fan on-time curve in Fig. 2 (c) indicates that the attic fan is on for 6.82 hours on an average day, thus consuming 1446 watt-hours. The average daily ceiling heat fluxes for the two houses may be obtained from the area enclosed by the T_A - T_U curves in Fig. 2(a) and the horizontal axis. The difference between the two values is the reduction in ceiling heat flux on an average day — 2147 Btu (2264kJ), again assuming that the ceiling is R-26 (4.58 m^2 °C/W). This drop in heat flux will result in a reduction in air conditioner energy use of 286 watt hours on an average day. The average daily air conditioner energy uses, obtained from Fig. 2(b), are 17.6 kWh and 23.4 kWh for TR13 and TR27 respectively. The reduction in daily air conditioner energy use of 286 watt hours

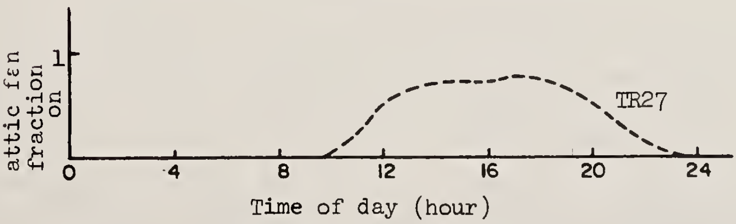
* The insulation effectiveness for two of the six Twin Rivers townhouses had been evaluated by the National Bureau of Standards. One of these houses had R-11 insulation, and the other R-30. The overall R-values of the attic floor were measured to be 10.3 and 26.4, respectively [11]. These values are within 12% of the nominal values of the insulation itself.



a) Attic - upstairs temperature differences



(b) Air conditioner power consumption



(c) Attic fan on-time

Figure 2. Average variation with time of day

brought about by attic fan use is a negligible part of the total air conditioner energy use. The attic fan uses an additional 1446 watt hours of energy per day, so that the total energy consumption for cooling is higher by (1446 -286) watt hours if attic fans are used.

The most striking feature of the data in Fig. 2(b) is that the air conditioner use patterns in the two dwellings are quite different. The house with the attic fan, TR27, uses more energy during the day. Large differences are also seen between the air conditioner power for the other houses in our study. Even though the units are nominally identical in construction, where minor differences might explain some of the variation in air conditioner use, the bulk of the variation is believed to depend on the behavior of the resident. In reference 8, Sonderegger showed statistically that a major part of the variation in heating energy use is ascribable to resident behavior. Air conditioner energy use is even more resident-dependent for the following reasons:
(a) Cooling need is more discretionary.
(b) The inside-outside temperature differences are much smaller in summer so that thermostat readjustment of 1° has a larger percentage effect.
(c) For a significant portion of the summer, the outside temperature is low enough to permit cooling by opening windows. But if windows are not opened then excessive internal heat built up from appliances, people and the sun can only be removed by the air conditioner. The ability or desire to take advantage of cool outside air, therefore, may make a noticeable difference in air conditioner use.

C) One house - two summers

Given these unavoidable house-to-house variations in air conditioner energy use, it is difficult to discern the effect of small energy-saving retrofits unless a very large sample is available for comparison. The alternative, of comparing a house for two intervals in the same summer with and without the attic fan on, was precluded earlier because of the variability of summer weather and the lack of exact comparison situations. A third approach to identify the effect of the attic fan was to look at data from the same house for two different summers where the attic fan was installed in the intervening winter. Instead of attempting to find identical days in the two periods, this method identifies the dependence of air conditioner use on weather by scatter plots and regressions.

The relevant weather variables are outside air temperature, solar flux, wind velocity and direction, and outdoor humidity. The data may be aggregrated into hourly or longer intervals. In order to reduce the effects of thermal storage, longer time intervals are desirable. However, the air conditioner use between midnight and sunrise is low and the attic fan never runs during this period. The daylight period - 8 a.m. to 8 p.m. - was the interval examined in this study.

Woteki has shown that both daily (24-hour) and twelve-hour (8 a.m. to 8 p.m.) air conditioner use may be equally well modeled whether one takes "cooling degree days" or "cooling degree hours" as the independent variable [9]. His analysis also shows that regression fits are not improved by including the solar flux, wind, or humidity. This lack of fit is surprising but is convenient because we may model the air conditioner use by a single weather-dependent variable, the inside-outside temperature difference.

For the present study the average temperature difference between outside and inside (T_O - T_I) between 8 a.m. and 8 p.m. was taken to be the independent variable. The inside temperature itself was obtained by averaging the house upstairs and downstairs temperature readings. The dependent variable was the average air conditioner power (A/C) during the interval. Scatterplots of A/C vs T_O - T_I for TR16 and TR18 are presented in Fig. 3 and 4 for both summer 76 and a part of summer 77. The periods covered are from June to mid-September of 1976 and June and August of 1977. Both attics have R-11 insulation (1.94 m^2 °C/W) and fans were installed late in 1976.

The most important observation to be made is that the scatter plots for the two summers in both houses overlay each other. The air conditioner energy use patterns for the summers with and without the attic fan are indistinguishable.

Thus, using three different methods of analysis we were unable to discern any energy savings brought about by the use of attic fans. Data were presented for four houses - two

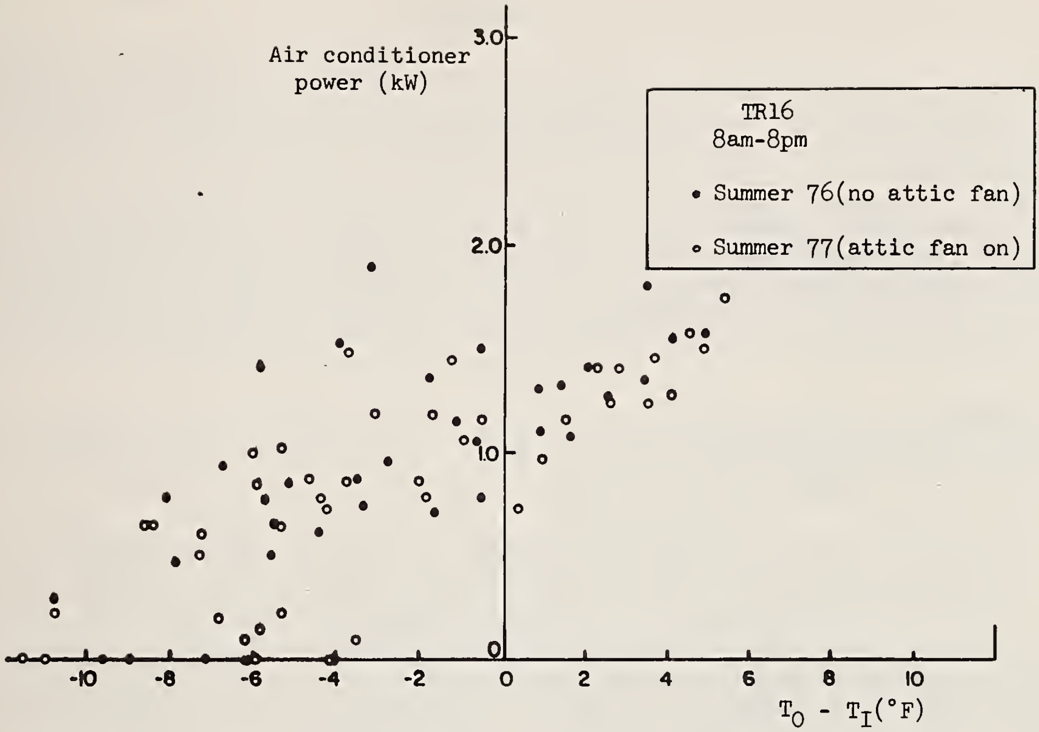


Figure 3. Air conditioner power vs. indoor-outdoor temperature difference

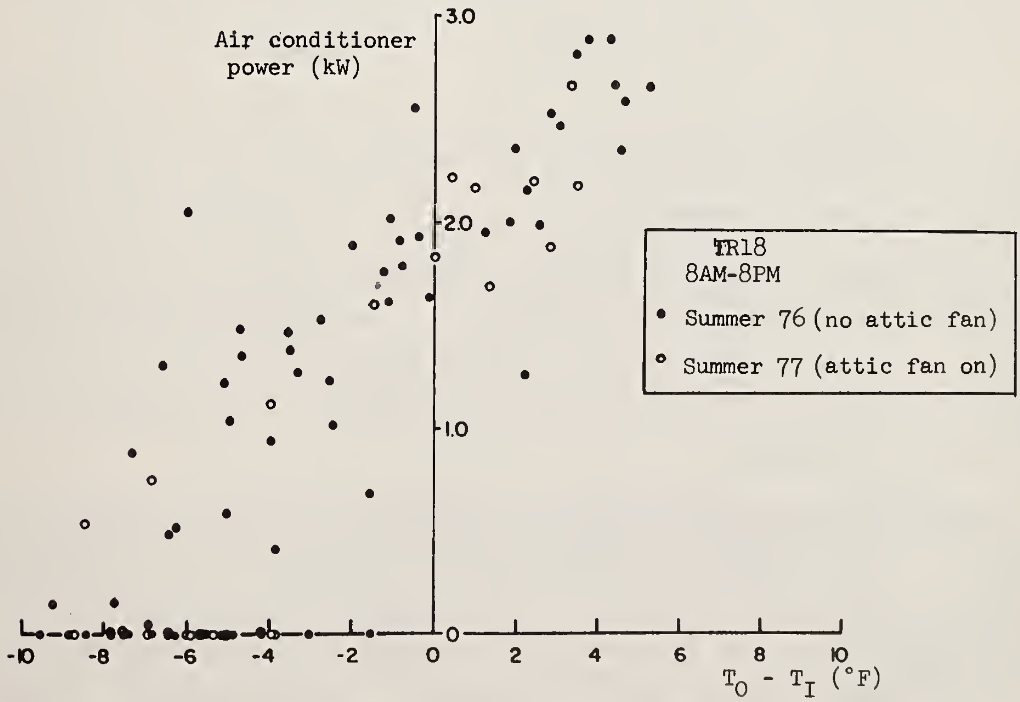


Figure 4. Air conditioner power vs indoor-outdoor temperature difference

with R-11 (1.94 m^2 °C/W) insulation on the attic floor and the other two with R-30 (5.29 m^2 °C/W)

The second approach showed that the maximum attic cooling from the use of attic fans would result in negligible air conditioner energy savings and would in fact increase both the total energy used (by the air conditioner and attic fan) as well their peak electrical power requirement.

The summer in Twin Rivers, New Jersey, is relatively mild - at least there are many days when it is cool outside, i.e. $T_O < T_I$. It has been claimed that on mild days, use of attic fans may eliminate the need for air conditioning altogether and result in energy savings. However, even for mild days, when the hot attic is a larger component of the cooling load, we do not see reduction in air conditioner use when attic fans are operating (Figs. 3 and 4).

4. DISCUSSION

The attic fans tested in this study showed no savings in net energy used for cooling. These results match those of other researchers in different climatic regions of the U.S. [1,4,5].

The reason that attic fans do not save energy is because the heat gain through R-11 (1.94 $m^2 \cdot$ °C/W) or R-30 (5.29 $m^2 \cdot$°C/W) insulation is small (even though the attic is quite hot) and the fan requires more energy to operate than the savings from reduced ceiling heat gain. If the insulation level was much less, then attic fans would actually save energy, although even in that case adding insulation would save energy in a more cost-effective way. The optimum level of insulation, determined from winter energy requirements, is quite large - larger than the present standards - and larger than the R-30 (5.29 m^2 °C/W) used here [5]. For such highly-insulated attics, the ceiling heat gain during summer would be so small that attic fans would be even less effective than for present houses.

One way by which forced ventilation may be effective in reducing cooling energy needs is in whole house fans. For instance, a glance at Fig. 3 or 4 reveals that much of the air conditioner use took place when the outside was cooler than the house interior: $T_O - T_I < 0$. In such a case ventilation should replace the air conditioner to save energy. The optimum ventilation system to meet this need warrants further study.

We have indicated why expending energy to cool the attic is a losing proposition in heavily-insulated attics. But if increased ventilation could be obtained without power, then it may be cost effective although the percentage savings would be very small. Most natural ventilation systems are wind-dependent, and increased ventilation entails increasing the vent opening area. This is beneficial in summer but increases winter heat loss so that it is not an ideal system. One venting arrangement - the ridge vent - takes advantage of the stack effect in the attic. The venting rate is much higher in summer than in the winter, which is preferable [10]. Other venting arrangements may be devised, using a reflective material below the roof joists to both cool the roof directly by the chimney effect and isolate the attic floor from infrared roof radiation. Such a "solar-powered" attic vent would be most effective on sunny hot days, when it is most needed. Other arrangements, including closing and opening vents seasonally, may also be devised for optimal venting. However, it should be borne in mind that the heat gain through an insulated attic is small and only small energy savings may be expected.

Some energy conserving measures, like increased attic or wall insulation, are probably effective both in the summer and winter in making the living space comfortable with reduced energy use. Other conservation measures have to be selected to optimise the house for both summer and winter, e.g. location of windows and overhangs to maximize winter solar heat gain and minimize summer solar heat gain. A third category of conservation measures is based on separate optima for summer and winter but involving a simple changeover in between. The optimum strategy for attic ventilation falls in either the second or the third category. Double season optimization is a promising direction for future conservation research.

Appendix

Temperature Distribution in Attics

The sun heats the roof, which conducts heat to the attic air. Also, the hot underside of the roof radiates infrared energy directly to the attic floor. It is the temperature on the attic floor surface that primarily determines the extent of heat transmission to the living space below the attic. Therefore, we need to know the attic floor temperature and how it is altered by the fan. We will show in this appendix that the heat transmitted into the living space, calculated using the mid-attic air temperature, is somewhat larger than that based on the attic floor surface temperature. Furthermore, we will show that this calculation exaggerates the <u>reduction</u> in cooling load brought about by the use of the fan. Thus the energy savings calculated in Section 2 using the data of Fig. 2 are larger than the actual savings, i.e. the attic fan reduces cooling load even less.

Let T_A, T_{AF}, T_U be the average temperature of the mid-attic air, the attic floor surface and the upper floor air respectively. The heat conduction through the attic insulation is

$$Q = U_F A_F (T_{AF} - T_U)$$

where U_F is the thermal conductance of the attic floor and A_F is its area. If the mid-attic temperature is used to estimate Q, then this estimate is

$$\bar{Q} = U_F A_F (T_A - T_U)$$

The question is how does $\bar{Q}$ compare with Q. Six of the houses had thermistors both in mid-attic and on the floor surface. During the day the mid-attic air was always warmer than the attic floor, so that $\bar{Q} > Q$. A typical day with no attic fan operating in TR 9 is depicted in Fig. 5. Denoting conditions with the fan on by primed quantities, we have

$$Q' = U_F A_F (T'_{AF} - T_U)$$

and

$$\bar{Q}' = U_F A_F (T'_A - T_U)$$

The reduction of $\bar{Q}$ by the attic fan is:

$$\Delta\bar{Q} = \bar{Q} - \bar{Q}' = U_F A_F (T_A - T'_A)$$

while the true heat flux reduction is:

$$\Delta Q = U_F A_F (T_{AF} - T'_{AF})$$

The data show that with the attic fan operating, the temperature difference between the mid-attic and the attic floor is reduced, i.e:

$$T_A - T_{AF} > T'_A - T'_{AF},$$

so that:

$$\Delta\bar{Q} - \Delta Q = U_F A_F (T_A - T_A' - T_{AF} + T'_{AF}) > 0$$

In other words, the actual reduction in heat flux is less than that calculated using the mid-attic air temperatures.

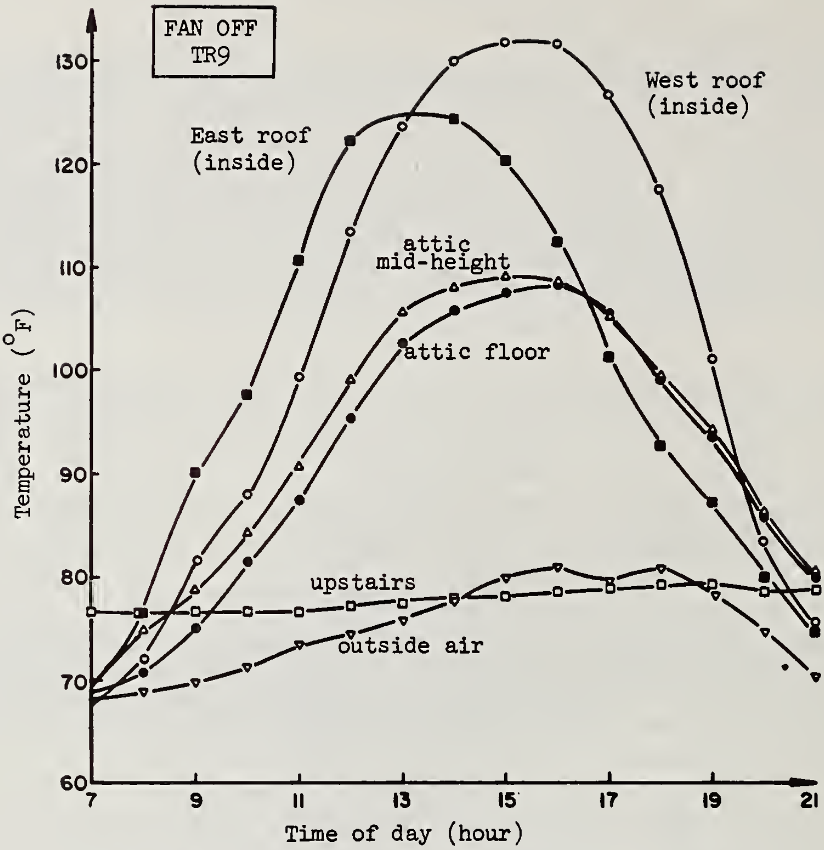


Figure 5. Distribution of temperature in the attic

References

1. M. Jacobs, "Attic Fans and Air Conditioners," Dimensions (NBS), Magazine, June 1976.

2. "Ventilation, Insulation Key Cost Cutters in Overhead Energy Loss," Professional Builder, June 1974.

3. J. M. Russ, B. G. Haynie, and F. B. Clark, "Analysis of Attic Ventilation Test," Arkansas Power and Light Company, Little Rock, Arkansas, 1976.

4. Jones and Hendrix, "Residential Energy Requirements and Opportunities for Energy Conservation," ASHRAE Transactions 82, 417, 1976.

5. C. A. Washburn, "Sacramento Attic Fan Test," Dept. of Mechanical Engineering, California State Univ., Sacramento, Jan 1978.

6. Energy and Buildings, Volume 1, No. 3. The entire issue is dedicated to the Twin Rivers research effort on residential energy consumption.

7. D. T. Harrje, "Details of the First-Round Retrofits at Twin Rivers," Energy and Buildings 1, 271

8. R. C. Sonderegger, "Movers and Stayers: The Resident's Contribution to Variation Across Houses in Energy Consumption for Space Heating," Energy and Buildings 1, 313

9. T. H. Woteki, unpublished work.

10. H. S. Hinrichs and C. K. Wolfert, "Fundamentals of Residential Attic Ventilation," H. C. Products Co., Princeville, Illinois, 1974.

11. C. I. Siu, "Attic Ventilation Study at Twin Rivers, New Jersey, During the period August 28 - September 20, 1976." Draft letter report, Building Environment Division, Center for Building Technology, National Bureau of Standards.

Acknowledgements

This work was supported by the U.S. Department of Energy under contract. We wish to acknowledge helpful discussions with and support from the Home Ventilating Institute, who supplied the calibrated attic fans for TR10, TR16, TR18, and TR27. The National Bureau of Standards permitted us to use their instrumented houses for our experiments and even provided us with their computer programs. We wish to thank Mr. Kenneth Gadsby and Mr. Roy Crosby for help with the experiments.

Questions and Answers

John A. Reagan, NASA Langley Research Center, Hampton, Va.:

How do your measurements and calculations of reduction in roof heat gain compare with ASHRAE Method predictions which ascribe an additional R-value to the attic air space based on the added cfm/ft^2 ventilation produced by the addition of attic ventilators?

Dutt and Harrje: The ASHRAE method for calculating the heat gain through attics is based on experiments conducted by Professor Joy.* The experiments were conducted under steady-state conditions and the air flow was across the roof and ceiling joists through gable vents. The Twin Rivers houses with soffit vents and a roof-mounted attic fan subject to actual weather conditions represent a different case. Nevertheless, if Professor Joy's data (Fig. 8 in his paper) are used, the following results are obtained (approximate).

R-values

	Ceiling	Roof	Attic Space with fan	Attic Space w/o fan
R-11 insulation	10	1	9	3
R-30 insulation	26	1	9	3

Notes: The flow rate with the fan on is 1.0 cfm/ft^2 of floor area, while with the fan off it is around 0.2 cfm/ft^2. The effective resistance of the attic air space can then be obtained from Fig. 8 (Professor Joy) for kraft-breather insulation and a ventilation air temperature of 85°F. These conditions are closest to our experimental case at max. $T_A - T_U$ (see Fig. 1 and 2 of our paper). The R-values of the ceiling are based on measurements.

The total R-value of the attic from living-space ceiling to the roof exterior surface (R total) increases from 14 to 20 for R-11 attics and from 30 to 36 for R-30 attics. The corresponding reductions in attic heat flux for any given sol-air temperature and room temperature are 6/14 (or 43%) and 6/30 (or 20%) for R-11 and R-30 attics. For the data shown in our Fig. 2(a) the peak heat flux reduction is about 30% with R-30 insulation, which is higher than Professor Joy's steady-state value of 20%. Despite this reduction in heat flux, no measurable A/C savings are observed, as might be expected since heat·gain through an R-30 attic represents only 5% of the A/C energy use (see Fig. 2 and discussion in our paper.)

Home Ventilating Institute (HVI): 5 questions with authors' responses.

1. The paper states that "corresponding variation of air conditioner use, attic fan use (where applicable) and attic-upstairs temperature differences were calculated from a number of townhouses, six of which had attic fans installed." Does Fig. 2 represent such calculations or actual measurements at the two specific houses?

Response: Fig. 2 (a, b, and c) represents data from actual measurements in the two houses. The top figure (a) carries two labels for the ordinate: T_A-T_U (°F) and Q_{AU}. The data shown are the temperature differences T_A-T_U. These temperature differences may be used to calculate the heat transfer through the attic floor, Q_{AU}, by multiplying by the "UA" value of the floor.

* F. A. Joy, "Improving Attic Space Insulating Values," ASHRAE Transactions, 64, 251 (1958).

2. Were comparisons made betwen Houses TR 16 and TR 18 with R-11 ceiling insulation as for the two R-30 houses in Fig. 2? If so, were there different patterns of temperatures, air conditioner use and attic fan on-time? Were differences in the two houses' thermostat settings, air conditioner efficiency and load factors considered?

Response: TR 16 and TR 18 with R-11 insulation both had attic fans installed and exhibit almost the same (T_A-T_U) pattern as shown in Fig. 2(a) for TR 27, which has R-30 insulation.

We measured but did not include the thermostat settings data in the paper. Since the air conditioner is somewhat undersized, the interior temperature often exceeds the thermostat setting. The interior temperature is therefore a more meaningful variable. Consequently, the analysis was based on measured temperatures only. The air conditioners were of the same make and rating. Sizing of the units was such that operation was continuous for a number of hours on the hottest days.

3. Were respective thermostat settings, air conditioner efficiency and load factors considered in the comparisons of the two R-30 houses in Fig. 2?

Response: The data in Fig. 2 are based on a comparison of two identical houses - one with fan on, the other with fan off, for one month's identical weather. The temperature difference is the important parameter.

4. Is the statement that no energy savings resulted from power attic venting in R-11 houses based on calculations or measurements?

Response: Experimental data were used to determine that no energy savings resulted from using attic fans.

5. Inasmuch as the paper's conclusions are based on R-30 and higher optimums of ceiling insulation for northern heating requirements, perhaps it would be appropriate to note that many authorities consider the economics of insulation quite distinct for heating and cooling. The trade-offs between insulation and ventilation can prove quite different where cooling rather than heating is the main energy user.

Response: Our study was conducted at Twin Rivers, where cooling is a significant energy user, though certainly less than heating energy, and is typical of the Northeast. Savings in the form of reduced energy entering from the attic using R-30 insulation during the summer cooling season are shown in heat flux versus time plots in the paper by Dr. Richard Grot in this workshop.

Arnold M. Kronstadt, P.E., Collins & Kronstadt, et al, Silver Spring, Md. 5 questions with authors' responses.

1. In view of the "large differences in air conditioner use" among houses in the study, with the bulk of variation ascribed to resident behavior, what measured data support a conclusion that powered attic fans saved no cooling energy? Can any valid conclusion be reached about the effect of any other factor on air conditioning when behavioral variations are not identified and quantified?

Response: The data were taken in real houses under conditions of actual use. The effect of the attic fan was, as predicted, so small that it was undetectable compared to heat transmission factors due to the other causes. The conclusion that can be supported is that measurable savings were not present using attic fans. Behavorial variations do not appear to be quantifiable to the degree that they may be factored out so that any other effect must be studied superimposed on the noise of behavioral variation of A/C use.

2. Were causes determined for the much higher air conditioner use in House TR 27 with power ventilated attic as compared with House TR 13 (Fig. 2) without such ventilation? Might it have been impossible for any attic fan to have had energy impact in this house but possible in another house?

Response: The difference in the patterns of consumption for Houses 27 and 13 shown in Fig. 2 is clearly a behavioral factor. This is why it is vital to make comparisons based on temperature differences between indoor, outdoor, and attic, and to do so over time spaces sufficiently long to cancel out any heat storage effects. Comparison using individual points from data such as shown in Figs. 3 and 4 would result in erroneous conclusions because of large variation caused by behavioral and other non-quantifiable factors in individual houses.

3. The attic fan on-time curve and hours of fan running time related to House TR 27, which had high air conditioner demand, and covered only August. Are data on this house the basis for conclusions that total energy consumption increases and peak electrical demand increases when attic fans are used? If so, would not data for more houses and a longer period be more applicable? Are data available on total attic fan running time related to air conditioner running time, covering other houses and the whole summer?

Response: The conclusion that peak electrical demand increases when an attic fan is used is based on measured attic fan usage rate, and integrated differential temperature from attic to upstairs. This differential temperature is used to calculate the heat load through the attic insulation. We do not rely on actual air conditioner use data, whose variations are determined by many other factors as well.

The late afternoon period on even a warm sunny day activates the attic fan. This is the same period for high A/C usage. Combined, this can only result in higher peak loading on the electric utility.

4. Was radiant heat from the ceiling as affecting comfort measured and evaluated, especially on days when outdoor maximum temperatures were appreciably higher than the low 80s reported in this study?

Response: Ceiling temperatures were measured and found to be reduced by about a degree with attic fan operation. The impact of change of ceiling radiation was not measured by instruments but was rather left to the discretion of the resident. The ultimate judge of comfort is the resident, who adjusts the thermostat till he/she is comfortable. If the resident raised the thermostat when the attic fan was cooling the ceiling, its effect on A/C use was not detectable. Since the A/C usage is fairly sensitive to thermostat manipulation, it is reasonable to conclude that the thermostat adjustment did not accompany attic fan operation. This fits in with our finding of very slight reduction in ceiling temperature and the fact that the affected ceiling is on the second floor while the thermostat is on the first. The temperature reported in the study is that of an average day in August, obtained from averaging a month of data. The occasional very hot day does not contribute significantly to the cooling requirement of the entire season.

5. Can the conclusions about Twin Rivers, which are based on a few two-story townhouses for a short time in cool weather, be applied without qualification to one-story or two-story detached houses in the same community in hotter seasons, or to houses elsewhere with differing roofs, orientation, and heat-humidity-wind conditions?

Response: The relative influence of attic heat load is typical of a wide variety of housing and geographical areas and we would expect these conclusions to hold. However, for single-story construction, one expects the attic to have a greater influence on summer heat loads. In such cases, larger A/C savings from attic retrofits are possible.

EFFECT OF POWERED ATTIC VENTILATION ON CEILING HEAT TRANSFER AND COOLING LOAD IN TWO TOWNHOUSES

by

Richard A. Grot and Chock I. Siu
Center for Building Technology
National Bureau of Standards
Washington, D.C. 20234

The thermal response of the attics of two townhouses at Twin Rivers, New Jersey is presented. The townhouses were extensively instrumented and the data collected were utilized to assess the effect of power venting on the ceiling heat gain and the energy consumption of the central air conditioning equipment. It is shown that for townhouse #1 with a nominal ceiling insulation level of R-30 the ceiling heat gain is only about 5% of the daily cooling load. For townhouse #2, with a nominal ceiling insulation level of R-11, the ceiling heat gain is about 10% of the daily cooling load.

The effect of a power attic ventilator on the ceiling heat gain was studied and it is shown that the attic fan reduced the maximum ceiling heat gain by about 26% for townhouse #1 and 5% in townhouse #2. The difference in percentage reduction in ceiling heat gain was probably due to the difference in the operating hours of the two attic fans caused by the different thermostat settings for the two fans, and not how much insulation was present. The effect of powered attic ventilation on daily or peak cooling load of the air conditioner was too small to be measurable because of the large variations in solar load and interior general lighting load.

Key words: Air conditioning load; attic ventilation effect on cooling load; ceiling heat gain; power ventilator evaluated; thermal response; Twin Rivers, N.J.; ventilation studies.

Introduction

The national trend is towards increased use of central air conditioning to cool living spaces of houses. In 1950 only 1% of all households had central air conditioners. By 1970 this figure increased to 11%, amounting to 7.6×10^6 households consuming 26×10^9 kWh of electricity annually [1]. The operation of central air conditioners contributes heavily to peak electric loads. Today, an important concern in cooling living spaces is to attain reasonable comfort levels with a minimal use of energy. Living spaces can be cooled, under certain conditions, by means of air conditioners, or by natural ventilation and forced ventilation. The latter involves whole-house fans, power attic fans and ventilators. Attic fans, ridge vents, and turbine ventilators are being used during the cooling season to increase the amount of attic ventilation, in an attempt to reduce the attic temperature and hence the heat gain through the ceiling. In homes which have central air conditioners, these ventilators are used in an attempt to decrease the electrical consumption of air conditioning equipment through a reduction in the ceiling heat gain. However, it is controversial whether the use of these ventilators is an effective energy conservation procedure [2,3].

In late August 1976, the National Bureau of Standards, jointly with Princeton University, initiated at Twin Rivers, N.J. an attic ventilation study in two townhouses equipped with central air conditioners and powered attic ventilators. The purpose of this venture was to obtain data which would: 1) provide for better understanding of the heat transfer processes within attics, 2) determine the merits of added ceiling insulation in reducing air conditioner energy consumption, and 3) determine the effectiveness of attic fans in reducing compressor operating hours at different levels of attic floor insulation.

Two townhouses used in previous studies [4] were re-instrumented with additional sensors which provided a detailed description of the thermal performance of the townhouse attics. Data were collected during the months of July and August of 1977. This paper describes the test homes, the instrumentation used and the results of the data analysis.

Description of Test Houses, Instrumentation, and Data Analysis Method

A three-bedroom and a four-bedroom townhouse, each with two stories, an attic and a basement, were selected for this study. The four-bedroom townhouse contained 3 1/2-inch fibrous batt insulation in the attic (nominal R-11) while the three-bedroom townhouse contained 9 inches of fibrous glass wool blown-in insulation installed on top of 3 1/2-inch fibrous glass batt insulation, for a total nominal resistance of R-30. These houses adjoined each other in a row of townhouses, separated by cinder block masonry party walls. The houses were of frame construction and had 2-inch fibrous insulation (nominal R-7) in the exterior walls. They had the same orientation, with the ridge of the roof lying 7° east of true north. The three-bedroom townhouse (referred to as townhouse #1) had a 2-ton central air conditioner. The four-bedroom townhouse referred to as townhouse #2 had a 2 1/2-ton central air conditioner. Each townhouse had a complete set of major electric appliances: refrigerator, dishwasher, range-oven combination in the kitchen; electric water heater, and clothes washer and dryer in the basement. Each townhouse was occupied by two adults and two children (ages 11 and 14).

The attic floor area of townhouse #1 was 697 ft^2 and that of townhouse #2 was 760 ft^2. The two attics have a ridge height of approximately 7 ft above the attic floor, resulting in attic volumes of 2469 ft^3 and 2690 ft^3, respectively. Using a tracer gas, the natural ventilation rates for the attics were measured during August 1976 to be from 1.7 to 2.3 air changes per hour. The forced ventilation rate with the power ventilator on was measured to be 12 air changes per hour in each attic. Measurements indicated that the attic fan in townhouse #1 turned on when the attic temperature reached approximately 91°F, while the attic fan for townhouse #2 turned on at a temperature of approximately 98°F.

The townhouses were fully instrumented and data were recorded by means of a data acquisition system located in the basement of townhouse #1. The data acquisition system had the capability of recording 100 channels of data in a scan time of six seconds and storing this data on a seven-track incremental magnetic tape recorder. Electric meters were installed on the air conditioner, refrigerator, water heater, range, dishwasher, and the general lighting circuit. These electric meters were modified such that they produced an electric pulse for each 1.8 watt-hour of electricity consumed. These pulses were totalized by specially designed electronic counters which produced a voltage directly proportional to the total number of pulses received. Copper-constantan thermocouples were installed in the attic, in the interior of the townhouses and in the mechanical system (see Table 1). Heat flux meters were installed under the insulation on the ceiling of the second floor and also on the attic roof surface.

Two long-wavelength radiometers were installed in one townhouse to measure the radiation exchange between the underside of the roof and the attic floor. Relays were installed on the powered attic fan and on the air conditioner fan in such a way that whenever the devices were turned on or off, the data acquisition system would record the event. Table 1 lists the data collected for each townhouse.

The data acquisition system scanned the townhouses every 5 minutes. Once a week the data tapes were collected and shipped to the National Bureau of Standards by Princeton University researchers. These tapes were then analyzed on the minicomputer of the Center for Building Technology. The data analysis programs produced hourly and daily data summaries of all measured quantities and these were printed in tabular form and stored on disc files in the computer. These data were further reduced to the 23 quantities listed in Table 2 by averaging attic heat fluxes and attic temperatures at the same level to produce one quantity. Data files were established which consisted of the hourly values of these variables, the daily averages and totals, the daily maximums and the daily minimums. On a daily basis the ceiling heat flux and corresponding differential temperatures across the insulation were separated into positive values (heat gain) and negative values (heat loss) and each quantity was totaled separately to give the total heat gain and the total heat loss across the insulation

TABLE 1. DATA COLLECTED FOR EACH TOWNHOUSE

Energy Consumption:

1. Air Conditioner
2. Water Heater
3. Dishwasher
4. Refrigerator
5. Range/oven
6. General Lighting Circuit

Temperature:

Interior of Dwelling

1. 1st Floor
2. 2nd Floor
3. Kitchen

Attic

1. Inner Surface of Roof
2. Air Temperature near Peak of Roof
3. At Attic Fan Exhaust
4. At Attic Fan Thermostat
5. 1 ft. Above Ceiling (3-4 locations)
6. 3 ft. Above Ceiling (3-4 locations)
7. At Ceiling Surface (4 locations)
8. Attic Inlet Vents (East and West)
9. Under Attic Insulation (4 locations)
10. Inside Surface of Ceiling

Mechanical Systems

1. Return Duct
2. Main Supply Ducts (3 locations)

Heat Flux:

1. Inner Surface of the Roof
2. Under Attic Insulation (4 locations)

On-Time:

1. Attic Fan
2. Air Conditioner Fan

Exterior Environment:

1. Air Temperature
2. Solar Radiation on Horizontal Surface
3. Wind Velocity and Direction

TABLE 2. LIST OF PARAMETERS ANALYZED

	Parameter	Units
1.	Electricity Air Conditioners	kWh
2.	Electricity Water heater	kWh
3.	Electricity Refrigerator	kWh
4.	Electricity Range-Oven	kWh
5.	Electricity Dishwasher	kWh
6.	Electricity General Lighting	kWh
7.	Sensible Cooling Output of Air Conditioners	kWh
8.	Ceiling Heat Flux	$Btu/(ft^2 \cdot hr)$
9.	Ceiling Heat Gain	kWh
10.	Temp. Differential Ceiling	°F
11.	Attic Surface Temperature (floor)	°F
12.	Attic Temperature at 1-foot Level	°F
13.	Attic Temperature at 3-foot Level	°F
14.	Roof Temperature	°F
15.	East Roof Heat Flux	$Btu/(ft^2 \cdot hr)$
16.	West Roof Heat Flux	$Btu/(ft^2 \cdot hr)$
17.	Interior Temperature 1st Floor	°F
18.	Interior Temperature 2nd Floor	°F
19.	Solar Radiation	$Btu/ft^2 \cdot hr)$
20.	Outside Air Temperature	°F
21.	Difference Outside-Interior Temperature	°F
22.	Time Attic Fan On	hours
23.	Time Air Conditioner Fan On	hours

layer for the day, and the corresponding integrals of the differential temperatures. This last procedure was followed because at Twin Rivers during most of the summer the diurnal temperature variation is such that the house experiences heat gains in the day and heat losses in the evening due to heat transfer from the attic.

Discussion of Test Results

Data were collected on the performance of the attics and the central air conditioners during the months of July and August 1977. During a period of approximately 14 days in August, the attic fans were turned off. A summary of the average conditions during these test periods is given in Table 3. In Table 3 and in the following graphics, "fan on" refers to the periods during which the fan operated for some time during the day under thermostat control; "fan off" refers to the test period when the fan was turned off. In Table 4 the average daily hourly maximums for each test period are listed. It should be noted that the test period during which the attic fans were off was milder than the test period during which the fans were on. The average daily differential temperature across the building envelope was -1.6°F for townhouse #1 and -5.7°F for townhouse #2 during the test period when the attic fan was on, and -7.3°F for townhouse #1 and -9.3°F for townhouse #2 during the test period when the attic fan was off. The observed differences in electricity consumption for air-conditioning -- 27.1 kWh per day when the fan was on and 9.1 kWh per day when the fan was off for townhouse #1 and 38.5 kWh per day when the fan was on and 20.9 kWh per day when the fan was off for townhouse #2 -- can be mostly explained by the differences in weather conditions.

The important point to be made by examining the data presented in Tables 3 and 4 is that the total daily ceiling heat gain in both test periods is a small fraction of the daily sensible cooling output of the air conditioner. For townhouse #1 the ceiling heat gain averaged only 4.7% of the sensible cooling output of the air conditioner during the test period when the attic fan was on; during the period when the attic fan was off it averaged 7.6% of the sensible cooling output of the air conditioner. For townhouse #2 these percentages are 9.1% and 7.1%, respectively. This same conclusion is obtained by studying Figure 1, in which the daily electric consumption of the air conditioner is plotted versus the total ceiling heat gain. In Figure 2, the daily maximum hourly electrical consumption for air conditioning is plotted versus daily maximum hourly heat gain, and the same conclusion is indicated -- that even at maximum operating conditions the ceiling heat gain is a small fraction of the cooling load. At daily maximum operating conditions the ceiling heat gain of townhouse #1 averaged 6.9% of the maximum sensible cooling output of the air conditioner when the attic fan was on and 7.7% when the attic fan was off. For townhouse #2 these percentages are 7.8% and 9.3%, respectively. The flattening of the plots of Figure 2 for higher attic heat gains is due to the undersizing of the central air conditioner in both townhouses -- a common practice. Thus, under extreme weather conditions, a sizable reduction in ceiling heat gains would still have a negligible effect on the energy used for air conditioning.

TABLE 3. DAILY AVERAGES AND STANDARD DEVIATIONS FOR TEST PERIODS

	Townhouse #1				Townhouse #2			
	Fan On		Fan Off		Fan On		Fan Off	
	Mean	Std. Deviation	Mean	Std. Deviation	Mean	Std. Deviation	Mean	Std. Deviation
Electric Energy Use, A/C (kWh)	27.1	18.1	9.1	8.9	38.5	21.8	20.9	16.7
Diff. House Temp (°F)	-1.6	6.1	-7.3	4.5	-5.7	5.6	-9.3	3.2
Electric Energy Use, General Lighting (kWh)	57.3	8.7	47.8	8.9	50.7	10.1	44.3	8.9
Cooling Output, A/C (kWh)	30.3	19.5	10.1	10.2	41.7	23.6	24.0	19.0
Total Ceiling Heat Gain (kWh)	1.43	0.67	0.77	0.61	3.80	0.75	1.69	1.25
Integral Diff. Ceiling Temp (°F-hr)	136.5	58.9	74.5	63.2	173.1	33.4	79.7	56.4
Fan On-Time (hr)	12.0	3.1	-	-	5.8	1.9	-	-
Total Solar Radiation (Btu/ft^2)	1700.4	435.6	1049.8	591.1	1814.4	318.0	1003.4	530.6
1st Floor Air Temp (°F)	77.1	2.0	77.3	1.7	80.8	0.9	80.7	1.7
2nd Floor Air Temp (°F)	79.0	1.5	78.6	1.5	82.3	1.6	82.6	2.3
Temp at Upper Surface of Insulation(°F)	80.9	3.1	77.5	2.9	81.8	3.4	77.2	2.8
Attic Temp. at 1-ft Level (°F)	81.5	3.7	77.5	3.0	81.1	3.4	76.7	2.9
Attic Temp. at 3-ft Level (°F)	81.8	3.8	77.8	3.1	83.1	3.2	78.0	3.0
Roof Surface Temp (°F)	85.5	4.4	78.8	4.0	82.5	3.0	77.8	2.8

TABLE 4. AVERAGES AND STANDARD DEVIATIONS OF DAILY HOURLY MAXIMUMS FOR TEST PERIODS

Maximum Values	Townhouse #1				Townhouse #2			
	Fan On		Fan Off		Fan On		Fan Off	
	Mean	Std. Deviation	Mean	Std. Deviation	Mean	Std. Deviation	Mean	Std. Deviation
Electric Energy Use, A/C (kWh)	2.33	1.29	1.38	1.08	3.45	1.58	2.37	1.69
Diff. House Temp (kWh)	7.9	7.1	-0.5	3.23	3.4	5.4	-3.1	4.0
Electric Energy Use, General Lighting (kWh)	3.82	0.68	2.93	0.73	4.39	0.93	3.92	1.10
Ceiling Heat Gain (kWh)	0.18	0.06	0.12	0.09	0.44	0.06	0.24	0.15
Temp Diff. Ceiling (°F)	17.7	5.6	12.3	9.0	20.6	2.95	11.9	7.0
Solar Radiation (Btu/hr·ft^2)	219.1	39.9	147.5	73.9	230.0	25.3	144.6	62.2
1st Floor Temp (°F)	79.7	2.8	79.6	2.6	85.3	1.8	84.3	2.1
2nd Floor Temp (°F)	81.4	2.0	80.5	2.3	85.6	1.8	85.4	2.5
Attic Temp at Ceiling Surface (°F)	95.0	4.5	90.2	9.5	96.9	3.6	88.0	7.3
Attic Temp at 1-ft Level (°F)	97.5	5.1	91.4	10.4	95.9	3.1	87.9	7.5
Attic Temp at 3-ft Level (°F)	97.9	5.4	91.7	10.1	98.6	3.0	90.6	8.8
Roof Surface Temp (°F)	112.7	7.9	100.7	15.2	97.1	2.8	89.0	7.9

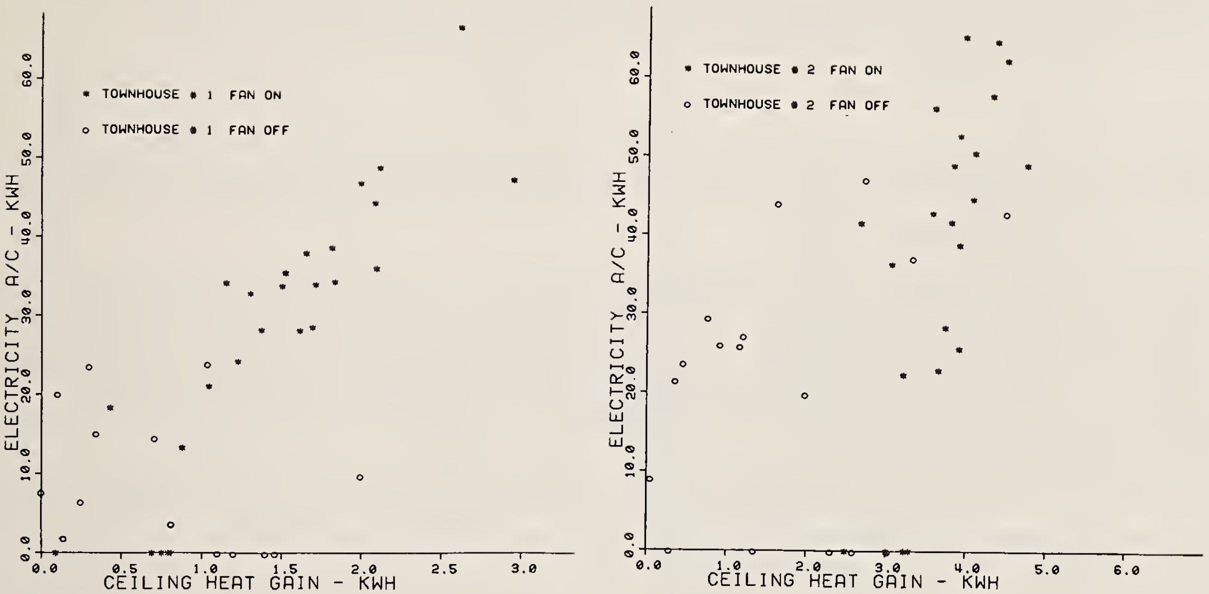


Figure 1. Plot of Daily Electricity for Air Conditioning Vs. Daily Ceiling Heat Gain

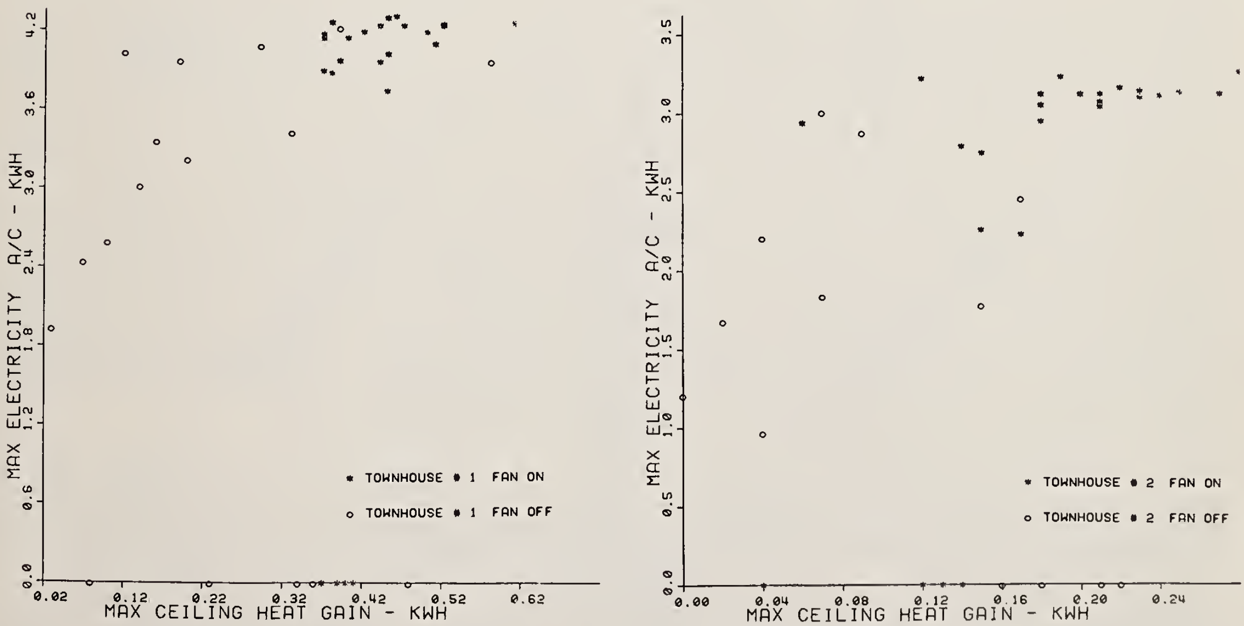


Figure 2. Plot of Daily Maximum-Hour Air Conditioner Electricity Vs. Daily Maximum-Hour Heat Gain.

In Figure 3, the daily ceiling heat gain is plotted versus the integral of the positive differential temperature across the ceiling insulation. Figure 4 shows similar plots for the maximum hourly differential temperature across the ceiling insulation. Figures 3 and 4 show that the relationship between the ceiling heat gain and differential temperature across the ceiling insulation is essentially between the ceiling heat gain and differential temperature across the ceiling insulation is essentially linear both for daily totals and daily maximum conditions. If a best-statistical-fit of the equation

$$Y = a_o + a_1 X \tag{1}$$

is made to the data presented in Figure 3 and 4, where a_o is the intercept and a_1 is the slope, then the effective R-value of the ceiling insulation can be obtained from the relation

$$\text{R-value} = A/(a_1 \times 3413) \tag{2}$$

where A is the ceiling area. The coefficients a_o and a_1 of equation 1 for several data sets are given in Table 5. The straight line fits to the data sets are shown in Figures 3 and 4, a solid line indicating the straight line for the period during which the fan was on, the dashed line the period during which the fan was off. Using equation 2 and the coefficients in Table 5, the ceiling insulation in townhouse #1 has an R-value between 18.3 and 21.7 (about 30% below the nominal value of R-30)* and the insulation in townhouse #2 has an R-value between 9.6 and 11.0.

In order to assess the influence of the power attic fan on the maximum ceiling heat gain, the simple attic model depicted in Figure 5 was statistically fitted to the data

$$H = b_o + b_1 * S + b_2 DT \tag{3}$$

where H is the ceiling heat gain, S is the solar radiation on a horizontal surface and DT is the temperature difference across the envelope.

Equation 3 was fitted using least-squares regression analysis both to the daily total ceiling heat gain data (H) using the average daily solar radiation on a horizontal surface (S) and the average daily temperature difference across the exterior envelope (DT) as independent variables and also to the daily maximum ceiling heat gain data (H_{max}) using the daily maximum hourly solar radiation on a horizontal surface (S_{max}) and the daily maximum hourly temperature difference across the exterior envelope (DT_{max}). Table 6 gives the coefficients b_o, b_1 and b_2 for each test condition, the correlation coefficient R^2 and the standard deviation, S.D., of the fitted model for the daily total ceiling heat gain. The effect of each variable S and DT on the total ceiling heat gain H can be ascertained by studying Figures 6 and 7. In Table 6, $\overline{H}$, $\overline{S}$ and $\overline{DT}$ are the averages of the respective variables. The low correlation coefficient R^2 for townhouse #2 when the fan was on is probably due to the small variation in the data (see standard deviations in Table 6) during this test. If the results in Table 6 are used to predict what the ceiling heat gain would have been under the weather conditions when the power attic fan was on, if the power attic fan had been left off, a ceiling heat gain of 1.97 kWh would have occurred under average conditions for townhouse #1 and 4.02 kWh for townhouse #2. This represents a reduction of approximately 27% (0.54 kWh) for townhouse #1 and approximately 5% (0.22 kWh) for townhouse #2 in the total ceiling heat gain attributable to the operation of the attic fan.

Table 7 gives the coefficients b_o, b_1 and b_2 for each test condition and the correlation coefficient R^2 and the standard deviation, S.D., of the fitted model at peak loads. The effect of each variable S_{max} and DT_{max} on the maximum heat gain H_{max} can be ascertained by studying Figures 8 and 9. In Table 7, $\overline{H}_{max}$, $\overline{S}_{max}$, and $\overline{DT}_{max}$ are the averages of the respective daily maximum. The low correlation coefficient R^2 for townhouse #2 when the fan was on is a statistical quirk due to the small variation in the data (see standard deviations in Table 7 during this test (note that the standard deviation, S.D., from the

---

* The discrepancy is not yet explained. Similar calculation during the 1970-1971 heating season gave R-values (≈26) for townhouse #1 within 15% of the nominal value.

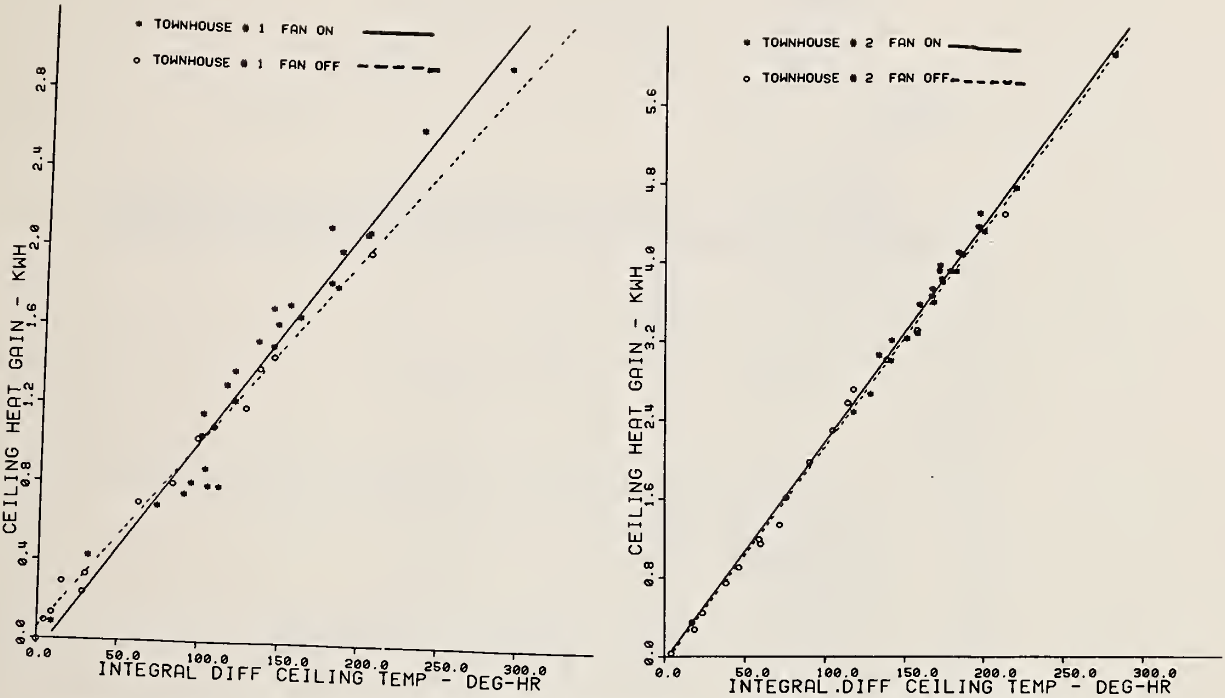


Figure 3. Plot of Daily Ceiling Heat Gain Vs. Integral of Differential Ceiling Temperature

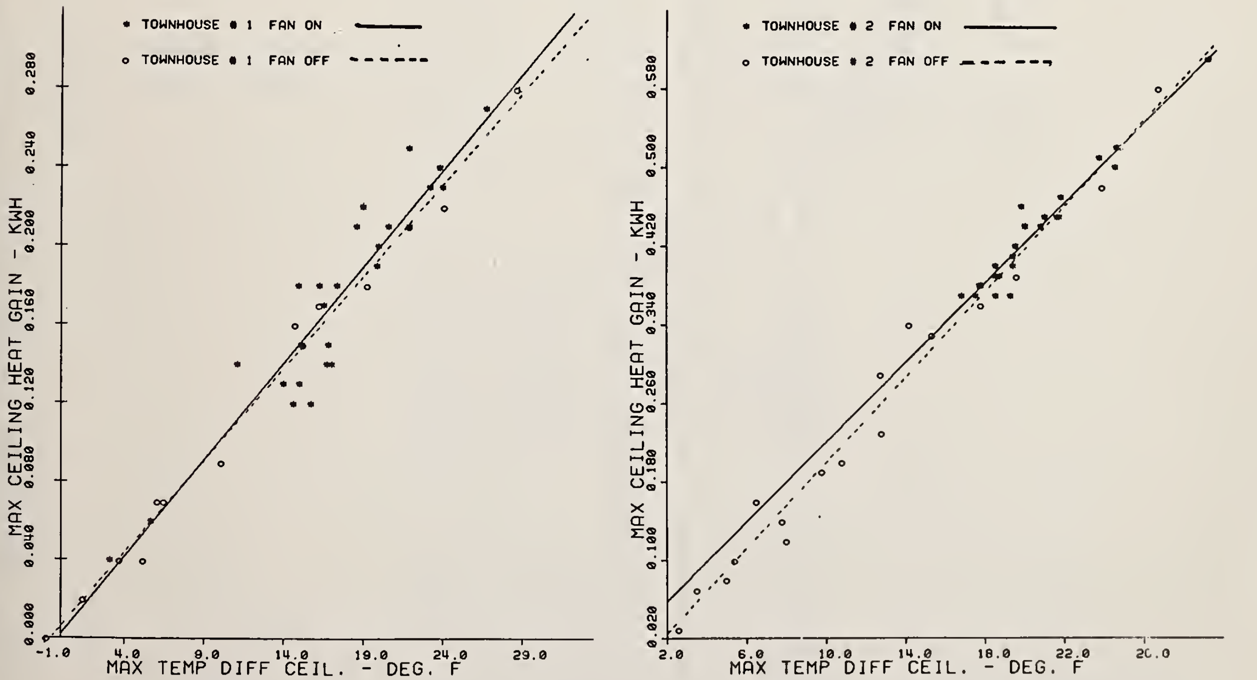
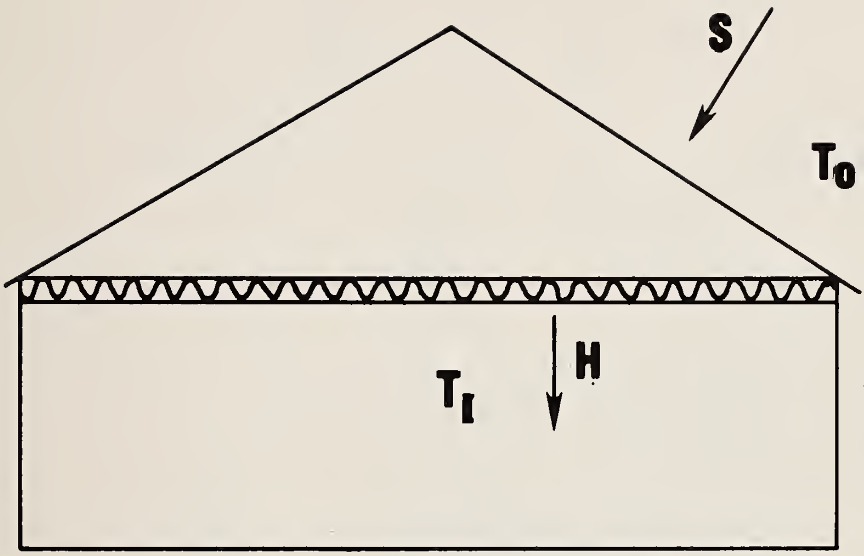


Figure 4. Plot of Daily Maximum Ceiling Heat Gain Vs. Daily Maximum Differential Ceiling Temperature

TABLE 5. SUMMARY OF COEFFICIENTS OF STRAIGHT-LINE FITS TO DATA SETS

		Townhouse #1						Townhouse #2					
		Fan On			Fan Off			Fan On			Fan Off		
Y-Variable	X-Variable	a_0	a_1	R^2	a_0	a_1	R^2	a_0	a_1	R^2	a_0	a_1	R^2
Ceiling Heat Gain (kWh)	Integral Diff. Ceil. Temp (F-Hr)	-0.778	0.0111	0.95	0.056	0.0097	0.99	-0.039	0.0222	0.98	-0.070	0.0221	0.99
Max. Ceiling Heat Gain (kWh)	Max. Temp Diff. Ceil. (F)	0.002	0.0099	0.89	0.005	0.0094	0.99	0.018	0.0203	0.94	-0.019	0.0217	0.98
Electric Energy Use A/C (kWh)	House Diff. Temp (F)	31.4	2.64	0.80	18.9	1.35	0.47	57.3	3.31	0.71	61.6	4.4	0.71
Electric Energy Use A/C (kWh)	Electricity General Lighting (kWh)	-62.8	1.57	0.58	-25.3	0.72	0.51	-40.2	1.55	0.52	18.1	0.41	0.27
Cooling Output (kWh)	Electricity A/C (kWh)	1.26	1.07	0.98	-0.24	1.14	0.99	0.56	1.07	0.98	0.44	1.13	0.99



$$DT = T_0 - T_I$$

$$H = b_0 + b_1 \cdot S + b_2 \cdot DT$$

Figure 5. Simplified Attic Model for Normalizing Attic Heat Gain Data.

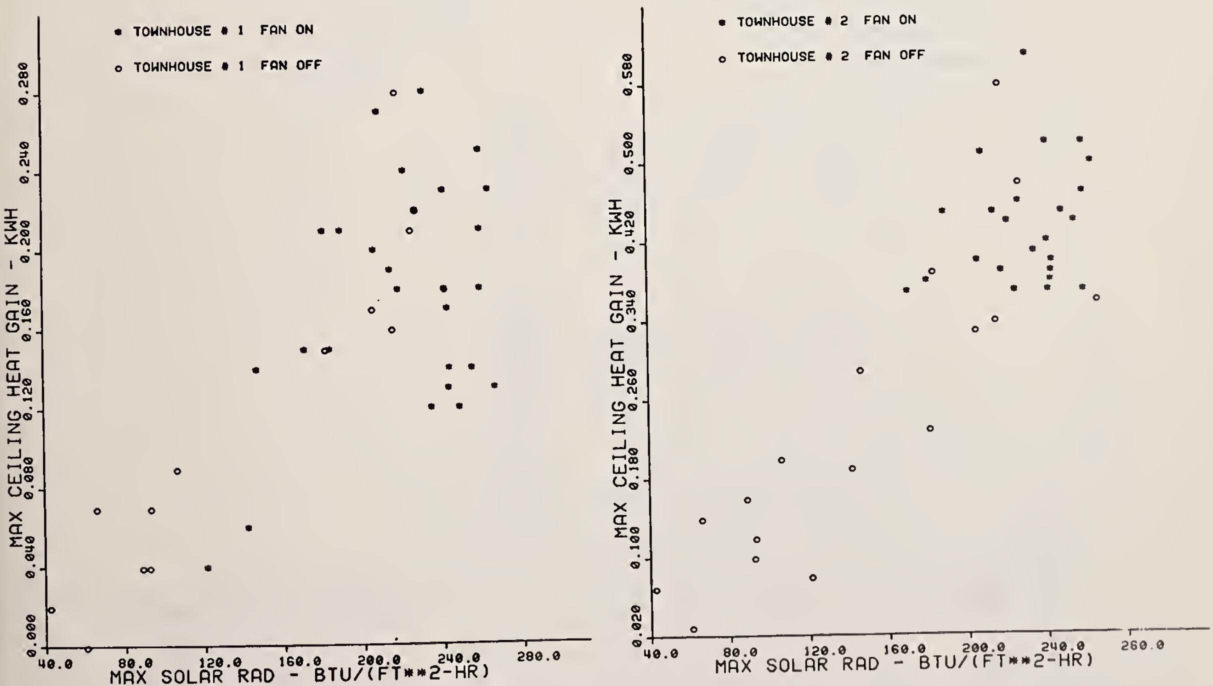


Figure 6. Plot of Daily Maximum Ceiling Heat Gain Vs. Daily Maximum Solar Radiation on a Horizontal Surface.

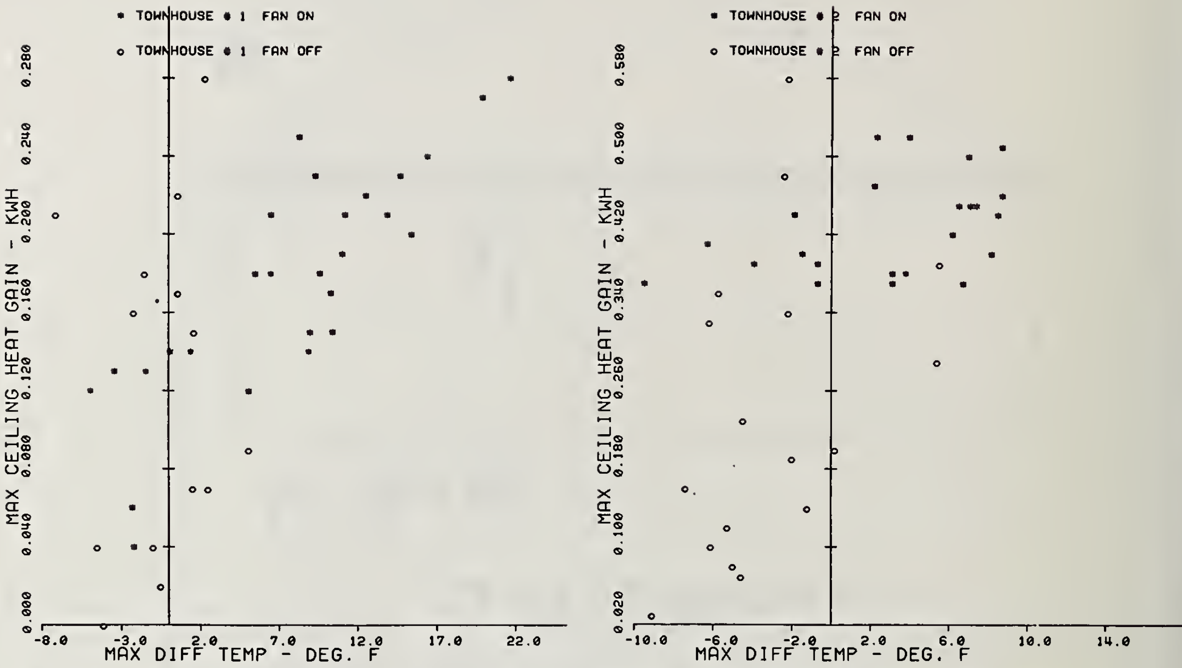


Figure 7. Plot of Daily Maximum Ceiling Heat Gain Vs. Daily Maximum Differential Temperature across the Building Envelope.

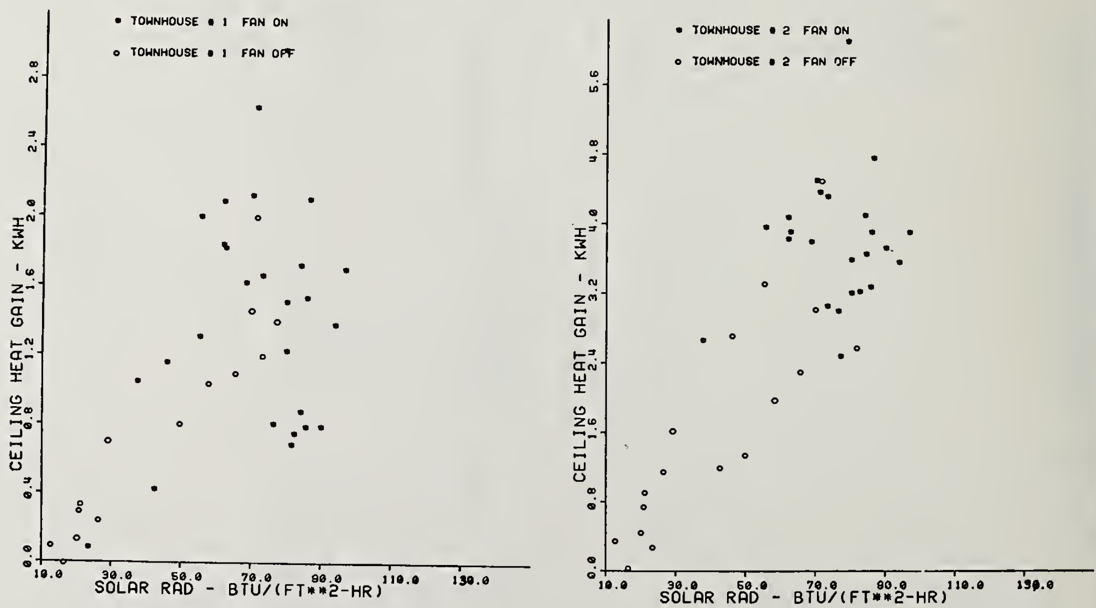


Figure 8. Plot of Daily Ceiling Heat Gain Vs. Daily Average Solar Radiation on a Horizontal Surface.

TABLE 6. COEFFICIENTS OF ATTIC MODEL FOR DAILY TOTALS

$$H = b_0 + b_1 S + b_2\ DT$$

	b_0	b_1	b_2	Standard Deviation of Predicted $\overline{H}$	R^2	$\overline{H}$	$\overline{S}$	$\overline{DT}$	Predicted $\overline{H}$ If Fan Off
#1 Fan On	0.57 ($\pm$0.19)*	0.014 ($\pm$0.003)*	0.101 ($\pm$0.008)*	0.24	0.88	1.43 ($\pm$0.67)+	70.9 ($\pm$18.1)+	-1.6 ($\pm$ 6.1)+	1.97
#1 Fan Off	-0.17 ($\pm$ .08)*	0.032 ($\pm$0.002)*	0.061 ($\pm$0.013)*	0.13	0.95	0.78 ($\pm$0.61)*	43.7 ($\pm$24.6)*	-7.3	----
#2 Fan On	2.67 ($\pm$0.37)*	0.024 ($\pm$0.008)*	0.115 ($\pm$0.018)*	0.45	0.66	3.80 ($\pm$0.75)+	75.6 ($\pm$13.3)+	-5.7 ($\pm$5.5)+	4.02
#2 Fan Off	0.90 ($\pm$0.37)*	0.054 ($\pm$0.005)*	0.163 ($\pm$0.037)*	0.46	0.88	1.67 ($\pm$1.25)+	41.8 ($\pm$22.1)+	-9.3 ($\pm$3.2)+	----

* Standard errors

+ Standard deviations

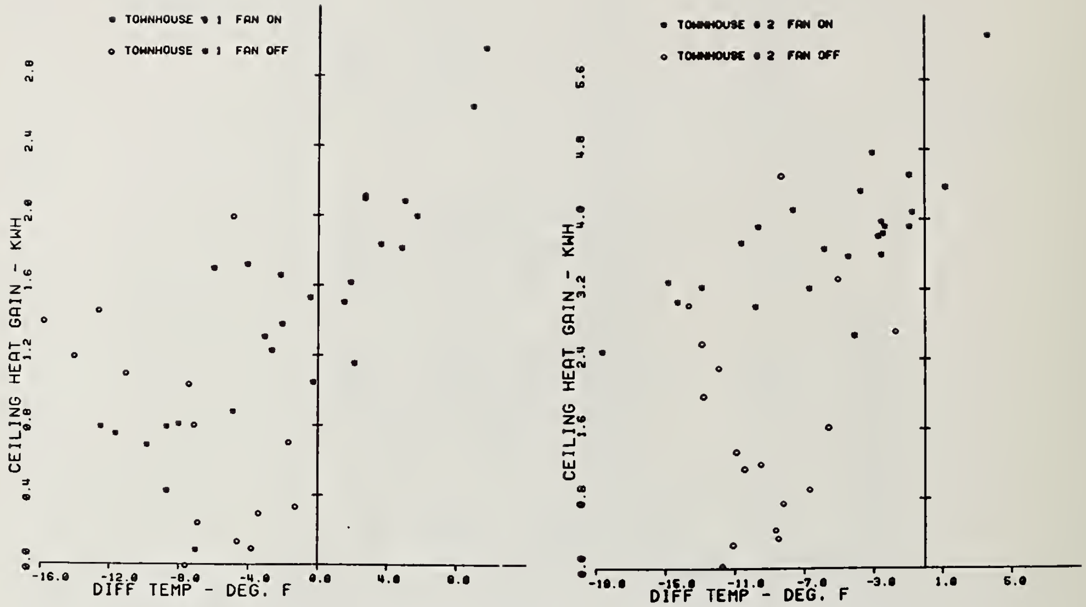


Figure 9. Plot of Daily Ceiling Heat Gain Vs. Daily Average Differential Temperature across the Building Envelope.

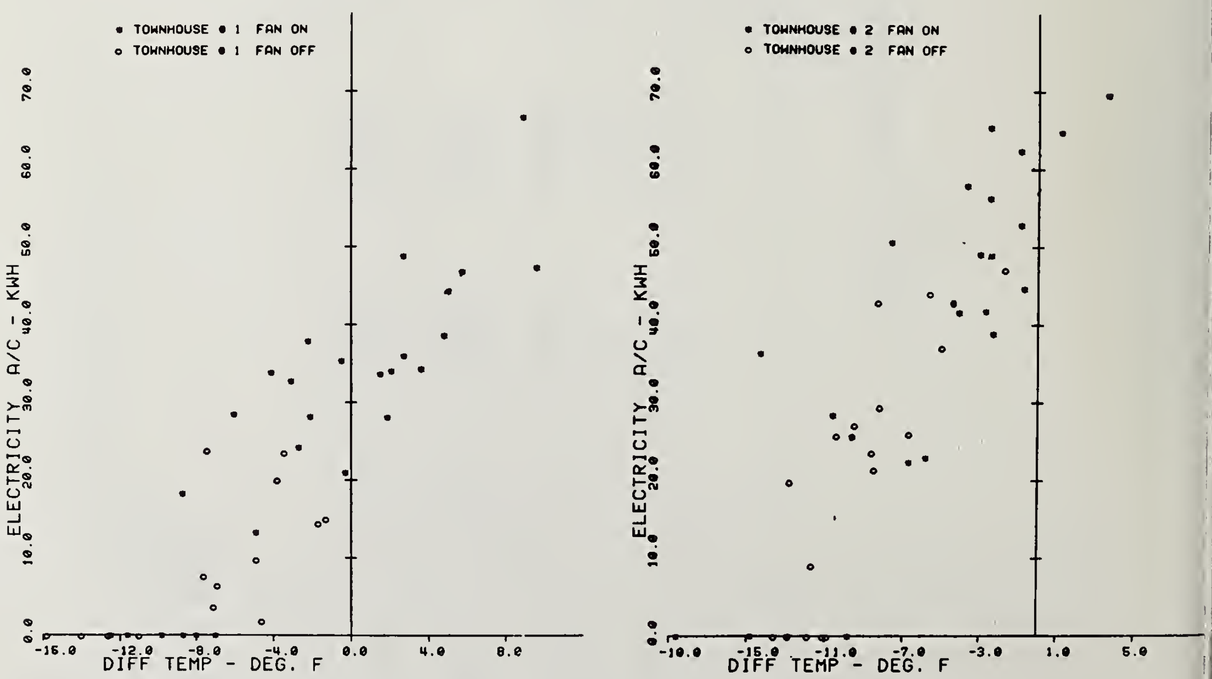


Figure 10. Plot of Daily Air Conditioning Electricity Vs. Daily Average Differential Temperature across the Building Envelope.

fitted equation is small). If the results in Table 7 are used to predict what the ceiling heat gain would have been under the weather conditions when the power attic fan was on, if the power attic fan had been left off, a reduction of approximately 26% (0.06 kW) in ceiling heat gain would have occurred in townhouse #1 and a 5% (0.02 kW) reduction in townhouse #2 due to the operation of the fan.

In principle, the operation of the attic fan should have produced the same percentage reduction in the ceiling heat gain in townhouse #1 and #2. It is supposed that the ineffectiveness of the attic fan in reducing the ceiling heat gain in townhouse #2 is due to the thermostatic setting of the attic fan control which led to that fan being on only 5.8 hours per day as compared to 12.0 hours per day in townhouse #1. However, as our previous discussion indicated, the ceiling heat gain represented 10% at most of the sensible output of the air conditioner and even a 25% reduction in the ceiling heat gain would lead to a reduction in air conditioner operation of less than 3%.

For completeness, a plot of the daily air conditioner energy usage versus daily-average temperature differential across the dwelling is presented in Figure 10. Figure 11 gives a plot of the daily energy consumption of the air conditioner versus daily energy usage for general lighting and 120V appliances in order to provide an indication of the internal heat load to the dwelling. The coefficients of equation (1) for these data sets are listed in Table 5, along with those of the sensible cooling output of the air conditioner versus air conditioner energy (Figure 12).

Since the reductions in cooling load on the air conditioners resulting from attic fan operation were shown to be less than 3% of the total cooling load, it is unlikely that such a small variation could be detected in the daily energy use of the air conditioner, especially when the day-to-day variations in general lighting load and solar radiation are as large as are shown in Figure 8 and Figure 11. Direct measurement of the heat transfer through the ceiling by means of heat flow meters is a much more effective way to evaluate the effect of attic fan operation on cooling load.

As the above analysis of the ceiling heat gain data indicated, no reduction in air conditioner energy usage can be expected which can be attributed to the operation of the power attic ventilator.

Conclusions

The results of these tests have shown that for ceiling insulation levels of R-11 and R-30, the ceiling heat gain for a two-story townhouse is only a small portion (less than 10%) of the sensible cooling load of the air conditioner for a climate similar to that of central New Jersey. Though the attic fan can reduce the ceiling heat gain by as much as 25%, no difference in the operation of the air conditioner could be observed either under average conditions or at maximum conditions.

References

1. Transil, J., "Residential Consumption of Electricity 1950-1970." Oak Ridge National Laboratory Report ORNL-NSF-EP-51, July 1973.

2. "Ventilation, Insulation Key Cost Cutters in Overheat Energy Loss," Professional Builder, June 1974.

3. Wright, J. R., Bahnfleth, D. R., Brown, E. J., "Comparative Performance of Year-Around Systems Used in Air Conditioning Research Residence No. 2." University of Illinois Engineering Experimental Station Bulletin No. 465.

4. Harrje, D.T. and Grot, R.A., "Instrumentation for Monitoring Energy Usage in Residential Buildings at Twin Rivers" Energy and Buildings, Volume 1, No. 3, April 1978.

TABLE 7. COEFFICIENTS OF ATTIC MODEL FOR DAILY MAXIMUM CONDITIONS

$$H_{max} = b_0 + b_1 \cdot S_{max} + b_2 \cdot DT_{max}$$

	b_o	b_1	b_2	Standard Deviation of Predicted H_{max}	R^2	$\overline{H}_{max}$	$\overline{S}_{max}$	$\overline{DT}_{max}$	Predicted $\overline{H}_{max}$ If Fan Off
# 1 Fan On	-0.006 (+/-0.026)	0.00059 (+/-0.00011)	0.0068 (+/-0.0006)	0.023	0.86	0.177 (+/-0.059)	219. (+/-39.)	7.9 (+/-7.1)	0.240
# 1 Fan Off	-0.037 (+/-0.019)	0.00109 (+/-0.00012)	0.0049 (+/-0.0027)	0.031	0.89	0.121 (+/-0.086)	147. (+/-74.)	-0.5 (+/-3.2)	----
# 2 Fan On	0.201 (+/-0.097)	0.00091 (+/-0.00041)	0.0072 (+/-0.0019)	0.049	0.43	0.436 (+/-0.062)	230. (+/-25.)	3.4 (+/-5.4)	0.459
# 2 Fan Off	0.002 (+/-0.050)	0.00186 (+/-0.00028)	0.0096 (+/-0.0048)	0.074	0.80	0.241 (+/-0.155)	144. (+/-67.)	-3.1 (+/-4.0)	----

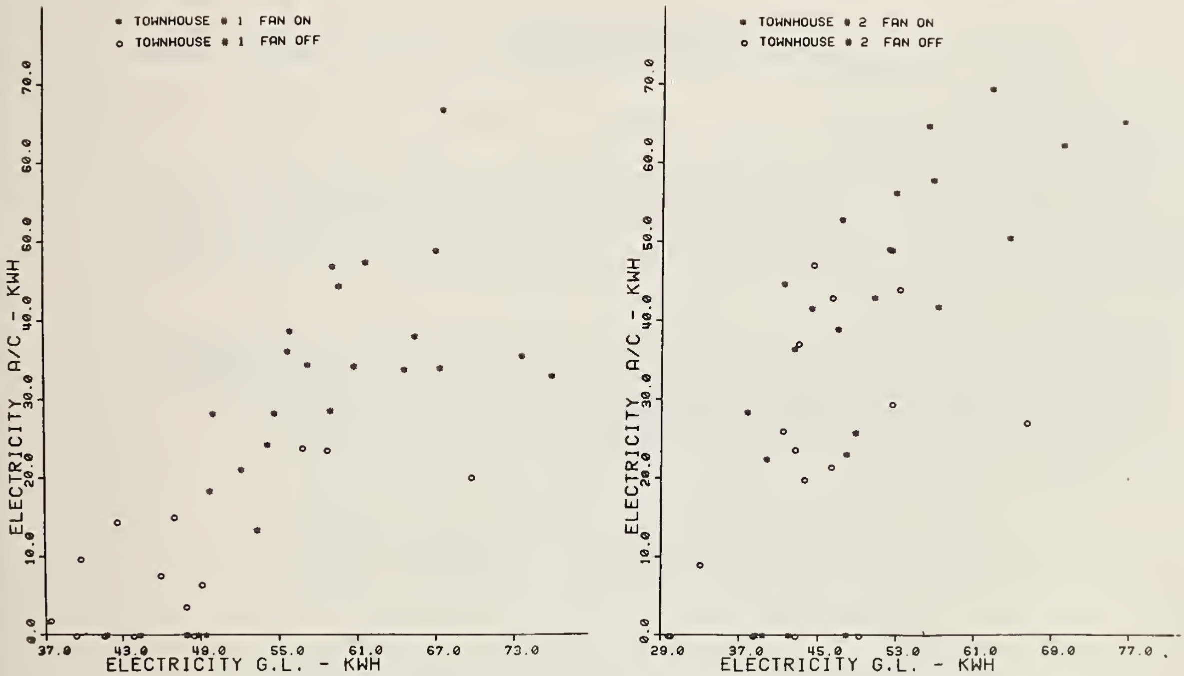


Figure 11. Plot of Daily Air Conditioning Energy Vs. Daily Electricity for General Lighting Circuit.

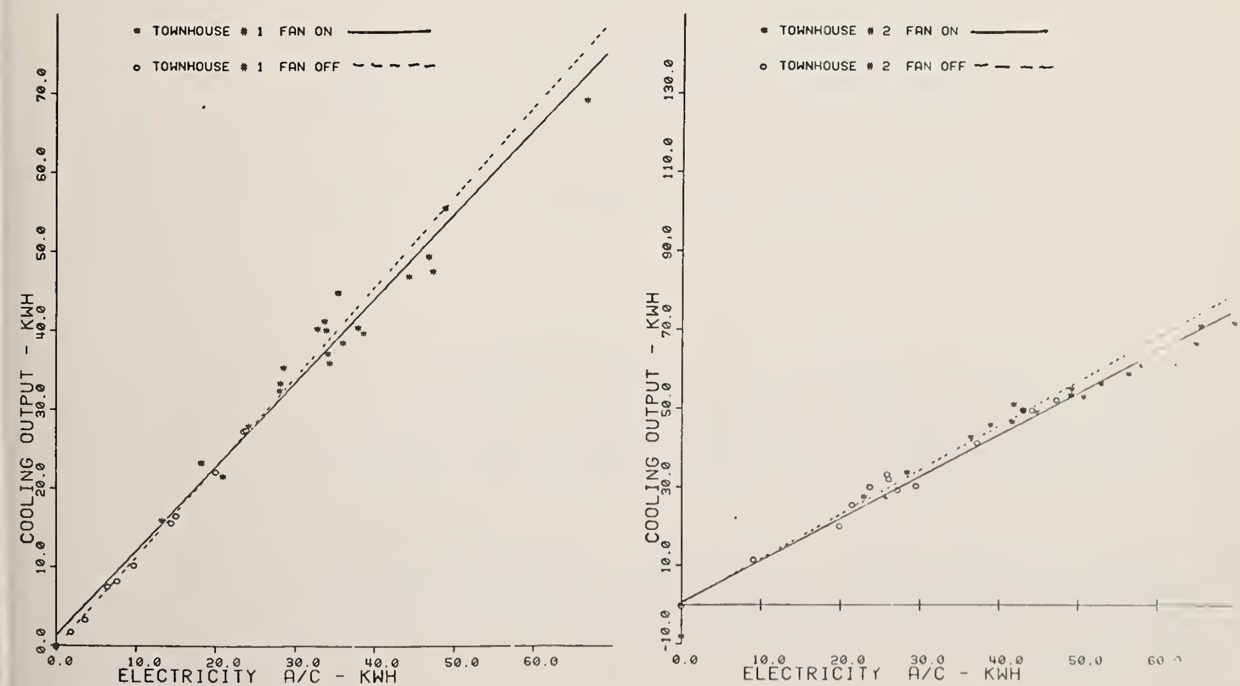


Figure 12. Plot of Daily Air Conditioner Sensible Output Vs. Daily Air Conditioner Energy.

Questions and Answers

Home Ventilating Institute (HVI), 230 N. Michigan Ave.,Chicago, IL 60601:

It was mentioned that both home owners refused to let their powered attic ventilators be turned off for longer than two weeks for the study. Were the home owners asked details of their experience or observation, such as improved comfort or less energy use with power venting, and was there any attempt to take measurements specific to these details?

R. A. Grot: No effort was made to evaluate the comfort of the occupants; however, the reluctance to have the power ventilator turned off was expressed before it was turned off (that is, the fan-on phase of the evaluation was performed before the fan-off phase), and therefore could not have been empirically based.

ATTIC VENTILATION RESEARCH CONDUCTED BY ARKANSAS POWER AND LIGHT COMPANY

by

Fred B. Clark, P.E.
Manager of Technical Services
Arkansas Power and Light Company
P. O. Box 551
Little Rock, Arkansas 72203

Arkansas Power and Light Company conducted an attic ventilation test during the summer of 1976 to compare the effects of power ventilators and wind turbines on residential cooling loads. Both systems were installed on a home in North Little Rock, Arkansas, and operated during alternate periods of time with energy inputs and temperatures recorded on magnetic tape. The results showed comparable savings, with the power ventilator having a slight advantage.

In view of the limited test facilities and unusually cool weather in 1976, the project has been expanded to include continuous ridge vents, and equipment has been installed on six homes for testing during the summer of 1978. Each type of system is installed in two of the six homes along with gravity ventilation systems meeting HUD Minimum Property Standards. Data will be recorded on all homes during the same operating periods and the results compared with gravity ventilation on the same house for days with similar weather conditions.

Data recording will begin June 1, 1978, and the test results will be available in the fall of 1978.

The benefits of attic ventilation have long been recognized, but very little has been done to document the relative energy efficiencies and cost effectiveness of different ventilating modes. Many utility companies have recommended thermostatically controlled power ventilation for all residential applications, on the theory that a much greater decrease in attic temperature would be realized than with other ventilating methods and that positive control is necessary on many days when outdoor temperature is high and wind velocity is low.

In 1976, the increasing awareness of rising costs and the need for conservation prompted Arkansas Power and Light Company to perform some limited testing to either verify or to alter its previous assumptions and recommendations. The objective was to observe changes in energy consumption, attic temperature and ceiling temperature on comparable days using gravity (gable only) ventilators, turbine ventilators with soffit intake, and power ventilators.

A 2,310 square foot ($207.9m^2$) home near North Little Rock, Arkansas, was selected for the test. It had a dark green roof, a westerly orientation, and was shaded only briefly in the early morning hours. The attic had six inches (15.2cm) of loosely blown glass fiber insulation; 54 square feet ($4.9m^2$) of glass area was exposed to solar radiation; and a 4-ton cooling unit was installed to serve a calculated 3-ton requirement. Cooling supply ducts were located in the attic and were tightly wrapped with R-7 insulation. The return air system was located within the conditioned space.

Magnetic tape load survey recorders were installed to measure kW input to the compressor and condenser fan, temperature of attic air, and temperature of ceiling drywall at two locations. Indoor air temperature and relative humidity were measured with a recording hygrothermograph and energy used by the power ventilator was recorded by kWh meters.

Attic air temperature was measured with shielded temperature probes placed three feet above the insulation, and ceiling temperatures were measured above the drywall over a hallway and near the center of the den. Outdoor temperatures and wind velocities were recorded three miles away by the US Weather Bureau.

The following ventilating systems were tested in sequence and are identified by the indicated letters A, B, C and D in future references:

A. Gable End Ventilation: Two triangular gable vents with 1,316 square inches ($0.86m^2$) of net free area were tested as originally installed by the builder. Orientation was north-south.

B. Wind Turbines with Soffit Intake: Two wind turbines rated 668 cfm ($18.9m^3$/min) at 11 mph (17.7 km/hr) wind velocity were installed 1/3 the house length from each end. 1,075 square inches ($0.7m^2$) of net free soffit intake area was installed and the gable ends blocked. This would produce 10 air changes per hour at 11 mph (17.7km/hr) wind velocity.

C. Two Power Ventilators: Two thermostatically controlled power ventilators certified by HVI at 1,130 cfm ($32m^3$/min), were installed to replace the wind turbines and the net free area of the soffit vents was increased to 1,920 square inches ($1.24m^2$), or 0.8 square inches per rated cfm ($.15cm^2/m^3$/min). This provided 16 attic air changes per hour. The thermostats were set to turn the fans on at 100° F (38° C) and off at 85° F (29° C) attic temperature.

D. One Power Ventilator: One power ventilator was de-energized, leaving one operating (1,130 cfm; $32m^3$/min.) to provide eight attic air changes per hour. Soffit vent area was not changed from the 1,920 square inches ($1.24m^2$) used on Test C.

Each system was tested for periods of one to two weeks to ensure comparable outdoor conditions. The indoor thermostat was set at 75° F (24° C) throughout the test but the recorded indoor temperatures varied from 72° F (22° C) to 78° F (26° C) during the test period. Data recording was limited by an abnormally cool summer with only two valid test days for System D.

Since the tests were performed on different days, only 19 days were selected as valid for comparison. There were days with maximum temperature above 92° F (33° C) and with at least 85 degree-hours cooling above 80° F (47 degree-hours above 27° C). Comparisons were made on a "per degree-hour" basis since the occurring temperatures and the length of test periods were difficult for each test.

The data collected are summarized in Tables 1 through 4. Table 1 shows a significant reduction in peak attic temperature by both power ventilators and turbine ventilators. The single power ventilator with eight air changes per hour appeared to be just as effective as two power ventilators with 16 air changes per hour.

Table 2 compares energy consumption for each test on a "per degree-hour" basis (degree-hours are the summation of differences between outdoor temperature and 80° F (27° C). The wind turbines used 18.4% less, two power vents 21.6% less and a single power vent 25.5% less energy than gable vent alone.

Table 3 is similar to Table 2 except that only those hours when attic temperature exceeded indoor temperature are included. The savings are of the same relative ranking but slightly greater magnitude.

Table 4 compares the summations of positive temperature differences between attic temperature and indoor temperature and correlates reasonably well with the comparisons presented in Tables 2 and 3.

Figure 1 is a plot of the data recorded on August 25th with one power ventilator operating. The attic temperature dropped well below the outdoor temperature, presumably due to the effect of radiation to the sky. This is typical of other days during the test. The attic air temperature was below the dewpoint of the outdoor air for 67 hours during 15 of the 17 days that the power ventilator was tested. This is not desirable but is not considered to be of major consequence.

The variations in ceiling drywall temperature were less than anticipated. This is probably because of the location of the temperature probes within six feet (1.8m) of the ceiling diffusers which swept cool air across the ceiling.

Conclusions

On the valid test days of the study, with outdoor temperatures between 92° F (33° C) and 97° F (36° C), the compressor and condenser fan used 18.4% less energy per degree-hour with wind turbines than with gable vent only. The total energy requirement, including the vent fan, was 21.6% less per degree-hour for two power ventilators and 25.5% less for one power ventilator than for gable ventilation only.

The number of valid test days were too limited to support a firm conclusion on the relative merits of each system. Both the turbines and power ventilators were energy effective during the warmer days but this does not necessarily extend to seasonal effectiveness or to cost effectiveness. Further testing is needed and is planned for a larger test sample. The amount of attic insulation also plays an important part in both energy and cost effectiveness and was not addressed in this study.

Proposed Tests

A more extensive test was scheduled for the summer of 1977. Continuous ridge ventilators were to be tested in addition to wind turbines and power vents and two systems of each type were to be installed on six homes in Jacksonville, Arkansas. The test equipment was not delivered in time to complete the test in 1977 but some data were recorded in August and testing will be resumed on June 1, 1978.

The six homes selected are of the same size and design but with different roof colors. Three face east and three face west but all have wide eaves, roof ridges with the same north-south orientation, and gable ventilators meeting HUD Minimum Property Standards.

Rather than replace the ventilating systems for sequential tests, power ventilators have been installed in two homes, wind turbines in two, and continuous ridge vents in two. The existing gable vent systems will be tested first and the other systems tested later and compared with gable vents on the same house. All systems will be tested at the same time on different houses rather than at different times on one house. Soffit vent area will also be varied for comparison.

The following items will be measured by magnetic tape load survey recorders:

1. kW input to the compressor and condenser fan

2. Attic air temperature

3. Duct inlet and outlet temperatures in three homes

4. Ceiling drywall temperature in three homes

Energy used by the power vents will be recorded by kWh meters and indoor temperature will be recorded at selected times with a hygrothermograph.

Some preliminary data were gathered during August of 1977 by operating all six homes for two weeks with gable ventilation only. For the next two weeks, two houses were tested with wind turbines, two with power ventilators, and two with gable vents only. The ridge vents had not been installed at that time. Ceiling temperatures showed a greater variation than in the previous North Little Rock test (75° F (24° C) to 87° F (31° C)) but even more noticeable were the variations due to roof color. Figure 2 shows attic temperature, ceiling temperature, and compressor kW for August 8, 1977, for a light brown roof with gable ventilation only and 94° F (34° C) maximum outdoor temperature. The cooling equipment appears to be undersized, which would contribute to the swing in ceiling temperature and detract from measured energy savings due to attic ventilation.

Figure 3 compares three homes, also on August 8th, with gable ventilation only. The recorded attic temperatures were highest for a dark brown roof, lower for dark gray and lowest for light brown.

Figure 4 compares three homes with different ventilation modes on comparable days (August 8th and September 1st). A dark gray roof with gable ventilation only recorded the highest temperature; dark gray with turbine ventilator was 10° F (5.5° C) lower, and a black roof with power ventilator 13° F (7.2° C) lower.

The results of this test will be available in the fall of 1978 and will hopefully give a better insight into the effects of roof color, ventilating mode, and attic duct system on energy efficiency, cost effectiveness, and the owner's comfort and satisfaction.

TABLE 1. EXTREME TEMPERATURE DATA FOR DAY OF HIGHEST ATTIC TEMPERATURE

System	Date	Outside		Attic		Outside		Attic	
		High ^{o}F	Time	High ^{o}F	Time	Low ^{o}F	Time	Low ^{o}F	Time
A	7-13-76	92	4 PM	123	4 PM	69	4 AM	60	6 AM
B	7-27-76	97	2&3 PM	119	4 PM	75	5 AM	67	9 AM
C	8-6-76	94	3 PM	104	4 PM	73	5 AM	66	7 AM
D	8-26-76	95	3 PM	103	5 PM	70	5&6 AM	63	8 AM

System A - Gable Vents - HUD Minimum Property Standards

System B - Two Turbine Ventilators

System C - Two Power Ventilators

System D - One Power Ventilator

Temperature $^{o}C = 5/9$ (Temperature $^{o}F - 32^{o}$)

TABLE 2. SUMMARY OF DATA FOR ALL HOURS OF VALID TEST DAYS

	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)
			Cooling Degree-Hours F		kWh				
System	Average Wind Velocity MPH	Valid Test Days	Total	Per Day	Compressor & Cond. Fan	Power Vent	Total	Total kWh* Per Cooling Degree-Hour F	Percent** Reduction (Col. h)
A	5.6	4	421	105.3	214.8	0	214.8	.510	--
B	7.4	6	796	132.7	331.1	0	331.1	.416	18.4
C	6.9	7	747	106.7	283.5	15.4	298.9	.400	21.6
D	5.8	2	200	100.0	73.4	2.5	75.9	.380	25.5

System A - Gable Vents - HUD Minimum Property Standards

System B - Two Turbine Ventilators

System C - Two Power Ventilators

System D - One Power Ventilator

Degree-Hours C = 5/9 X Degree-hours F.

Kilometers/hour wind velocity = 1.61 X miles/hour

*Column "g" divided by Column "c".

**Percentage reduction of total energy for cooling per degree-hour (column h) when compared to System A.

TABLE 3. SUMMARY OF ENERGY USE DURING HOURS WHEN ATTIC TEMPERATURE EXCEEDED INDOOR TEMPERATURE ON VALID TEST DAYS

	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)
			Cooling Degree-Hours F		kWh				
System	Average Wind Velocity mph	Total Hours	Total	Per Hour	Compressor & Fan	Power Vent	Total	Total kWh* Per Cooling Degree-Hour F	Percent** Reduction (Col. h)
A	6.7	51	379	7.43	166.3	0	166.3	.439	--
B	8.4	75	717	9.56	241.9	0	241.9	.337	23.2
C	8.2	81	689	8.51	203.0	15.4	218.4	.317	27.8
D	6.2	24	188	7.83	53.9	2.5	56.4	.300	31.7

System A - Gable Vents - HUD Minimum Property Standards

System B - Two Turbine Ventilators

System C - Two Power Ventilators

System D - One Power Ventilator

Degree-Hours C = 5/9 X Degree-hours F.

Kilometers/hour = 1.61 X miles/hour

*Column "g" divided by Column "c".

**Percentage reduction of total energy for cooling per degree-hour (Column h) when compared to System A.

TABLE 4. SUMMARY OF HOURLY TEMPERATURE DIFFERENTIALS WHEN ATTIC TEMPERATURE EXCEEDED INDOOR TEMPERATURE ON VALID TEST DAYS

	(a)	(b)	(c)	(d)	(e)	(f)	(g)
			Cooling Degree-Hours F				
System	Average Wind Velocity mph	Total Hours	Total	Per Day	Total** T.D. Hours	Temperature*** Differences Per Cooling Degree-Hour F	Percent Reduction (Col. f)
A	6.7	51	379	7.43	1243.1	3.28	--
B	8.4	75	717	9.56	1632.4	2.28	30.5
C	8.2	81	689	8.51	1373.2	1.99	39.3
D	6.2	24	188	7.83	427.4	2.27	30.8

System A - Gable Vents - HUD Minimum Property Standards

System B - Two Turbine Ventilators

System C - Two Power Ventilators

System D - One Power Ventilator

Degree-Hours C = 5/9 X Degree-hours F.

Kilometers/hour wind velocity = 1.61 X miles/hour

*"Hourly Temperature Difference": Difference between the attic and indoor temperatures.

**"Total T.D. Hours": Summation of differences between attic and indoor temperatures for each hour the attic air temperature exceeds the indoor temperature on valid test days.

***"Temperature Differences Per Cooling Degree-Hour": Column "e" divided by Column "c".

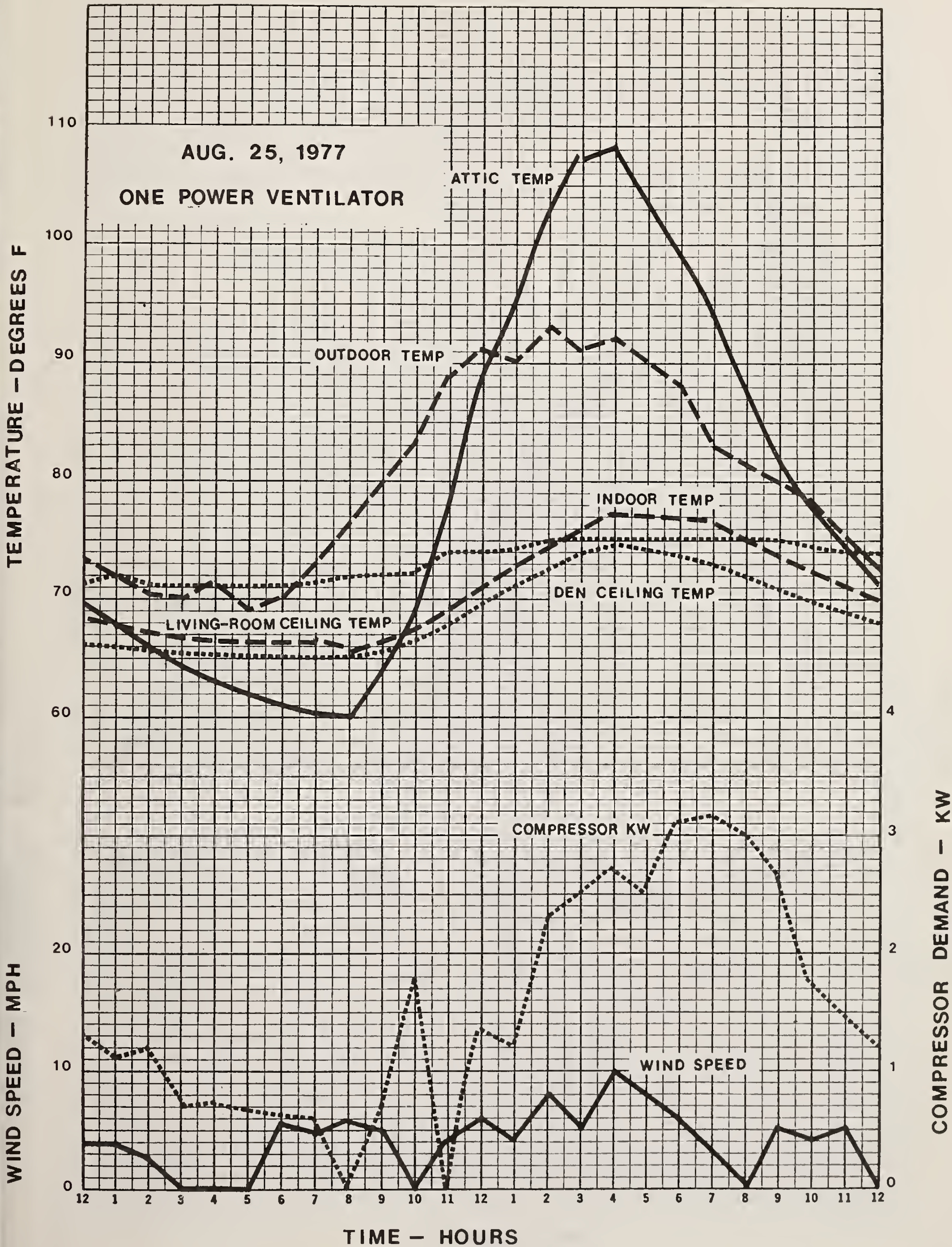


Figure 1. Temperatures and Energy Use For North Little Rock Test House

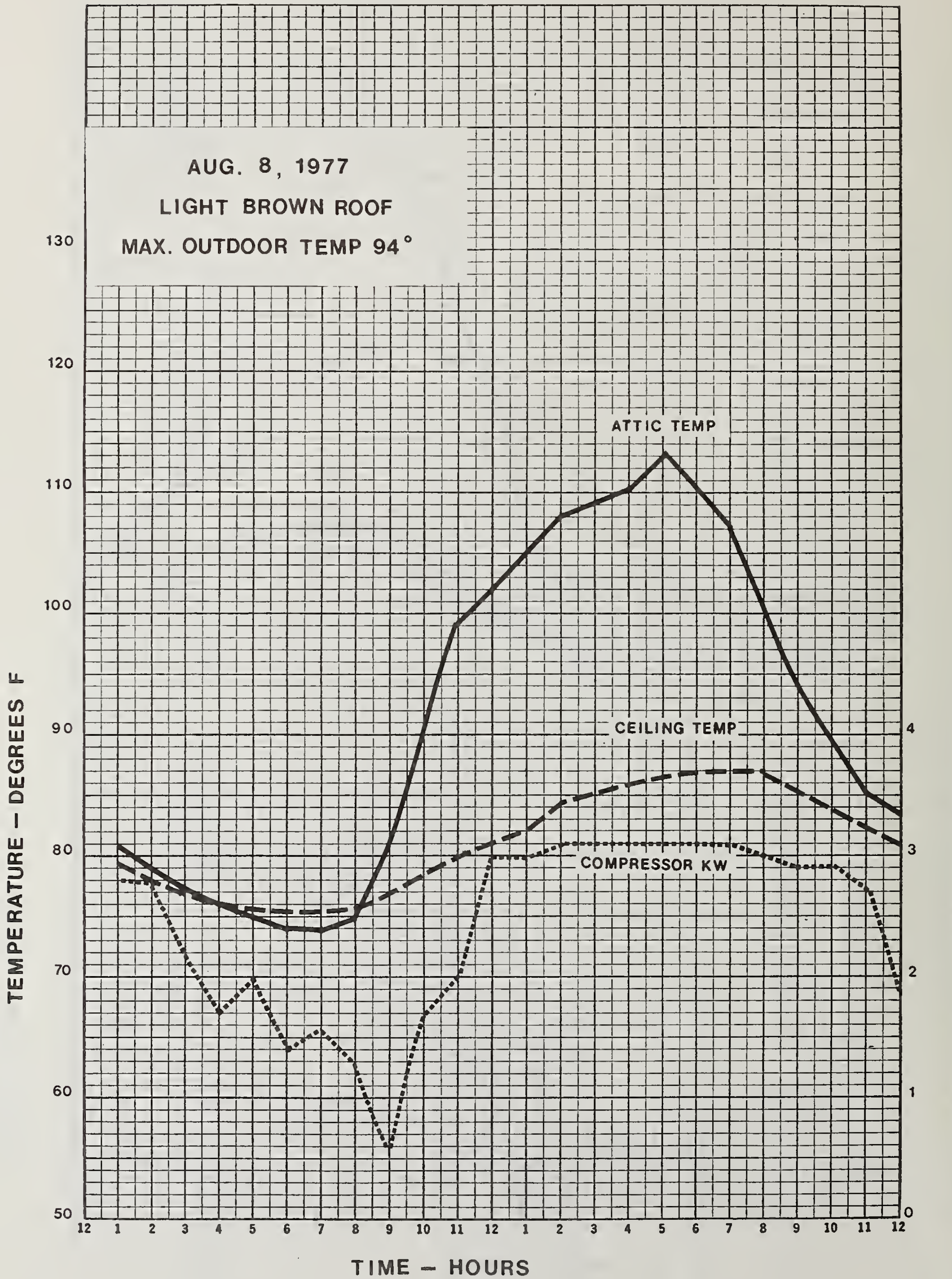


Figure 2. Temperatures and Energy Use For Jacksonville Test House

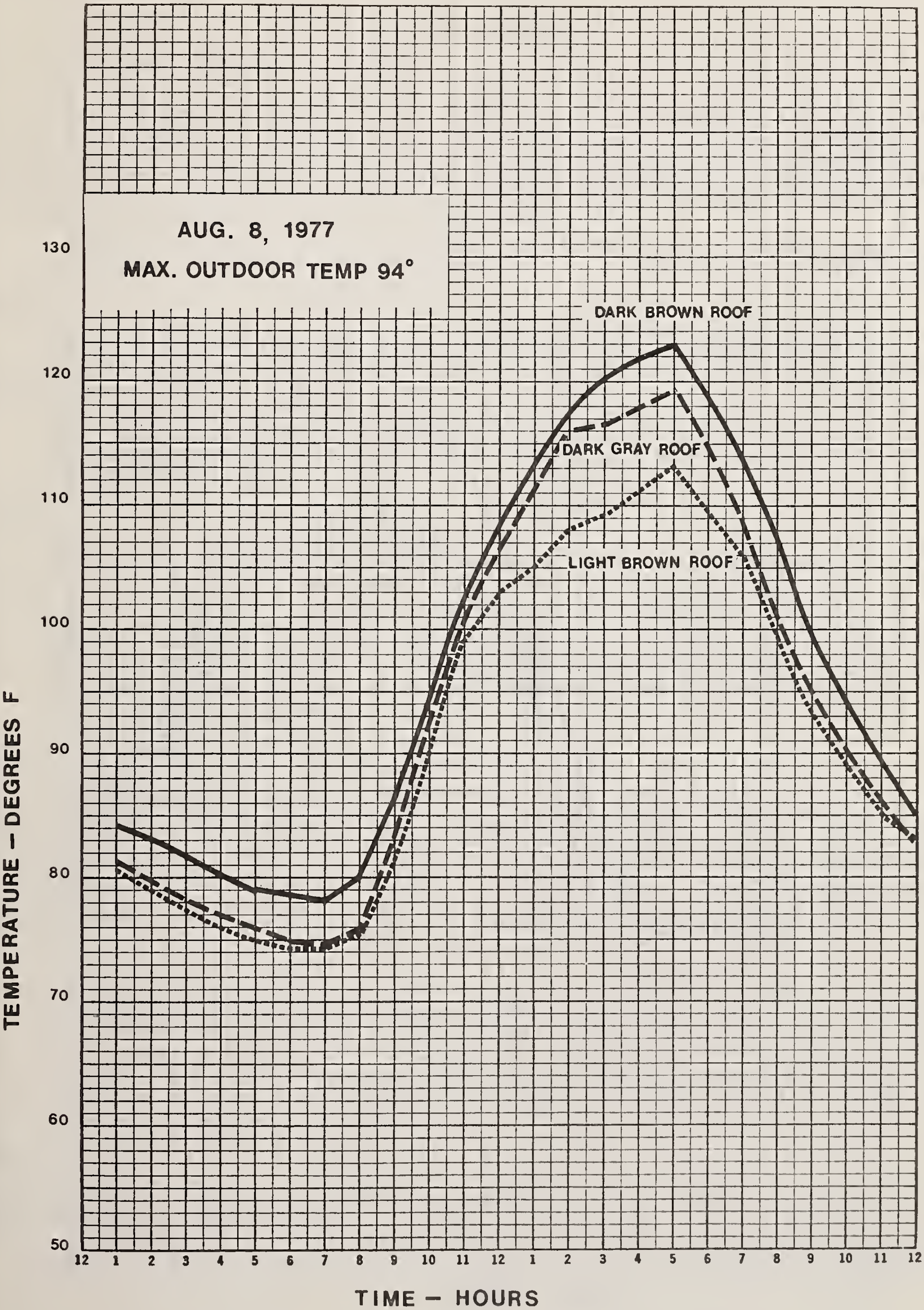


Figure 3. Attic Temperatures for Three Roof Colors on Three Jacksonville Test Homes

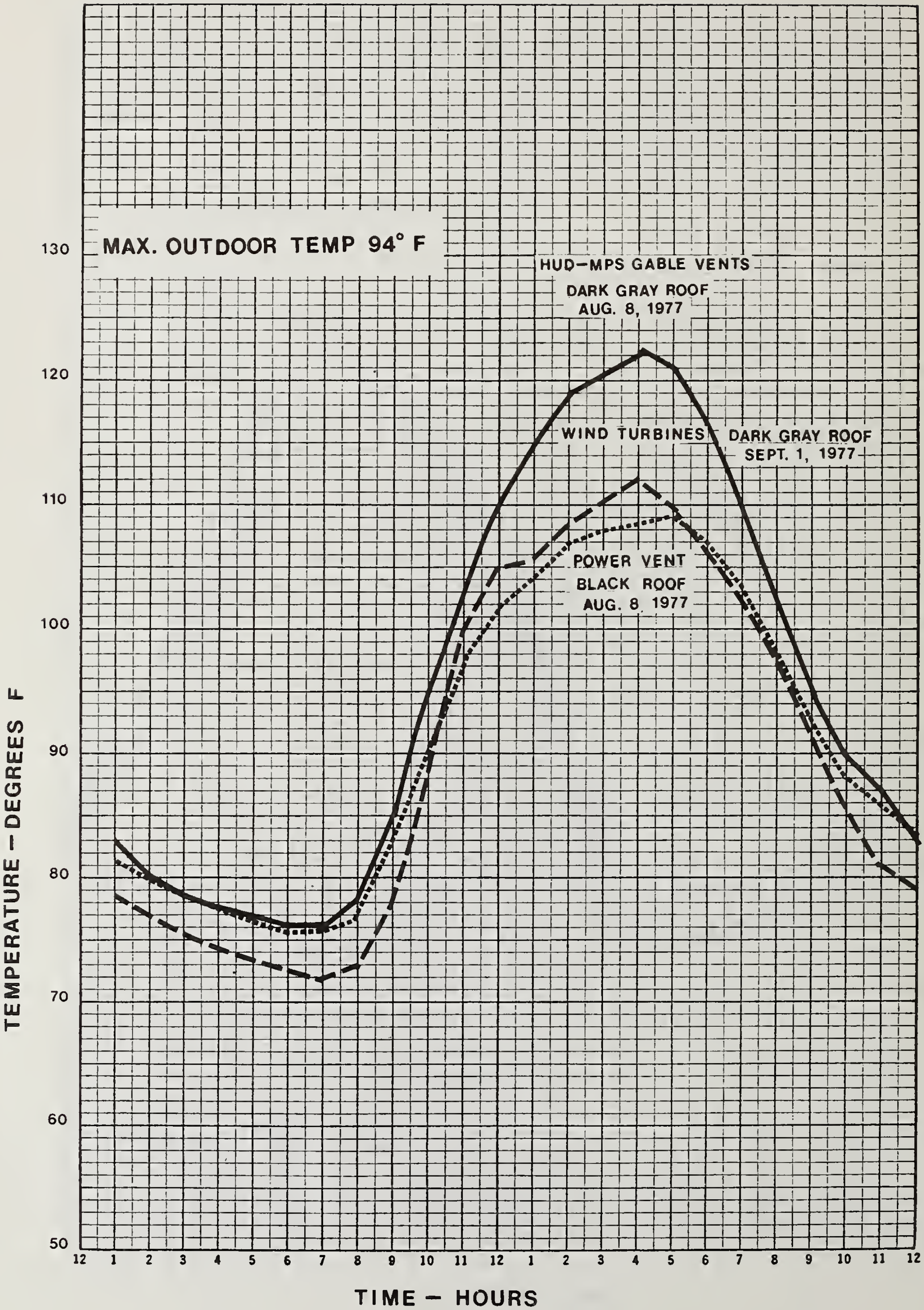


Figure 4. Attic Temperatures for Three Ventilating Modes on Three Jacksonville Test Homes

Questions and Answers

D.M. Burch, National Bureau of Standards, Washington, D.C.:

The cooling load of a house strongly depends not only on the outdoor temperature, but also on the amount of solar radiation and the wind velocity. In your paper, you have applied a cooling-degree-hour method to short periods of test (all test periods were less than one week) in order to derive differences in cooling energy consumption due to increased attic ventilation. The observed differences in cooling energy requirement could have been due to differences in solar radiation instead of attic ventilation.

F.B. Clark: Yes, we agree that solar radiation and wind velocity can affect the results.

Burch:

The analysis presented in your paper is valid only if there exists a correlation between air conditioner energy consumption and cooling degree hours. How do you know that such a correlation exists?

Clark: We cannot be sure that a direct correlation exists between energy consumption and degree hours. We used this, however, since we had no other correlation that seemed more acceptable.

Burch:

Are the reductions in cooling energy requirement presented in your paper applicable to maximum-load condition or are they reductions in daily cooling energy requirement?

Clark: The reductions observed in Table 2 of my paper were for all hours of the valid test days, while those in Table 3 were only for daytime hours when attic temperature exceeded indoor temperature. In both instances, however, they relate only to the hottest days of the month, and I would expect considerably less savings for an entire month or an entire cooling season.

B.A. Peavy, National Bureau of Standards, Washington, D.C.:

Wind velocities for the case with gable vents do not specify wind direction. For natural ventilation this is necessary, and comparisons should not be made without this parameter as well as other parameters. Gables usually have associated soffit vents to provide temperature head. A perusal of data indicates that the wind speed during gable tests was considerably less than that for the other tests, particularly for those time periods during the hottest parts of the day. The "gable" tests were performed at time of the year when the sun was higher in the sky when compared to the other tests. There can be considerably less solar radiation incident on a horizontal surface, which is what one effectively has for roofs.

Clark: It is very difficult to find comparable test days with the same solar radiation, temperature and wind conditions, but we attempted to pick days when all of these factors were reasonably in agreement. A table of excerpts from the July and August reports of the U.S. Weather Station #13963 at Little Rock, Arkansas, for the selected test days is attached. Wind and temperature conditions were measured at Adams Field, 5.7 miles south of the test site, and solar radiation was measured at the North Little Rock Airport, which is 2.3 miles north of the test site.

Peavy:

The paper has ignored the relative humidity of the outdoor air. No comparison can be made unless the parameter is known.

Clark: We agree that outdoor relative humidity is one of many things that affect cooling load, and this is one of the reasons that we have been reluctant to attribute

WEATHER CONDITIONS FOR ATTIC VENTILATION TEST

TEST	DATE	MAXIMUM OUTDOOR TEMPERATURE °F	MAXIMUM ATTIC TEMPERATURE °F	NORTH LITTLE ROCK, ARKANSAS 3:00 P.M. WIND VELOCITY KNOTS	WIND DIRECTION DEGREES	SKY COVER (TENTHS)	U.S. WEATHER STATION #13963 TOTAL DAY AVERAGE WIND KNOTS	RESULTANT WIND KNOTS	RESULTANT DIRECTION DEGREES	SUNSHINE % OF AVAILABLE
A	July 13, 1976	92	123	0	0	8	3.6	2.2	29	87
Gable	20	93	114	9	20	5	9.2	8.7	22	94
Vents	21	94	118	9	20	3	6.5	6.3	24	88
	22	93	113	10	17	5	6.9	6.0	21	100
B	July 23, 1976	94	114	10	23	5	3.6	1.2	22	100
Turbines	26	98	113	9	23	5	6.8	5.7	23	81
	27	97	119	12	24	3	10.1	9.3	22	87
	28	96	117	8	21	3	10.9	10.5	22	90
	29	99	114	10	22	6	10.4	8.9	21	59
	30	94	115	1	25	7	6.6	5.8	22	98
C	August 6, 1976	96	104	8	34	2	10.4	8.3	30	89
Two	11	95	103	6	15	0	6.0	5.3	17	97
Power	12	95	100	10	23	5	8.8	7.4	22	100
Vents	13	97	103	10	14	5	6.8	4.8	18	95
	16	96	103	11	9	5	6.5	3.6	9	85
	17	95	99	10	7	3	8.9	8.5	7	90
	20	94	100	12	11	2	7.2	5.2	9	85
D	August 25, 1978	94	107	10	5	3	5.2	1.8	6	83
One	26	97	103	4	14	3	4.6	3.7	15	93
Power										
Vent										

measured savings of compressor energy to any single factor in a test home. This is true not only in attic ventilation tests but in other thermal research projects we have conducted.

Peavy:

Where is the location of temperature sensing elements in the attic? Where is the schematic of the attic? Is their location really indicative of the average attic temperature for all the tests?

Clark: The temperature sensing element was located 12 inches above the insulation near the center of the house, as shown on the attached attic schematic.

Peavy:

Why wasn't the steady-state heat balance between attic and the inside performed?

Clark: With indoor temperature at 75° and duct temperature at 50°, the calculated steady-state sensible heat gain through the attic is as follows:

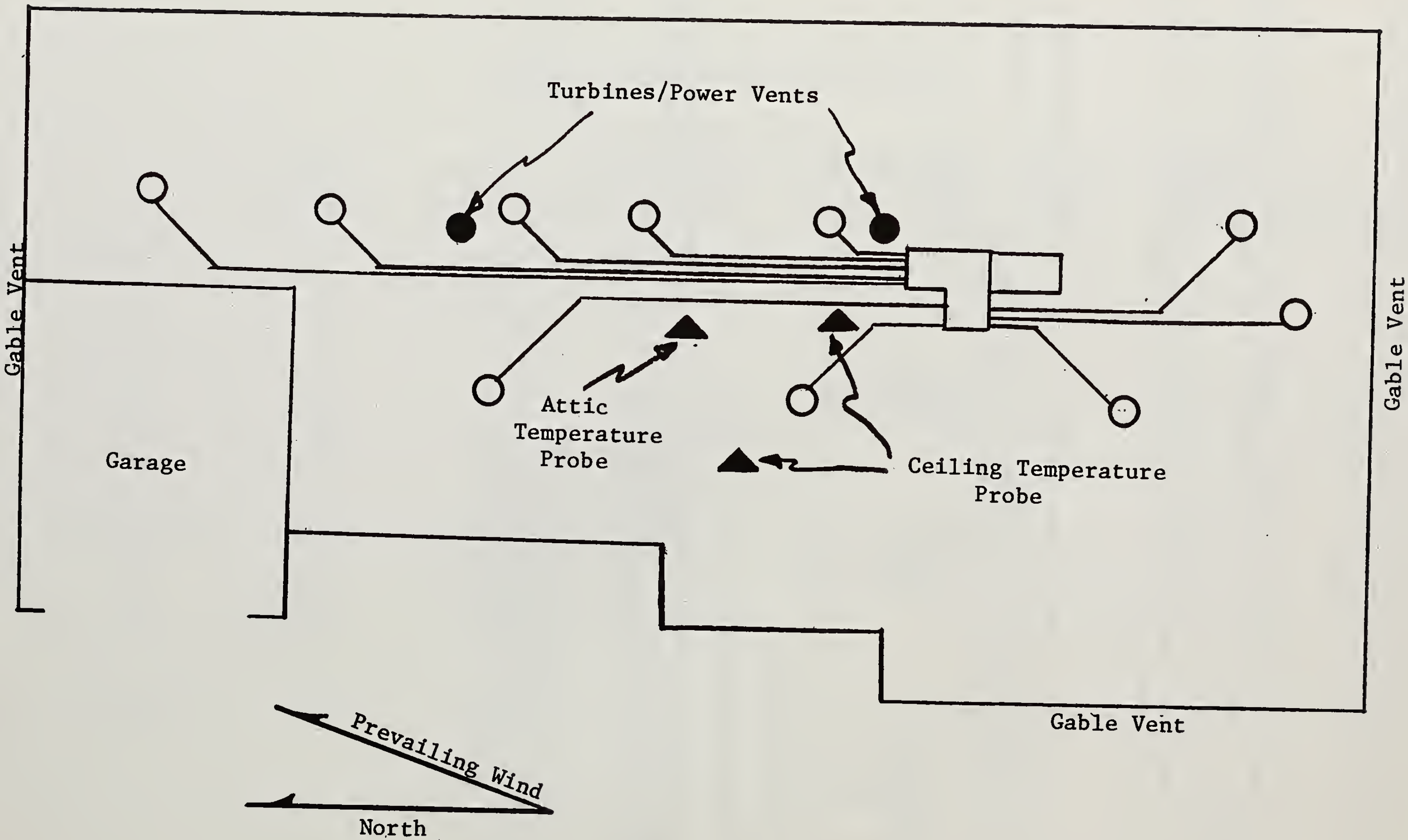
Attic Temperature	Ceiling Gain	Duct Gain	Total Gain BTUH
123°F	13,996	2,856	16,852
104°F	10,353	1,725	12,078

This is a 28.3% reduction.

D.T. Harrje, Princeton University:

How can you justify the stated savings with the turbines & power ventilators using the single test house when tests were run over a time span when the sun effects were changing, where the number of hours above inside temperature changed by 8%, and where only minor differences have been seen in attic temperatures between tests A and B?

Clark: We agree that testing a single house on different days for different ventilation systems is not the preferred method, and for that reason we have continued our testing through 1978 using six identical homes with two systems of each type. We have not attempted to justify the reported savings, but only to report the data recorded and the observed conditions.



Schematic of Attic For

Test Home in North Little Rock, Arkansas

Scale 1/8" = 1'

VENTILATING RESIDENCES AND THEIR ATTICS FOR ENERGY CONSERVATION
-- An Experimental Study

D.M. Burch and S.J. Treado
Center for Building Technology
National Bureau of Standards
Washington, D.C. 20234

Three identical houses in Houston, Texas were extensively instrumented for measuring their air conditioner energy consumption and ceiling and duct heat-gain rates. Comparative tests were conducted to investigate differences in house performance due to increased attic ventilation. The performances of a roof-mounted power ventilator, a ridge vent, and wind-driven turbines were compared to the performance of soffit venting meeting the requirements of the HUD Minimum Property Standards. The effect of various attic ventilation techniques on the indoor-comfort condition was also investigated.

Separate tests were conducted to investigate the effect of whole-house ventilation on the cooling energy requirement and the indoor-comfort condition.

Key Words: Attic ventilation; energy conservation; whole-house ventilation.

Introduction

This study was carried out during the summer of 1977 under the joint sponsorship of the Department of Energy and the National Bureau of Standards, and in cooperation with the American Ventilation Association and the Home Ventilating Institute, to provide technical information with regard to summer house ventilation that could be used in future broad-scale energy conservation programs for existing and new residences.

With the advent of increasing energy costs, homeowners have become interested in energy conservation procedures for reducing their daily utility costs. Utility companies are particularly interested in those technologies which reduce the electric energy requirement at peak-load condition. This paper focuses on two methods for reducing the energy requirement for residential space-cooling; namely, whole-house ventilation and attic ventilation. The effect of these strategies on the daily cooling energy requirement, the cooling energy consumption at maximum-load condition, and the indoor comfort condition is investigated.

An often overlooked method for saving energy in the summer is whole-house ventilation. In this method, a large fan is mounted in the ceiling at a central location of the living space of a residence. During mildly hot periods (i.e., outdoor temperature less than 82°F (28°C)), the air conditioner is turned off, the windows are opened, and the whole-house fan is operated to exhaust air from the living space at approximately one air change per minute. This air movement increases the evaporative and sensible cooling from occupants and produces a sensation of comfort even though the indoor temperature is elevated above the normal comfort point for still air. Since a whole-house fan consumes approximately one-tenth the energy of an operating central air conditioner, considerable energy savings are possible.

In the case of attic ventilation, various natural and power ventilation systems are available for removing heat from attic spaces during the hot part of the day. They reduce the heat gain through the ceiling and into air conditioning ducts that pass through the attic space. The effectiveness of attic ventilation in reducing air conditioning energy requirements is controversial. A recent study [1]* conducted by the Arkansas Power and Light Company showed large energy savings as a result of using power venting and turbine venting. Other studies [2-4], however, have shown that attic ventilation did not produce large energy savings. An important finding that has surfaced in these other studies is that the ceiling heat gain is a rather small fraction of the daily cooling load for moderately and heavily insulated ceilings.

Description of the Test Houses

The three test houses used in this study were new, wood-frame ramblers with brick-veneer outer covering. They were constructed over slabs-on-grade. They had gable roofs with light-brown shingles. The floor plan, orientation, and construction details for the houses were identical. They were located in a row along a residential street with a vacant lot between adjacent houses in a suburb of Houston, Texas. The living space had a floor area of 1020 ft^2 (94.8 m^2). Each house also contained a garage having a floor area of 439 ft^2 (40.8 m^2). A photograph of one of the test houses is given in Figure 1. A floor plan of the test house in given in Figure 2.

The walls of the test houses were insulated with full-thickness rockwool blanket insulation installed between 2 x 4 studs placed 16 inch (0.41 m) on center. Six and one-half inches (0.17 m) of loose-fill rockwool insulation was installed between 2 x 4 wood joists (placed 16 inch (0.41 m) on center) over the ceiling of one of the houses. The other two houses contained 4.0 inches (0.10 m) of rockwool ceiling insulation. Ceiling insulation was not placed over the garages of the houses.

The houses each contained six double-hung, single-pane, metal-sash windows. A sliding-glass door was installed in the living room. The area of the fenestration surfaces was 116 ft^2 (10.8 m^2) and represented 11% of the floor area of the houses. The interior of all fenestration surfaces was equipped with draperies.

The mechanical equipment for providing space heating and cooling was located inside a mechanical closet in the center of the living space. The heating plant consisted of a gas-fired, forced-air furnace having an output capacity of 64,000 Btu/h (18,800 W). The cooling system was a 2-1/2 ton (30,000 Btu/h or 8,790 W) split vapor-compression refrigeration system. An A-frame evaporator was mounted in the supply plenum above the heat exchanger of the gas-fired furnace. The compressor and condenser of the cooling equipment were contained in an outdoor unit located at the side of each house. The thermostat for the heating and cooling system was located on an interior partition wall in the hallway connecting the living room to the bedrooms.

The air-distribution system contained a common air return which drew air from the living space at the bottom of the mechanical closet. After passing through the heating and cooling plant, conditioned air was circulated through ducts (6.0 to 11 inches (0.15 to 0.28 m) in diameter), located in the attic space, to ceiling-mounted supply registers. The ducts were lined with 1.0 inch (0.025 m) (R-3.1) glass-fiber insulation and had an exterior surface area of approximately 180 ft^2 (16.7 m^2).

Description of Ventilation Equipment and Systems

Ventilation Equipment

Each of the test houses was equipped with the following ventilation equipment:

° Two 14-inch (0.36 m) diameter wind-driven turbines rated at 918 ft^3/min (0.433 m^3/s) at a temperature difference of 40°F (22°C) between the attic and outdoor air and a wind speed of 9 mph (4 m/s).

° One 42-inch (1.1 m) diameter, 2-speed, whole-house fan equipped with a 1/3 H.P. (250 W) motor. The whole-house fan was rated at 8,840 (4.17) and 13,200 ft^3/min (6.23 m^3/s) at low and high speeds, respectively.

° One 14-inch (0.36 m) diameter roof-mounted power attic ventilator. At a back pressure of 0.03 psig (200 Pa), it was rated at 1,260 ft^3/min (0.595 m^3/s). The attic ventilators were equipped with thermostats designed and adjusted at the factory to start at 100°F (38°C) and stop at 85°F (29°C). The thermostats were mounted about 1 foot (0.31 m) below the ridge and approximately 2 feet (0.61 m) to one side of the ventilator. The installed thermostats did not perform as designed and more than normal fan operation occurred.

* Numbers in brackets refer to literature references cited at the end of the paper.

Figure 1. Photograph of One of the Test Houses.

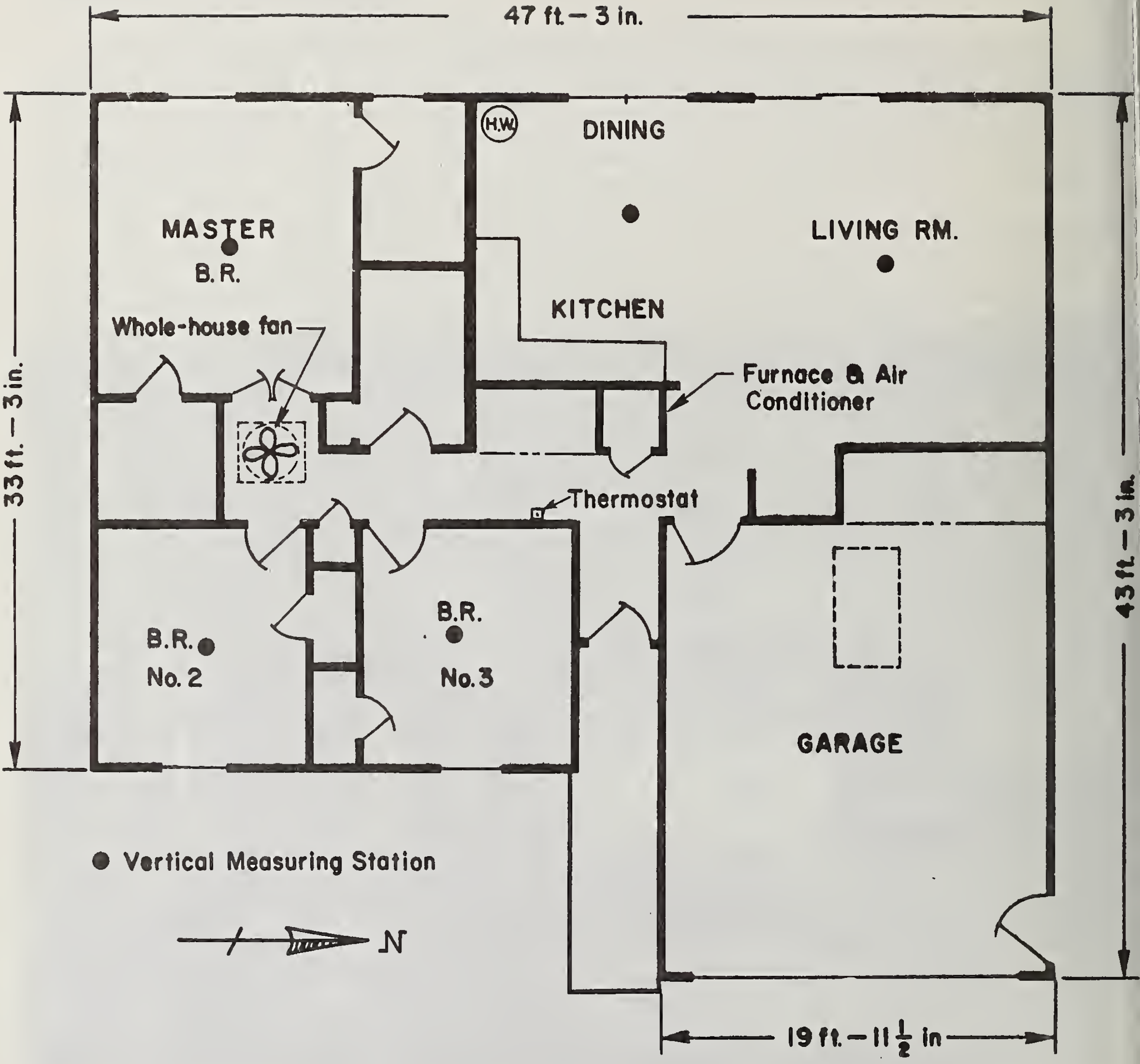


Figure 2. Floor Plan of the Test Houses.

- ° Forty 8x16 inch (0.20 x 0.41 m) soffit vent openings, each having a net free open area of 62 in^2 (0.040 m^2). Twenty-three of the soffit vents used in the house with 6.5 inches (0.17 m) of ceiling insulation had a net free open area of 49 in^2 each (0.032 m^2) instead of 62 in^2 (0.040 m^2).
- ° Three aluminum gable-end openings with fixed louvers. These openings were sealed off during the present study.
- ° One 49-foot (15 m) continuous ridge vent. The ridge vent had a net free open area of 18 in^2 per linear foot (0.038 m^2 per linear meter).

The location of the attic ventilation equipment is shown in Figure 3. The whole-house fan was mounted in the ceiling at the end of the hallway adjacent to the bedrooms (see Figure 2).

Ventilation Systems

In designing the attic ventilation experiments for the present study, it was decided to use the minimum amount of soffit ventilation specified in the HUD Minimum Property Standards [11] as a reference base of comparison for the other ventilation systems. The HUD Minimum Property Standards require 1 ft^2 for every 150 ft^2 (or 1 m^2 for every 150 m^2) of attic floor for an attic ventilated using soffit vents only. The test house of the present study had an attic floor area of 1770 ft^2 (164 m^2), so that the required net free open area for the soffit vents was 11.8 ft^2 (1.10 m^2). This amount of net free open area was achieved by sealing some of the soffit openings with duct tape. An attempt was made to provide uniformly distributed soffit openings in the eaves. This soffit ventilation will henceforth be referred to as "soffit venting".

The various attic ventilation systems evaluated included power venting, turbine venting, ridge venting, and roof venting. For roof venting, the two turbine ventilators on a house were removed and weatherization caps were placed over the two openings in the roof. In structuring these attic ventilation systems, it was decided that the primary ventilation equipment would be added as a retrofit to a house that already had soffit venting. The soffit venting would serve as the air intake opening for the other ventilation systems.

In the case of power venting, ridge venting, and turbine venting, the soffit venting provided a net free open area which exceeded the minimum recommended amount specified in Refs [5,6]. Separate air-infiltration measurements described in Ref. [9] indicated that the operation of the roof-mounted power ventilator did not draw living-space air through the ceiling into the attic space and thereby increase the air-infiltration rate for the living space.

Whole-House Ventilation Procedure

During whole-house ventilation tests, the windows of a house were closed at 0745 and the air conditioner was turned on with the indoor thermostat set at its selected level. Whenever the indoor temperature rose above the set point, the air conditioner would operate and cool the living space. The indoor space was cooled in that fashion until the outdoor temperature dropped below 82°F (28°C)*. At this point, the air conditioner was turned off, the windows were opened approximately 6 inches (0.2 m), and the whole-house fan was operated at high speed, thus providing an air-exchange rate for the living space of approximately 0.8 air change per minute. The whole-house fan was operated until 0400, at which time it was shut off by a clock timer. The foregoing procedure was repeated for six consecutive days of the test period.

When whole-house ventilation tests were conducted, all the ventilation openings (including ridge vent, gable vents, turbine openings, attic fan openings, and soffit vents) were fully opened. In addition, the attic stairwell door located in the garage was opened and the garage door was raised approximately 1 foot (0.3 m), providing additional ventilation exhaust opening for the whole-house fan. The combination of these openings provided a net free open area of approximately 42 ft^2 (3.9 m^2), which significantly exceeds the minimum recommended ventilation exhaust opening of 8.3 ft^2 (0.77 m^2) calculated using the procedures outlined in Ref. [5].

---

* The American Ventilation Association and the Home Ventilating Institute advocate that whole-house ventilation can be used at outdoor temperatures 82°F (28°C) and lower to produce comfortable indoor conditions.

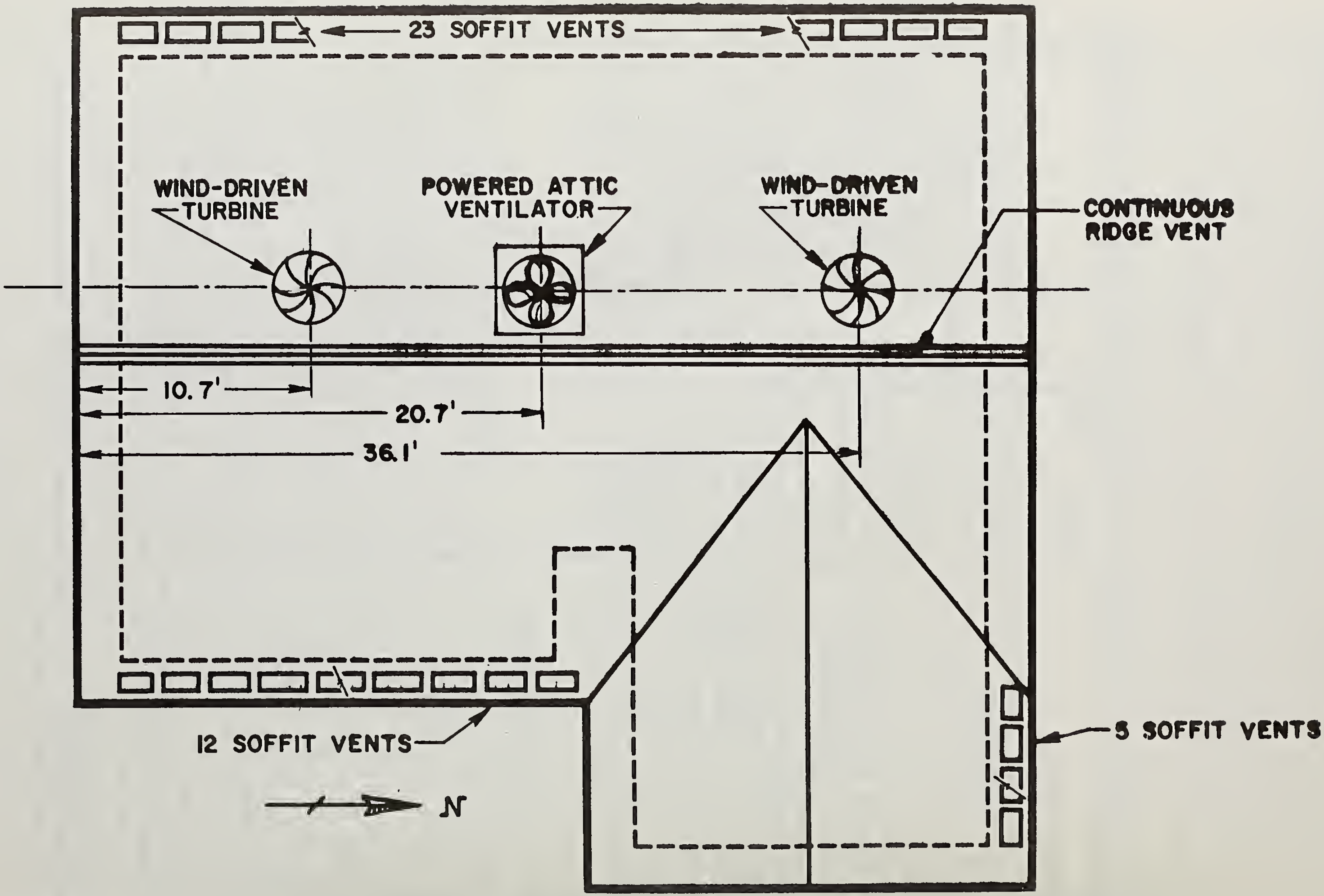


Figure 3. Layout of Attic Ventilation Equipment.

Instrumentation and Measurement Technique

Weather Station

A weather station was mounted on the roof at the north end of the house having 6-1/2 inches (0.17 m) of rockwool ceiling insulation. This weather station contained a rotating-cup anemometer for measuring wind speed and a vane for measuring wind direction. A humidity transducer which produced a millivolt signal proportional to the outdoor relative humidity was mounted inside a louvered enclosure which was part of the weather station.

The total solar radiation and long-wave sky radiation were measured with a pyranometer and a radiometer, respectively, which were mounted on the roof at the base of the weather station. A projecting radiation shield was attached to each house, extending approximately 10 feet (3.1 m) from the roof. A thermocouple for measuring the outdoor air temperature was placed under each radiation shield.

Energy Measurements

The integrated cooling load (q_s) of an air conditioning system during a period P is governed by the relation:

$$q_s = \dot{m} \cdot C_p \cdot \int_0^P \Delta T \cdot dT + W \cdot h_{fg} , \qquad (1)$$

where $\dot{m}$ = mass flow rate of air through the system

C_p = specific heat of moist air

ΔT = temperature difference across A-frame evaporator

T = time

P = period of time over which the blower operates

W = mass of water collected during the period P

h_{fg} = latent heat of vaporization.

The first and second terms represent the sensible and latent portions of the load, respectively.

The temperature drop (ΔT) across the A-frame evaporator was measured with a 36-junction copper-constantan thermopile. The thermopile generated a millivolt signal approximately proportional to the temperature drop (ΔT). This millivolt signal passed through a switching circuit which in turn was fed into an analog integrator, which integrated the temperature drop (ΔT) when the blower of the air conditioner operated.

Water which condensed on the A-frame evaporator was collected in a large metal trash can located on a weighing platform, permitting the mass of water (W) collected each day to be measured.

On each house, the electric energy consumption of the air conditioning system, attic fans, and whole house were measured separately with digitizing watt-hour meters which were read once a day, and with a recording watt-hour meter system* which recorded 15-minute electric energy consumptions on a magnetic tape.

The rate of heat gain through the ceiling of each house was measured with five heat-flow meters sandwiched between the ceiling and the insulation. They were glued to the top of the ceiling at the approximate center of each of the major rooms.

---

* This instrumentation was provided by the Houston Lighting and Power Company of Houston, Texas.

The integrated rate of heat gain (q_d) to the ducts during a period P in the attic space was determined from the relation:

$$q_d = \dot{m} \cdot C_p \cdot \int_0^P \Delta T \cdot dT. \tag{2}$$

The average temperature rise of conditioned air passing through the ducts was sensed with a 96-junction thermopile. The millivolt signal from this thermopile was fed through a switching circuit into an analog integrator, which integrated the temperature rise (ΔT) when the blower of the air conditioner operated.

Temperature and Humidity Measurements

Surface and ambient air temperatures were measured using 24-gage copper-constantan thermocouples. The temperatures of the indoor air, underside of the ceiling, the top of the ceiling insulation, and the attic air 1 foot (0.31 m) above the ceiling insulation were measured with thermocouples along vertical measuring stations in the approximate centers of the major rooms of the house (see Figure 2). The indoor air temperature thermocouples were suspended from the ceiling and placed 4.5 ft (1.4 m) above the floor. In the kitchen, thermocouples were also placed 2 inches (.051 m) below the ceiling and 2 inches (.051 m) above the floor. Duct tape was used to attach thermocouples to the underside of the ceiling. Umbrella-shaped aluminum foil radiation shields were placed over the attic-air-temperature thermocouples so that they would not sense the temperature of the hot roof above. Attic air temperatures were also measured with shielded thermocouples placed at 1.0 (0.31), 2.5 (0.76), and 4.0 ft (1.2 m) above the attic floor at three locations along the attic center-line. Temperatures of the underside of the roof sheathing were measured at six locations.

The indoor relative humidity was measured adjacent to the living room thermocouple, with a humidity transducer of the same type used in the weather station.

Transducer signals were fed into a data acquisition system which recorded parameter values on paper tape at hourly intervals.

Experimental Plan

Commencing on 18 July, 1977, the testing schedule outlined in Table 1 was carried out on the three houses. The first column of the table gives the test period, while columns 2 through 4 give the attic ventilation system that was employed on Houses 1, 2, and 3, respectively. In most instances the test period lasted one week, with the exception of test period 9, which lasted ten days. In Table 1, S stands for soffit venting, T for turbine venting, P for power venting, CO for attic closed off, WHF for whole-house ventilation, RDV for ridge venting, and ROV for roof venting.

Prior to starting the testing schedule of Table 1, an attempt was made to equalize the indoor temperatures of the houses. During the first two test periods of Table 1, the average indoor temperatures were 73.8 (23.2), 75.7 (24.3), and 76.5°F (24.7°C) for Houses 1, 2, and 3, respectively. On the second day of test period 3, adjustments in indoor thermostat settings were performed in order to equalize the indoor temperature in the three houses. The average indoor temperatures, after adjustment, were 76.7 (24.8), 76.6 (24.8) and 76.5°F (24.7°C) for Houses 1, 2, and 3, respectively.

On the first day of each test period (Monday), the ventilation systems outlined in Table 1 were activated in the houses. Other ventilation openings not a part of the particular ventilation system were sealed with duct tape. Measurements were carried out during the remaining days of the test period.

The heat release of lighting, equipment, and occupants was simulated with an equipment load of 150 watts and a constant lighting load of 1140 watts. Lamps were distributed throughout the houses. The total internal heat-release rate was 1290 watts, or 1.3 watts per square foot (14 W/m^2) of living space. The amount of lighting load was selected so that the daily-average internal heat release rate for the present test houses would be identical to that for the four-bedroom wood-frame townhouse reported in Ref [7]. The houses were unoccupied during the test, except for the activities of technical personnel

TABLE 1. TESTING SCHEDULE

Test Period	House 1 6-1/2-in.* (0.17 m) Ins.	House 2 4- in.* (0.10 m) Ins.	House 3 4-in.* (0.10 m) Ins.
1	S	S	S
2	S	S	T
3	P	P	T
4	S**	P	S
5	T	T/S***	S
6	S	T	P
7	S	S	P
8	S	S	S
9	S	CO	P
10	WHF	RDV	S
11	WHF	ROV	S

* Ceiling insulation thickness.

** The indoor air temperature was elevated to 77.9°F (25.5°C) instead of 76.7°F (24.8°C).

*** Turbine venting was implemented during the first four days of the test period and soffit venting during the remaining two days.

conducting measurements. Whenever technical personnel were inside one of the houses, the lighting load was reduced 100 W per person.

During the attic ventilation tests, draperies in front of all windows and the sliding glass door were maintained in a fully closed position.

Whole-House-Fan Test Results

The energy consumption of the air conditioner of House 1 is plotted as a function of daily-average outdoor temperature in Figure 4 for the two whole-house fan test periods. These data show that when the daily-average temperature was below 75°F (23.9°C), the whole-house fan was able to satisfy all the cooling requirements for the house; the air conditioner did not have to operate.

On days when the daily-average temperature was above 75°F (24°C), the whole-house fan was substituted for the air conditioner whenever the outdoor temperature was less than 82°F (28°C). The daily cooling energy requirement for the whole-house-fan test days (test periods 10 and 11 for House 1) is plotted as a function of daily-average outdoor air temperature in Figure 5. The solid line correlates the cooling energy requirement during periods when the house was cooled entirely with the air conditioner and soffit venting was employed. The broken line correlates the cooling energy requirement when the whole-house fan was used in conjunction with air conditioning. It includes both the energy requirement for the air conditioner and the whole-house fan. The cooling energy requirement is seen to be substantially reduced when whole-house ventilation was utilized.

Based on the cooling energy correlations presented in Figure 4, reductions in cooling energy requirement were calculated. These results are presented in Table 2. It can be seen that the percent reduction in cooling energy requirement increased progressively as the daily-average outdoor temperature was reduced in the range of daily-average temperatures down to 74°F (23.3°C). The absolute reductions in cooling energy requirement also became greater as the daily-average outdoor temperature decreased for the same range.

An analysis was performed of the indoor comfort conditions (using Fanger's comfort model [8]), during periods when the whole-house fan operated. For the analysis, the indoor air velocity was assumed to be 8 ft/min (0.04 m/s) when the air conditioner was operated and 40 ft/min (0.2 m/s) when the whole-house fan was operated. The analysis showed that sedentary occupants would experience comfortable indoor conditions when the outdoor temperature

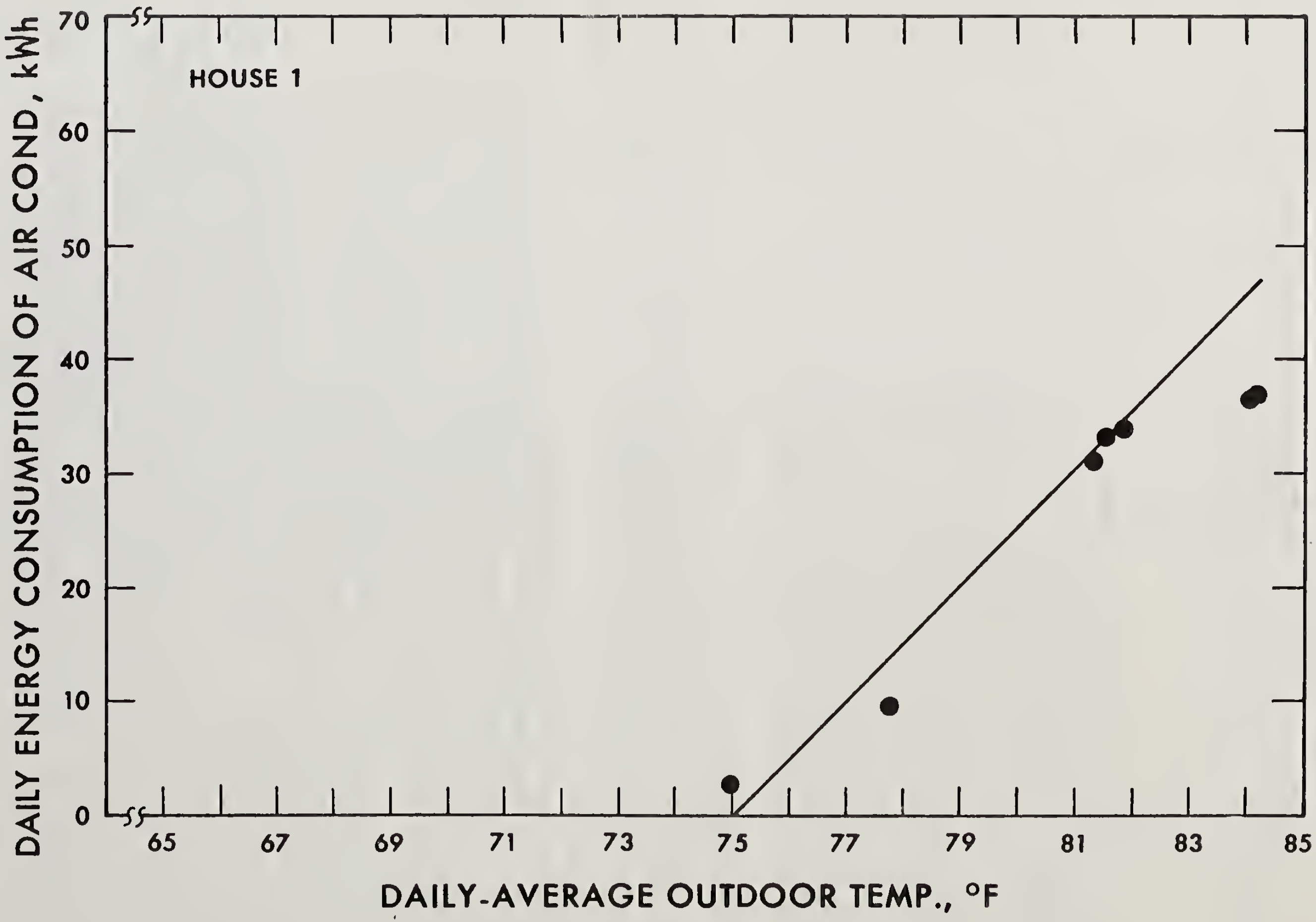


Figure 4. Energy Consumption of Air Conditioner of House 1 Plotted as a Function of Daily-Average Outdoor Temperature during Whole-House Ventilation.

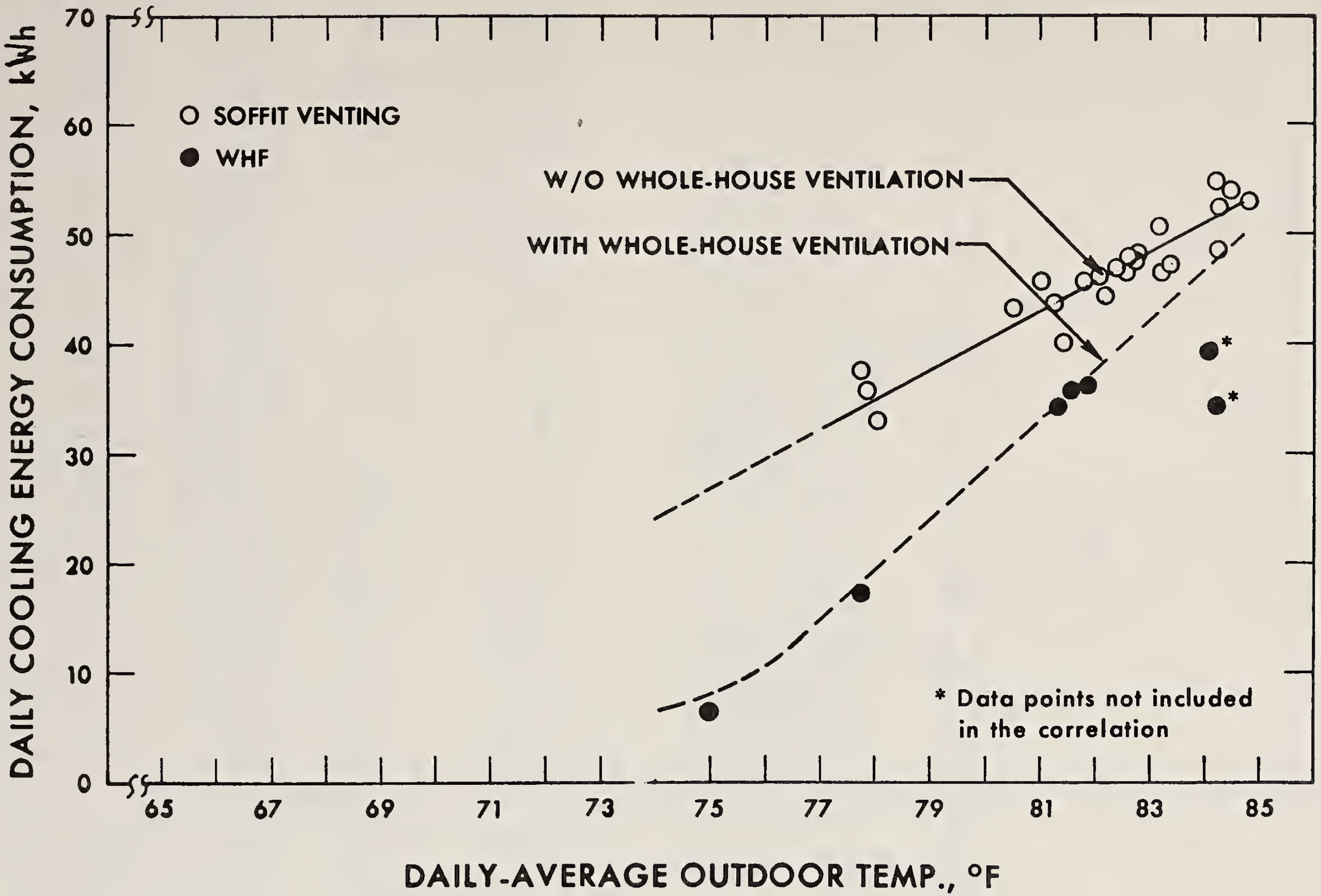


Figure 5. Daily Cooling Energy Requirement for House 1 Plotted as a Function of Daily-Average Outdoor Temperature for Whole-House-Fan Test Days.

was 82°F (28°C) and the outdoor relative humidity was as high as 75%. However, occupants performing moderate work would experience comfort levels ranging from warm to slightly warm.

TABLE 2. REDUCTIONS IN COOLING ENERGY CONSUMPTION ACHIEVED BY WHOLE-HOUSE VENTILATION

Daily-Average Outdoor Temperature °F (°C)	Reduction in Cooling Energy Requirement kWh (10^7•J)	%*
76 (24)	19.3 (6.95)	65.6
78 (26)	15.7 (5.65)	45.1
80 (27)	12.0 (4.32)	29.9
82 (28)	8.5 (3.06)	18.7
84 (29)	4.9 (1.76)	9.6

* This percentage is the difference in the energy requirement divided by the air conditioner energy requirement when the house is cooled entirely with the central air conditioner.

Although limited data are available and additional evaluation of whole-house ventilation is needed to refine energy conserving contributions, it is evident that energy savings resulting from the use of whole-house ventilation instead of central air conditioning will be substantial. The average monthly temperatures for July and August for the northern half of the United States are less than 75°F (24°C). This gives a general indication that whole-house ventilation may be used instead of air conditioning to provide indoor comfort during a major portion of the summer cooling season in the northern half of the United States. Since a whole-house fan consumes approximately one tenth of the energy of an operating air conditioner, the energy savings will be considerable.

Attic Ventilation Test Results

In order to determine the effect of attic ventilation on the attic air temperatures and the ceiling and duct heat-gain rates, it was necessary to devise a scheme to restrict the effects of divergent weather conditions. To reduce the scatter due to variations in solar radiation and wind speed, only days having the following characteristics were included in the analysis:

- ° sunny conditions (daily total solar radiation greater than 1800 Btu/ft^2 (2.044x10^4 KJ/m^2));
- ° wind velocities at maximum-load condition between 4.6 (2.1) and 8.0 mph (3.6 m/s); and
- ° maximum outdoor temperatures $\geq$90°F (32°C).

Maximum observed hourly attic air temperatures are plotted as a function of maximum outdoor temperature for Houses 2 and 3 in Figures 6 and 7, respectively. The attic air temperatures were obtained by averaging the eight attic air temperature thermocouples at the 1-foot level. From these figures, it is seen that the addition of power venting to soffit venting reduced the attic air temperature in both houses approximately 10°F (5.6°C) at an outdoor temperature of 95°F (35°C).

The data for House 2 (Figure 6) show turbine and ridge venting to be equally as effective as power venting in reducing attic air temperature. However, for House 3 (Figure 7), turbine venting was less effective than power venting but more effective than soffit venting.

Similar correlations for maximum observed hourly ceiling heat-gain rates for Houses 2 and 3 are given in Figures 8 and 9, respectively. The ceiling heat-gain rates in each house were obtained by averaging the responses of the five ceiling heat-flow meters. At an outdoor

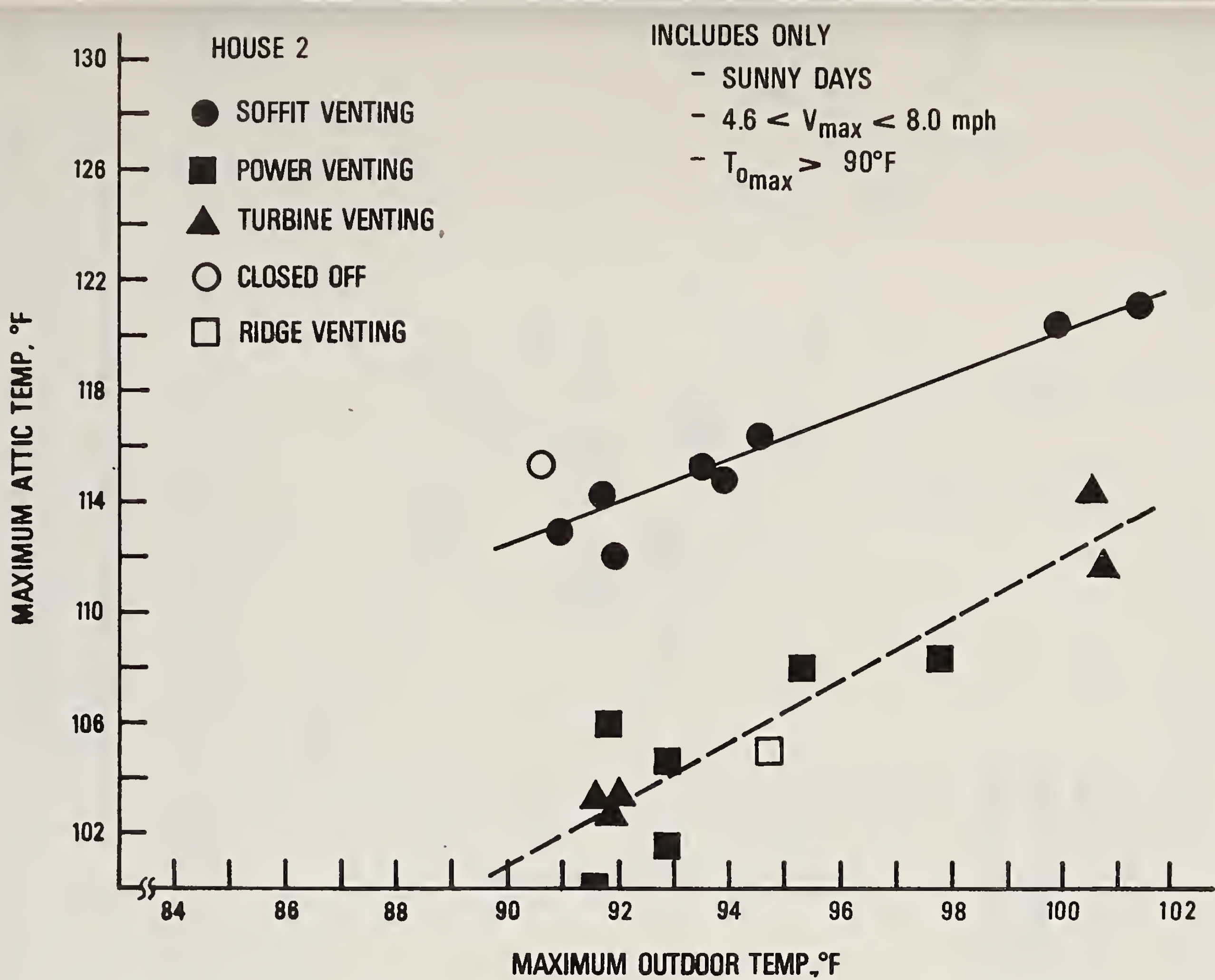


Figure 6. The Effect of Attic Ventilation on Maximum Attic Air Temperatures for House 2.

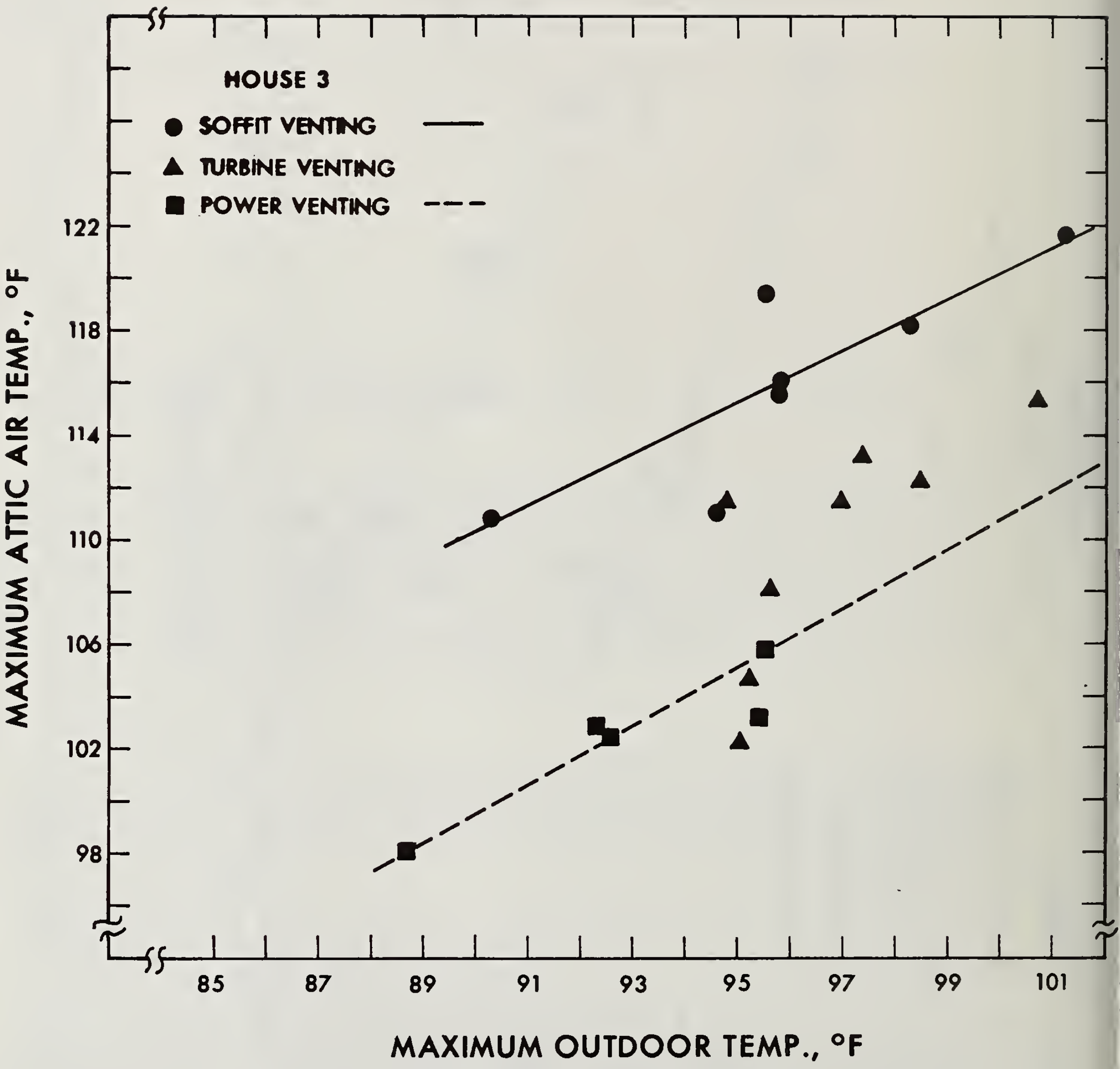


Figure 7. The Effect of Attic Ventilation on Maximum Attic Air Temperatures for House 3.

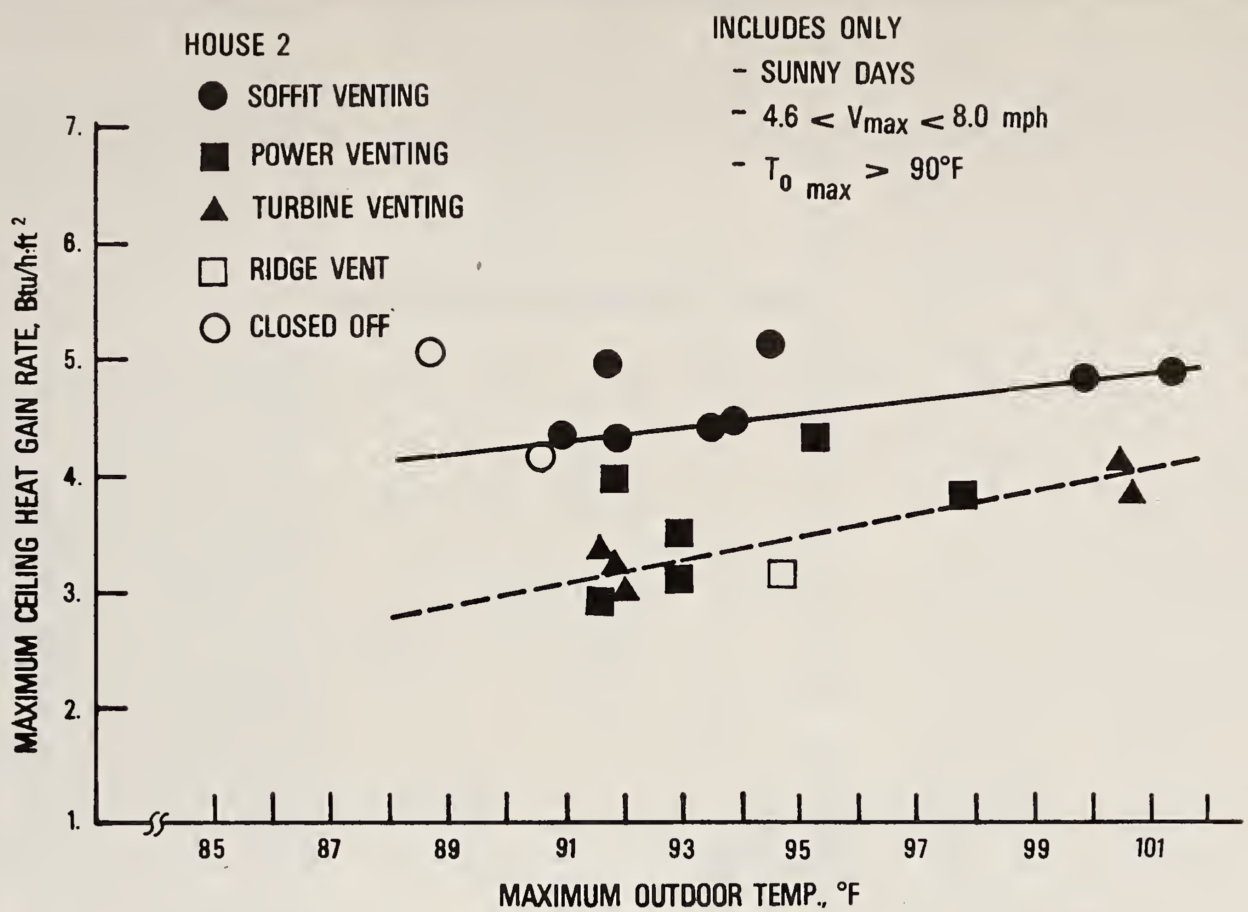


Figure 8. The Effect of Attic Ventilation on Maximum Rate of Ceiling Heat Gain for House 2.

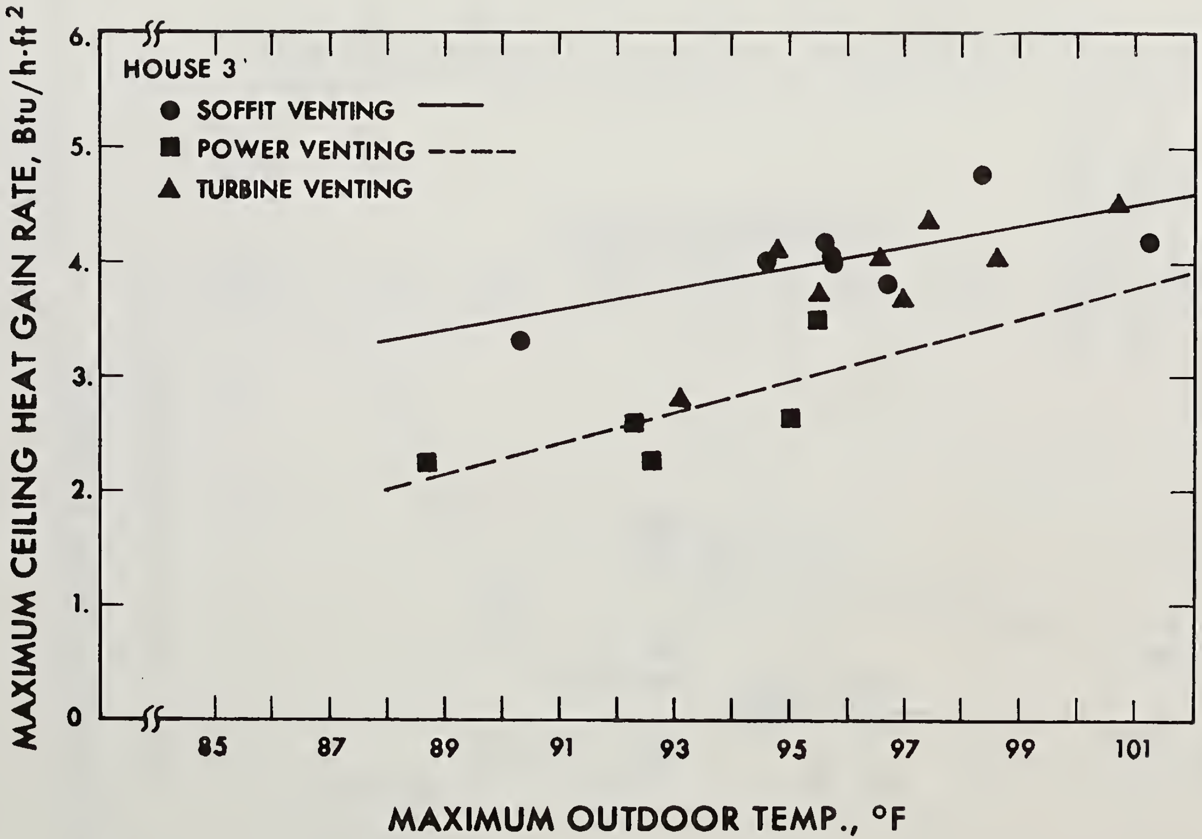


Figure 9. Effect of Attic Ventilation on Maximum Rate of Ceiling Heat Gain for House 3.

temperature of 95°F (35°C), the addition of power venting to soffit venting reduced the ceiling heat-gain rate 23 and 25% for Houses 2 and 3, respectively.

A similar correlation for maximum duct heat-gain rates for House 3 is given in Figure 10. Duct heat-gain rates were not available for House 2 due to malfunctioning equipment. At an outdoor temperature of 95°F (35°C) the addition of power venting to soffit venting reduced the duct heat-gain rate by 16%.

The components of the cooling load for House 3 with soffit venting at the hottest time of the day are given in Table 3. Heat gain through the ceiling and into the air conditioning ducts represented 17.1 and 9%, respectively, of the total cooling load. The effect of a 25% reduction in the ceiling heat-gain rate on the total cooling load is determined by taking 25% of 17.1% (the fraction of total cooling load due to ceiling heat gain). Thus, a 25% reduction in the ceiling heat-gain rate represents a 4.3% reduction in the total cooling load. In a similar fashion, a 17% reduction in duct heat-gain rate as observed for House 3 is found to represent a 1.4% reduction in the total cooling load. Therefore, a net reduction in the total cooling load of 5.7% was achieved at maximum load condition by adding power venting to soffit venting.

For these particular houses, the air conditioners were undersized and ran continuously for several hours during the hot part of the day. Therefore, no measured reductions in maximum air conditioner energy consumption were observed and the room temperature rose slightly as the outdoor temperature rose. If the air conditioners had been properly sized instead of undersized, then a reduction in electrical energy consumption should have occurred at maximum-load condition.

TABLE 3. COMPONENTS OF TOTAL COOLING LOAD AT MAXIMUM-LOAD CONDITION

COMPONENT	Btu/h	(W)	%
Latent Load	5,990	(1,760)	26.4
Sensible Load	16,686	(4,890)	
• Ceiling Heat Gain	3,888	(1,140)	17.1
• Duct Heat Gain	2,041	(598)	9.0
• Internal Heat Release	4,379	(1,280)	19.3
• Air Infiltration	1,506	(441)	6.6
• Other	4,872	(1,427)	21.5
TOTAL	22,676	(6,640)	100.0

In the case of power venting on House 3 at maximum load condition, the power consumed by the power ventilator was found to offset calculated reductions in the power consumption for an oversized air conditioner resulting from reduced ceiling and duct heat-gain rates. The ceiling heat-gain rate was reduced 1.1 Btu/h•ft^2 (3.47 W/m^2) (or a net reduction of 1120 Btu/h (328 W) for a ceiling area of 1020 ft^2 (94.9 m^2)), and the duct heat-gain rate was reduced 320 Btu/h (93.9 W). A total reduction in the cooling load of 1440 Btu/h (422 W) occurred. At maximum-load condition, the coefficient of performance of the air conditioner was found to vary between 1.4 and 1.6. Using an average value of 1.5, a reduction in the cooling load of 1440 Btu/h (422 W) would have produced a reduction of 281 W in the power consumption of the air conditioner, provided that the air conditioner had sufficient capacity to satisfy the cooling load at maximum-load condition. However, the energy consumption for the attic fan was 284 W.

The undersizing of the air conditioners had a small effect on the maximum-load ceiling heat-gain rates, because the temperature difference across the ceiling was not significantly changed due to the undersizing of the air conditioner. At the highest load condition, the temperature difference across the ceiling was only decreased 1.5°F (0.83°C) out of a temperature difference of 37°F (21°C). If it is assumed that the ceiling heat-gain rate is proportional to the temperature difference, such a decrease in temperature

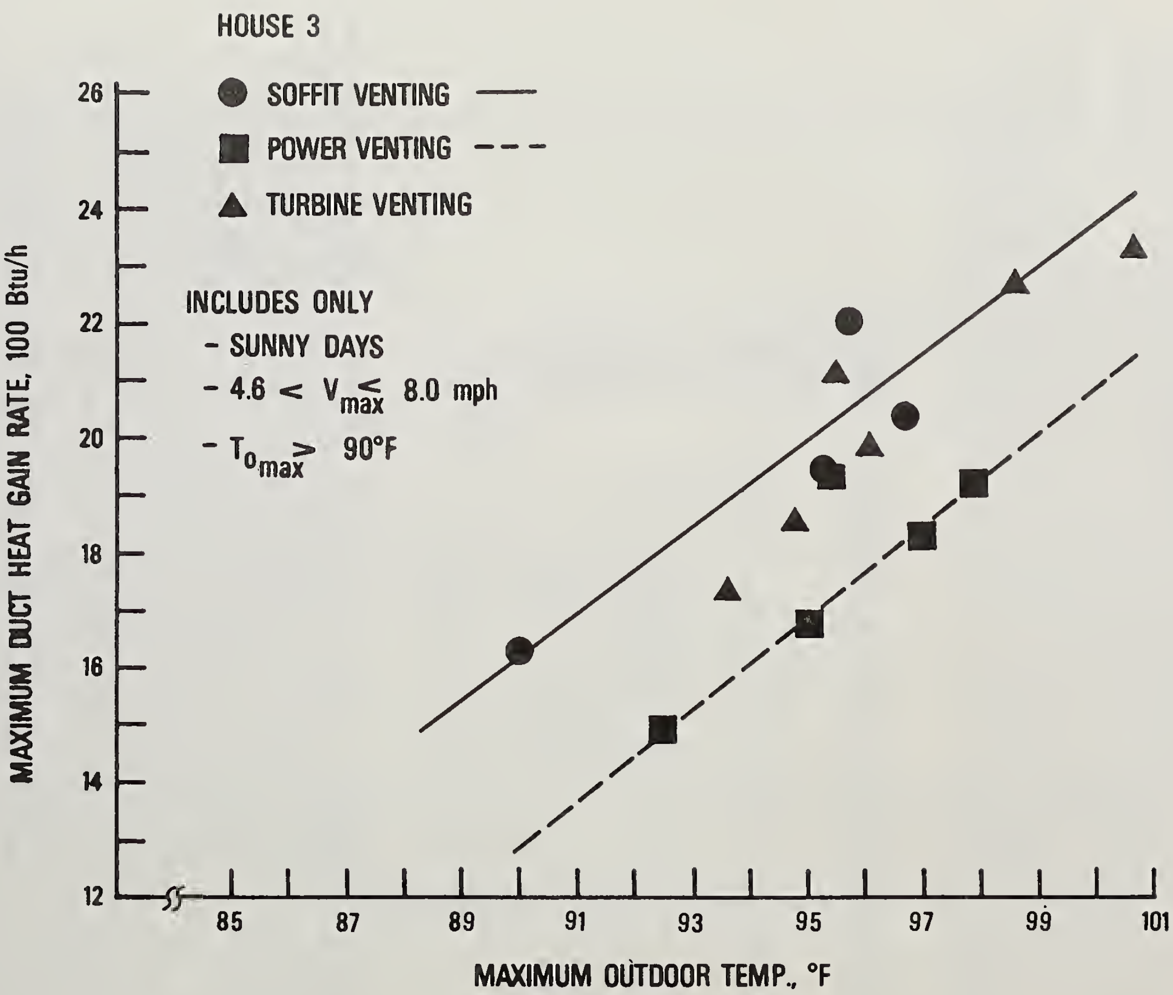


Figure 10. The Effect of Attic Ventilation on the Maximum Rate of Duct Heat Gain for House 3.

difference would produce only a 4% reduction in the ceiling heat-gain rate. However, most of the measured ceiling heat-gain rates should be unaffected, due to the fact that the temperature difference across the ceiling was not affected by the undersizing of the air conditioner.

Daily-average attic air temperatures at the 1-foot (0.30 m) level are plotted as a function of daily-average outdoor temperature for Houses 2 and 3 in Figures 11 and 12, respectively. At a daily-average outdoor temperature of 83°F (28°C), the average attic temperature is approximately 4.0°F (2.2°C) above the average outdoor temperature. The various attic ventilation systems compared to soffit venting usually reduced the average attic air temperature by less than 2.0°F (1.1°C). A perfect ventilation system, one which would reduce the attic temperature to the outdoor temperature at all times, would produce only a 4.0°F (2.2°C) reduction.

Similar correlations for the daily-average ceiling heat-gain rates for Houses 2 and 3 are given in Figures 13 and 14, respectively. An analysis of the data of Figures 13 and 14 shows that the various ventilation systems compared to soffit venting never reduced the daily-average ceiling heat-gain rate by more than 19% (maximum deviation below the soffit venting correlation line). Usually the reductions were observed to be considerably less than 19%.

A similar correlation for the daily-average duct heat-gain rates is given in Figure 15. An analysis of the data of Figure 15 shows that power or turbine venting as compared to soffit venting produced a maximum reduction of 17% (maximum deviation below the soffit venting correlation line) in the average duct heat-gain rate.

Using the correlations presented in Figures 14, 15, 16, it can be determined that at a daily-average outdoor temperature of 83°F (28°C) the ceiling and duct heat-gain rates represent only 10.8 and 6.5%, respectively, of the daily cooling load. Taking the reductions in ceiling heat-gain rate to be the maximum observed reduction of 19%, it is calculated that this reduction represents 2% of the total cooling load. Taking the reduction in duct heat-gain rate to be the maximum observed reduction of 17%, then we see that this reduction represents 1.1% of the total cooling load. Thus, the maximum observed reduction in daily cooling load for these houses under the conditions of test was only 3.1%. The reductions were usually observed to be considerably less than the 3.1% figure, since reductions in typically observed ceiling and duct heat gains were usually found to be considerably less than the maximum observed values.

Daily sensible and latent cooling loads for House 3 are plotted as a function of daily-average outdoor temperature in Figure 16. A similar plot for the daily air conditioner electric energy consumption for House 3 is given in Figure 17. In each of the figures, the solid lines correlate energy data for the various attic ventilation systems. From these figures, it is seen that the addition of power or turbine venting to soffit venting did not produce consistent reductions in the sensible load, latent load, or air conditioner energy consumption. Similar results were obtained for Houses 1 and 2. These results are consistent with the previous analysis of daily-average ceiling and duct heat-gain rates, which showed that reductions in these parameters were not sufficient to impact the cooling load significantly.

The effect of attic ventilation can also be determined by comparing the air conditioner energy consumption for Houses 2 and 3 (which were constructed to be identical except for differences in the attic ventilation system for the eleven consecutive periods of test). Both houses were exposed to identical weather conditions. Observed differences should reflect differences in the performance caused by the different attic ventilation systems used in the two houses during the various periods of test.

The results of this comparison are summarized in Table 4. For test periods 1 and 3 through 8, the air conditioner of House 3 consumed 6 to 9% more energy than the air conditioner for House 2, regardless of the type of attic ventilation. These data indicate that no systematic difference in the energy consumption of the air conditioning equipment occurred as a result of the type of attic ventilation. The fact that House 3 consumed more energy for space cooling than House 2 was attributed to factors other than attic ventilation. A factor causing the observed difference in air conditioner energy consumption was the fact that at maximum-load condition, when both units operated continuously, the air conditioner for House 3 consumed 6 to 9% more than the air conditioner for House 2. A comparison of the air conditioner energy consumption for Houses 2 and 3 for test periods 2, and 9-11, indicates small differences in the air conditioner energy consumption pattern for the houses. However, an analysis of ceiling and duct heat-gain rates for these test periods indicates that the observed differences in air conditioner energy consumption pattern could not be attributed to changes in ceiling and duct heat-gain rates.

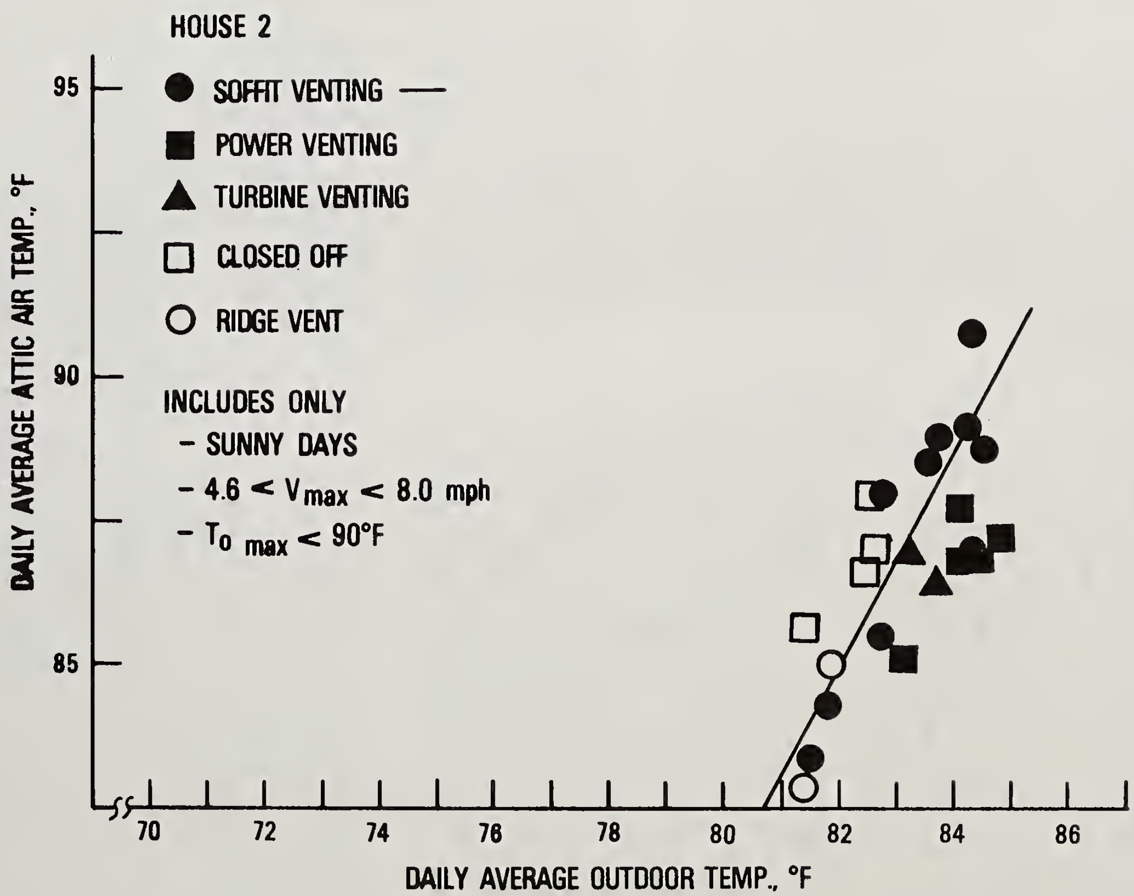


Figure 11. Daily-Average Attic Air Temperatures for House 2.

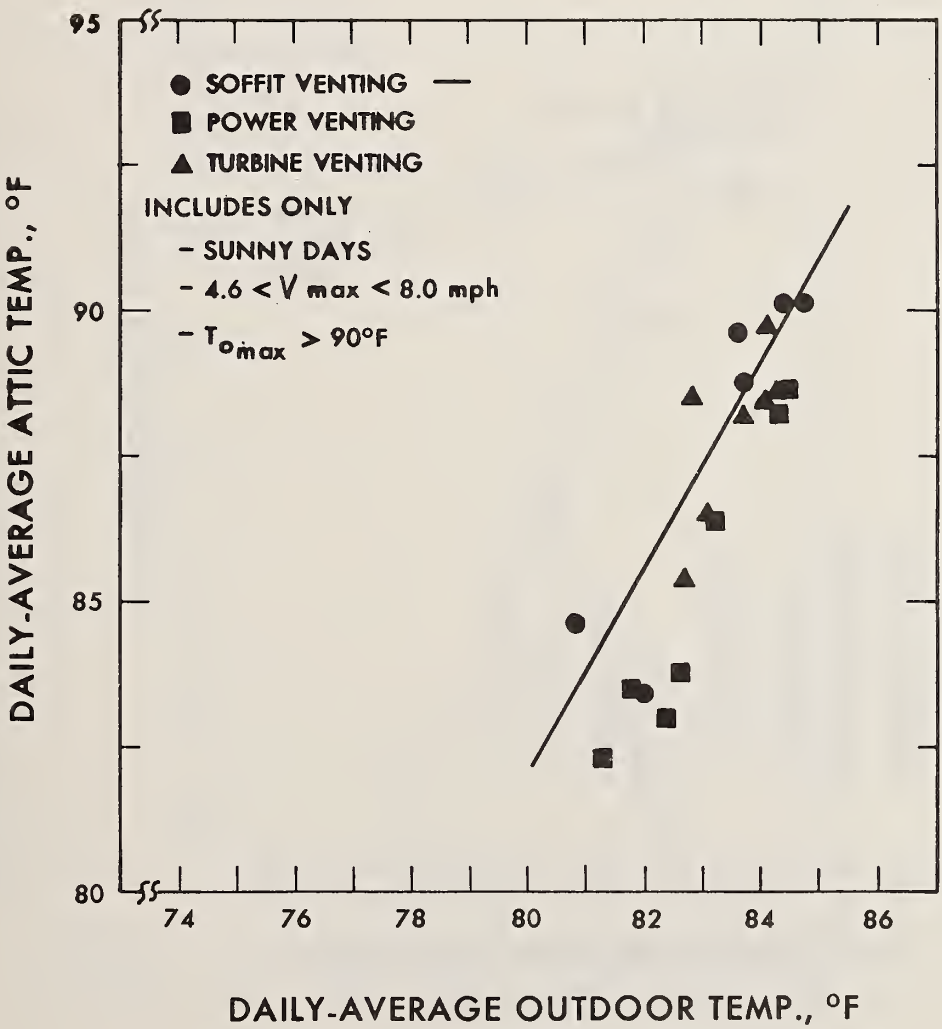


Figure 12. Daily-Average Attic Air Temperatures for House 3.

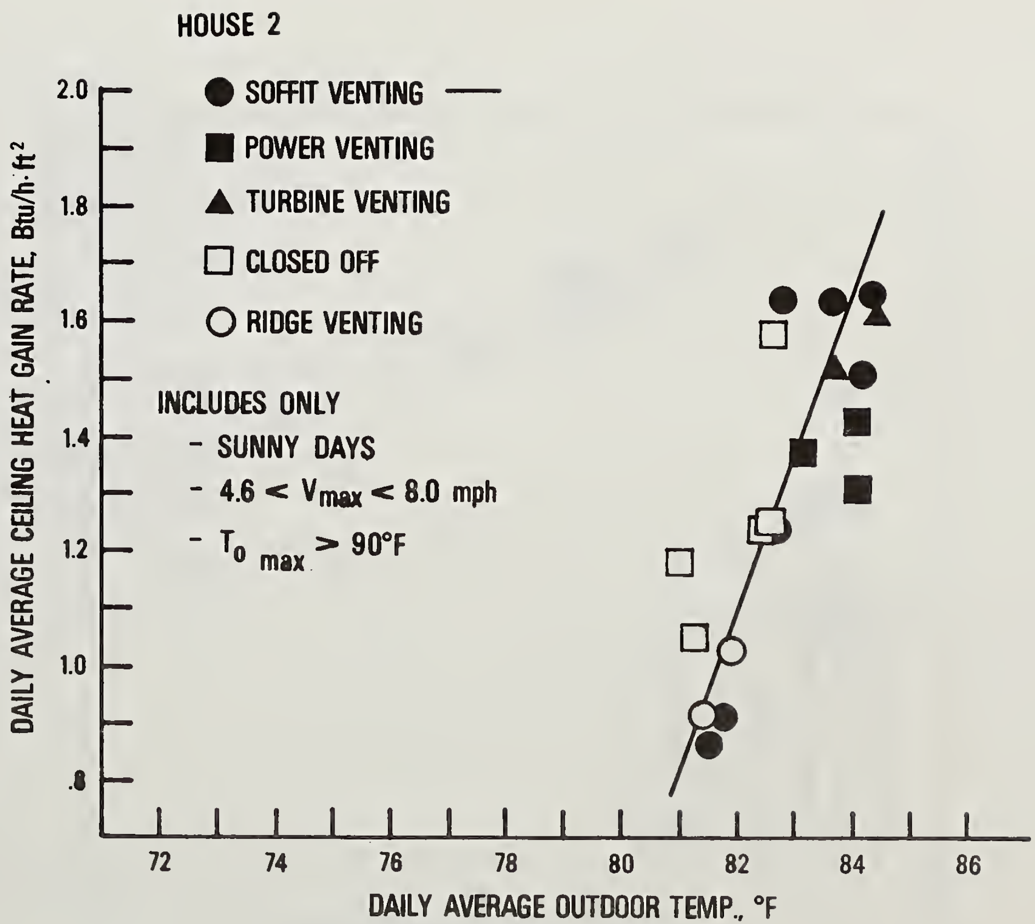


Figure 13. Daily-Average Ceiling Heat-Gain Rates for House 2.

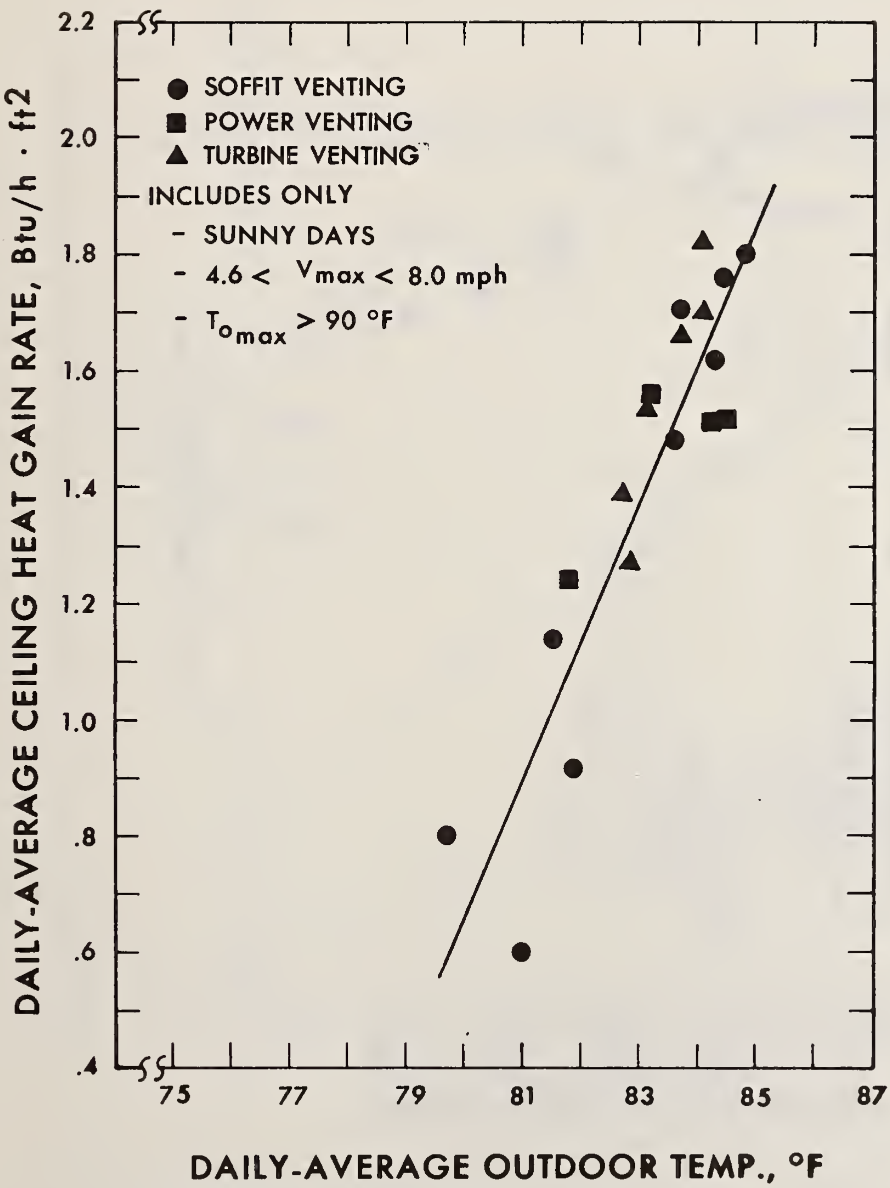


Figure 14. Daily-Average Ceiling Heat-Gain Rates for House 3.

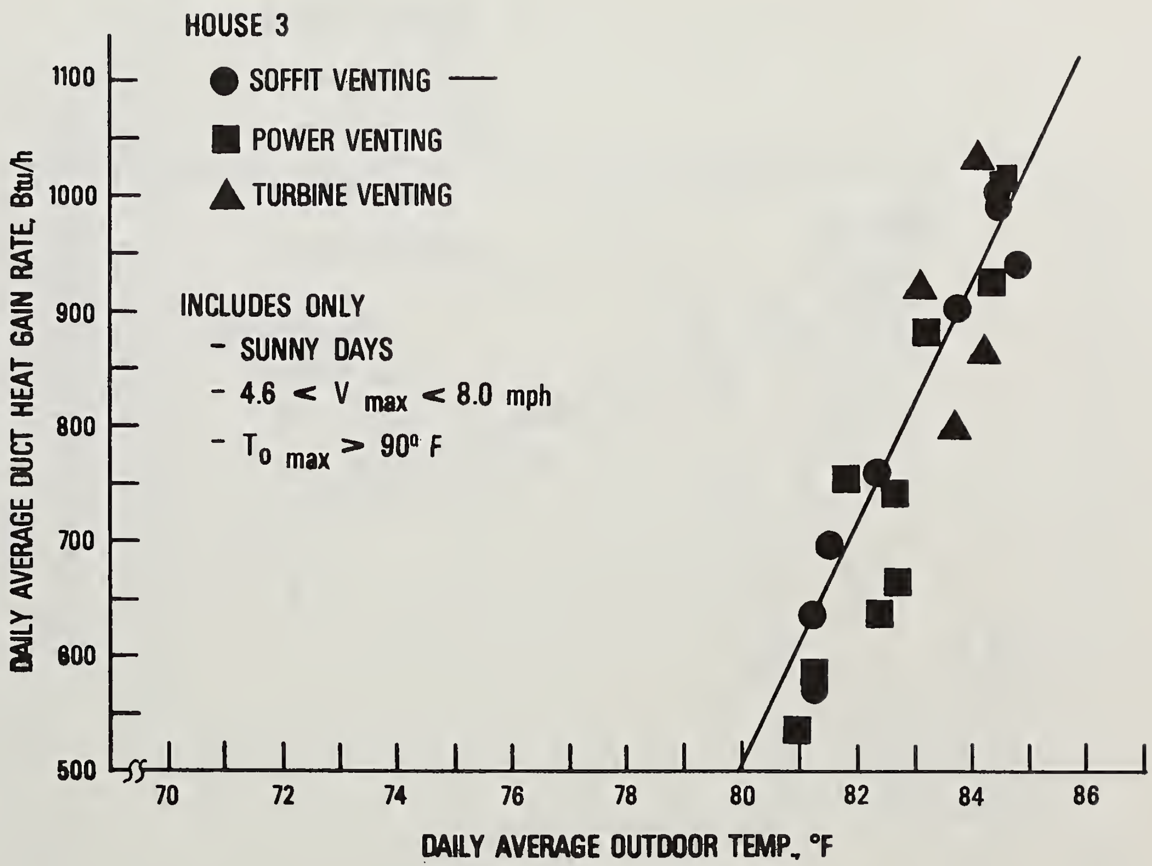


Figure 15. Daily-Average Duct-Heat-Gain Rates for House 3.

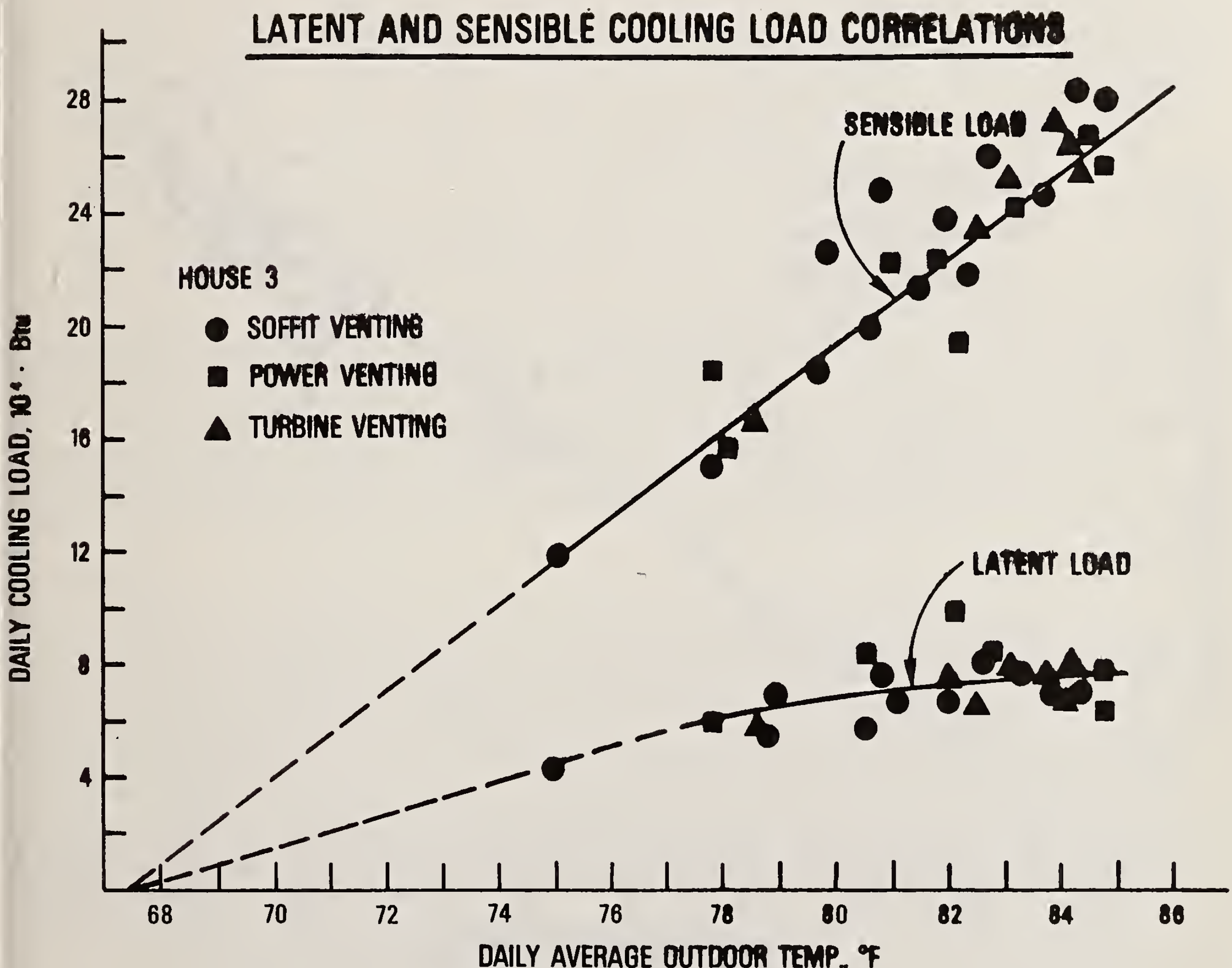


Figure 16. Daily Sensible and Latent Cooling Loads for House 3.

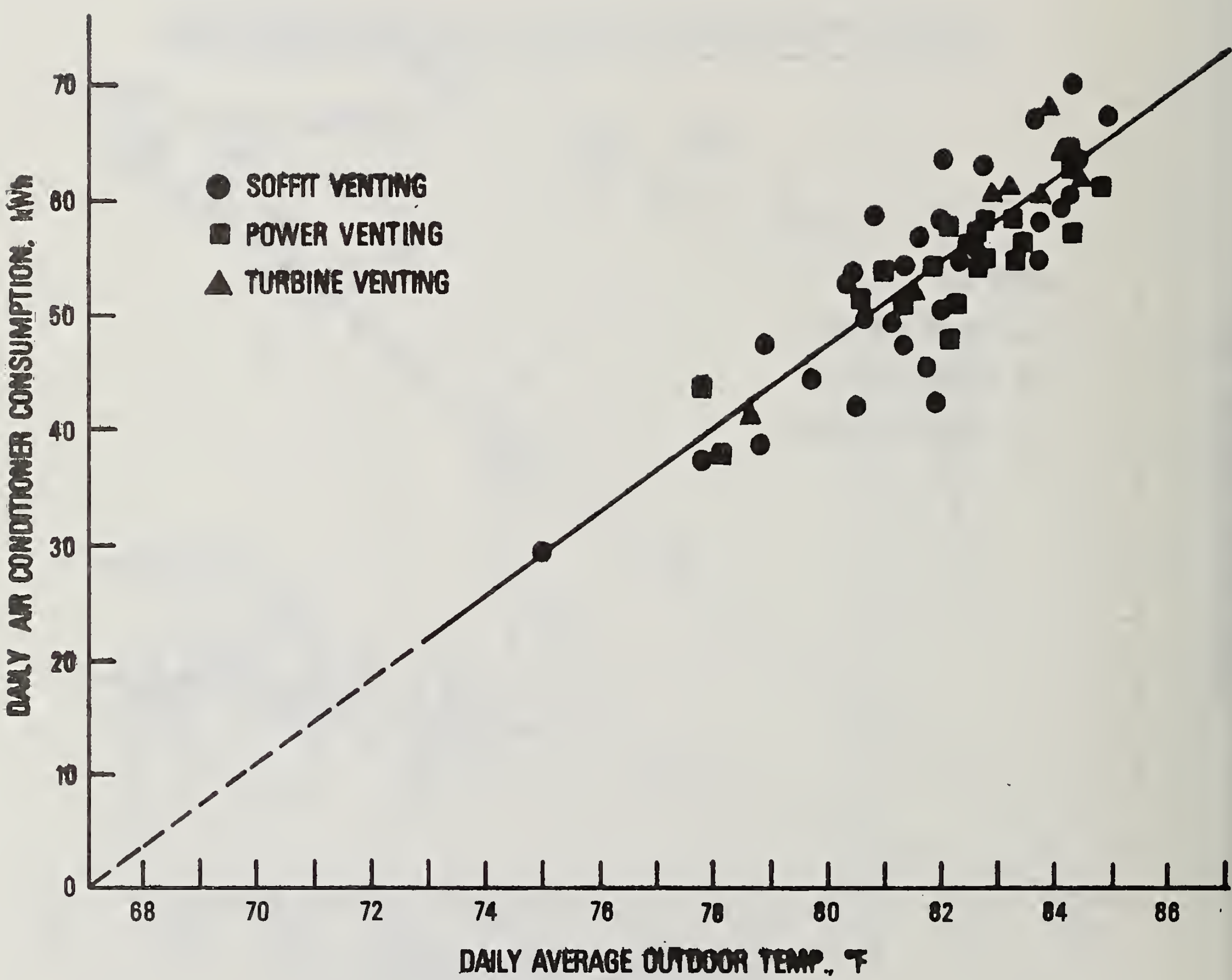


Figure 17. Daily-Total Air Conditioner Energy Consumption Plotted as a Function of Daily-Average Outdoor Temperature for House 3.

TABLE 4. COMPARISON OF AIR CONDITIONER ENERGY CONSUMPTION FOR HOUSES 2 AND 3

Test Period	Daily Average Outdoor Temp °F	(°C)	Daily Solar Radiation Btu/ft^2	(10^4 kJ/m^2)	Average Wind Velocity mi/h	(m/s)	House 2 Mode	kWh	(10^8•J)	House 3 Mode	kWh	(10^8•J)	Diff %
1	82.3	(27.9)	1860	(2.11)	5.1	2.3	S	57.9	(2.08)	S	61.5	(2.21)	6.2
2	83.7	(28.7)	1910	(2.17)	5.0	2.2	S	63.0	(2.27)	T	61.0	(2.20)	-3.2
3	83.0	(28.3)	1800	(2.04)	4.9	2.2	P	55.4	(1.99)	T	59.4	(2.14)	7.2
4	83.0	(28.3)	2000	(2.27)	4.3	1.9	P	51.8	(1.86)	S	56.3	(2.03)	8.7
5	82.5	(28.1)	1280	(1.73)	3.6	1.6	T*	49.2	(1.77)	S	52.6	(1.89)	6.7
6	83.9	(28.8)	1830	(2.08)	5.0	2.2	T	64.6	(2.33)	P	69.5	(2.50)	7.6
7	82.6	(28.1)	1770	(2.01)	5.0	2.2	S	53.1	(1.91)	P	56.9	(2.05)	7.2
8	82.7	(28.2)	1080	(1.23)	4.1	1.8	S	39.7	(1.43)	S	43.0	(1.55)	8.3
9	80.9	(27.2)	1700	(1.93)	4.8	2.1	CO	51.4	(1.85)	P	51.0	(1.84)	- .7
10	82.4	(28.0)	1550	(1.76)	3.9	1.7	RDV	54.5	(1.96)	S	56.3	(2.03)	3.4
11	72.9	(22.7)	917	(1.04)	2.3	1.0	ROV	32.3	(1.16)	S	32.7	(1.18)	1.1

* Last two days of this test period,for which the attic was ventilated with soffit venting, were excluded for this portion of the analysis.

An analysis of ceiling temperatures of the test houses showed that the various attic ventilation systems compared to soffit venting produced less than a 1°F (0.6°C) reduction in the ceiling temperature at maximum-load condition. Such reductions would be expected to have very little effect on the indoor-comfort condition. For instance, a person standing in a cube-shaped room will receive thermal radiation equally from the six surfaces of the room. The radiation received from the ceiling is 1/6 of the total amount received. A reduction in ceiling temperature of 1°F (0.6°C) would represent approximately a 1/6°F (0.09°C) reduction in mean radiant temperature, which is insignificant from a comfort standpoint.

Conclusions

Whole-house ventilation was shown to be an effective energy conservation procedure for saving air conditioning energy. When the daily-average outdoor temperature was less than 75°F (24°C), the whole-house fan provided all the cooling requirements for the test house. The average monthly temperatures for July and August for the northern half of the United States are less than 75°F (23.9°C). This provides a general indication that whole-house ventilation may be used instead of air conditioning to provide indoor comfort during a significant portion of the summer cooling season in the northern half of the United States. Since a whole-house fan consumes considerably less energy than a central air conditioner, the energy savings may be expected to be considerable. On days when the daily-average temperature was above 75°F (24°F), the whole-house fan was used instead of the air conditioner whenever the outdoor temperature fell below 82°F (28°C). Cooling energy savings ranging from 10 to 66% were shown to occur. The percent savings decreased as the daily-average outdoor temperature increased.

It was determined that the use of a whole-house fan when the outdoor temperature was 82°F (28°C) and the outdoor humidity was as high as 75% would produce comfortable indoor conditions for sedentary occupants exposed to the increased air movement induced by the whole-house fan. However, it was determined that occupants performing moderate work would experience comfort levels ranging from warm to slightly warm.

A comparison of the various attic ventilation systems to soffit venting showed that attic ventilation was not an effective energy conservation procedure for these houses which had ceiling insulation of thicknesses 4(10) and 6.5 inches (17 cm). At maximum-load condition, increased attic ventilation as compared to soffit venting produced a 25% and 16% reduction in the ceiling and duct heat-gain rates, respectively. However, the cooling load was only reduced 6%, due to the fact that these heat gains represented small fractions of the cooling load. In the case of using power venting at maximum-load condition, the reduction in energy consumption of a properly sized air conditioner was calculated to have been offset by the energy consumption of the power vent. When the effect of reduced ceiling and duct heat gains was considered over a period of a day, attic ventilation was found to produce less than a 3% reduction in the daily cooling loads for the test houses.

Under the test conditions, attic ventilation for these houses was shown to have an insignificant effect on the indoor-comfort condition. Measured reductions in ceiling temperatures achieved by the various attic ventilation systems as compared to soffit venting were shown to be less than 1°F (0.6°C) which would not change the mean-radiant temperature of the living space sufficiently to impact the indoor-comfort condition.

Acknowledgments

The authors are grateful to the American Ventilation Association and the Home Ventilating Institute for their many ideas, conceptual approach, and technical suggestions prior to and during the experimental phases of the study. The authors are especially grateful to the American Ventilation Association for providing the three test houses and most of the ventilation equipment used for the study. Appreciation is also expressed to Richard Mathews, Dan Abbott, Mark Lemay, and John Bean for their technical assistance in the installation of instrumentation, day-to-day operation in the experiments, and in carrying out a portion of the tedious task of reducing and analyzing the collected data. Special thanks is given to Mary Reppert for editing the manuscript.

References

1. J.M. Russ, B.G. Haynie, and F.B. Clark, "Analysis of Attic Ventilation Test," Arkansas Power and Light Company, Little Rock, Arkansas 1976.

2. L.W. Nelson and J.R. Tobias, "Energy Savings in Residential Buildings," ASHRAE Journal, Vol. 16, No. 2, Feb. 1974.

3. C.F. Sepsy, B.L. Moentenich, and M.F. McBride, "A Study of Attic Temperatures and Heat Loss in Residential Homes," Engineering Experiment Station, The Ohio State University, Columbus, Ohio, June, 1975.

4. J.R. Wright, D.R. Bahnfleth, and E.J. Brown, "Comparative Performance of Year-Around Systems Used in Air Conditioning Research Residence No. 2," University of Illinois Engineering Experiment Station Bulletin No. 465, 1963.

5. H.A. Dunlavy, "The Handbook of Moving Air," The American Ventilation Association, Houston, Texas, 1977.

6. C.K. Wolfert and H.S. Hinrichs, "Fundamentals of Residential Attic Ventilation," H.C. Products Co., P.O. Box 68, Princeville, Ill., 1974.

7. B.A. Peavy, D.M. Burch, F.J. Powell, and C.M. Hunt, "Comparison of Measured and Computer-Predicted Thermal Performance of a Four-Bedroom Wood-Frame Townhouse," NBS Building Science Series 57, 1975.

8. P.O. Fanger, Thermal Comfort, Danish Technical Press, Copenhagen, 1970.

9. D.M. Burch, S.J. Treado, B.A. Peavy, and J.W. Bean, "Ventilating Residences and Their Attics for Energy Conservation," NBS Building Science Series (in preparation 1978.)

10. Climatic Atlas of the United States, U.S. Environmental Service, NOAA, U.S. Department of Commerce, June 1968.

11. HUD Minimum Property Standards for One- and Two-Family Dwellings, MPS No. 4900.1.

Questions and Answers

Home Ventilating Institute (HVI). Ten questions with responses by Burch and Treado:

1. For test periods 1 through 10 (Table 4), the following differences are recorded between minimums and maximums: Average outdoor temperature 3.7%; daily solar radiation 85.2%; air conditioner kWh House 2, 64.6%; air conditioner kWh House 3, 69.5%. Do not the wide discrepancies between the variation in 24-hour average outdoor temperature as against variations in solar radiation and air conditioner usage suggest that periods of air conditioner demand rather than 24-hour averaging are more meaningful in evaluating impacts of attic ventilation on energy use, energy saving, heat flow, attic temperatures and ceiling temperatures?

D.M. Burch: Table 4 gives a comparison of the air conditioner energy consumption for houses 2 and 3 (which were constructed to be identical except for differences in the attic ventilation system) during eleven consecutive periods of test. Each test period lasted one week, with the exception of test period 9, which lasted ten days. The parameters given in this table are average quantities for the test period, not daily-average quantities. These results showed that no consistent change in the pattern of air conditioner energy consumption occurred during the test periods as a result of changing the attic ventilation system.

2. A second successive week of power attic ventilation (Table 4) showed these differences in solar load and air conditioning use: House 2--solar radiation +11.0%, air conditioner kWh -6.5%; House 3--solar radiation -3.3%, air conditioner use -18.1%. Granted that there are also other factors, do not these data suggest that power venting reduces air conditioning energy use significantly and that evaluation of this and other modes of attic ventilation should continue for much longer than two successive weeks?

Burch: I think you are missing the point of Table 4. The important point is that no consistent change in the pattern of air-conditioning use occurred for two identical test houses exposed to the same weather conditions for eleven consecutive periods of test as a result of changing the attic ventilation system. The fact that variations in the average weather conditions occurred from one test period to the next is not relevant.

3. Were attic ventilation rates measured for the various attic ventilation systems studied?

Burch: A tracer-gas technique was used to estimate attic ventilation rates for House 2. For these measurements sulfur-hexafluoride tracer gas was released at the level six inches above the insulation at eight locations distributed above the living space of the house. Tracer gas was not released in the attic space above the garage. Air samples were taken at 16 locations distributed throughout the attic space above the living space and combined into a single sample. Attic ventilation rates were determined by analyzing the dilution of the tracer gas. The results of these requirements are given in the table below.

Type Attic Ventilation	WD	$T_a - T_o$ °F	V mph	I_a h^{-1}	cfm	$\dot{V}$ cfm/ft^2
Soffit venting	N	13.7	10.6	5.4	490	.32
	E	2.8	8.0	10.2	920	.60
	SSE	11.2	4.7	2.7	240	.16
Attic closed off	SSE	23.3	8.4	5.6	510	.33
	SSE	15.7	8.4	4.4	400	.26
Ridge venting	WSW	9.1	6.0	18.5	1670	1.10
	WSW	9.1	6.0	16.8	1520	.98
Power venting	SSE	5.1	1.4	18.0	1630	1.1
	SSE	5.1	1.4	18.6	1680	1.1
Turbine venting	NE	0.6	5.1	13.3	1200	0.78
Roof venting	WSW	1.7	8.0	14.5	1310	0.85
	WSW	3.9	8.0	10.3	930	0.60

4. With all attic ventilation openings closed, 5.6 and 4.4 air changes per hour were measured, with .33 and .26 cfm/ft^2. The higher rate exceeded two of three soffit rates and was more than one third the recommended rate for power venting. Does not this leakiness of construction, perhaps not typical, limit seriously the potential of any mode of ventilation to achieve added increments of heat removal?

Burch: The attic ventilation measurements presented in the foregoing table are probably somewhat on the high side. Dilution of tracer gas occurred not only due to air infiltrating between the attic space and the outdoors but also due to movement of tracer gas from the portion of the attic over the living space into the portion of the attic over the garage.

The attic ventilation rates measured for the attic closed off were performed under conditions of high thermal lead (T_a-T_o) in contrast with the measurements for soffit venting. Since attic ventilation rates do not appear elsewhere in the literature, no direct comparison with other attic ventilation values can be made to determine whether the attics of these particular test houses are of leaky construction.

If the attics of the particular houses were more leaky than those for representative houses, then the observed attic air temperatures should also be lower. However, the attic air temperatures observed for this study are consistent with values reported elsewhere. For instance, a comparison between maximum observed attic temperatures for this study is made with corresponding observed values for the Arkansas Power and Light Company [1]. Maximum observed attic temperatures for these separate studies are seen to be in close agreement. This would give a general indication that the attics of these test houses are probably no more leaky than those of the Arkansas Power and Light Company.

5. Power venting at 1.4 mph average wind velocity was shown to achieve 18.0 and 18.6 air changes per hour--a level for which 6.0 mph was required for ridge venting. Were any data recorded to show air changes for passive modes of ventilation at or about 1.4 mph?

Burch: Natural ventilation rates for passive modes of attic ventilation were not measured at low wind velocities.

6. Do not the wide variations of wind velocity and direction and related differences in attic temperatures and air changes suggest the need for much more detailed study over much longer periods to evaluate various modes of power and passive ventilation, and to identify situations where each may prove most effective?

Burch: I agree that a strong need exists to conduct a much more comprehensive study to measure ventilation rates for various natural and powered attic ventilation systems under a wide range of outdoor conditions.

7. Are the air changes and cfm figures accurately calculated? The estimated cfm figure for power venting determined from the tracer gas method is higher than the actual cfm of the power ventilator.

Burch: As I pointed out above, the measured ventilation rates given in the table are probably somewhat on the high side because of dilution of tracer gas due to movement of tracer gas from the portion of the attic over the living space to that over the garage. Also, wind [V] and temperature head [T_a-T_o] may have contributed to the ventilation rate for the attic when the power vent was operated.

8. The attic ventilating fan was sized according to 1700 ft^2 total attic floor area, which included at least 439 ft^2 of garage ceiling area not air conditioned and therefore not requiring cooling energy. It might be justifiable to deduct such garage area in sizing the fan, as would be done if the garage were detached. Deducting garage area would have reduced fan size at least 17.6% and reduced fan energy use appreciably.

Burch: The size or capacity of the power attic space ventilator was determined from a joint consensus of the Home Ventilating Institute and the American Ventilation Association. We followed your original recommendations.

9. Actual measurements of normal cycling of properly sized air conditioners and the effects of different ventilation modes under different conditions would be much more meaningful than extrapolations based on continuous running of under-sized air conditioners.

Burch: I agree.

10. The much greater complexity and number of variables involved in evaluating various modes of attic ventilation, and the limits of time and location in which the data were taken, make the paper's observation on whole-house ventilation apply even more to attic ventilation: "These results are based on very limited data. Further testing to evaluate the energy savings ... is needed." This study produced a great volume of valuable data and showed the need for far more data.

Burch: The data presented in this paper are the results of a single case study. However, these results have been used to validate a mathematical model presented in Mr. Peavy's paper which was run through a wide variation of parameters.

Arnold M. Kronstadt, P.E., Collins & Kronstadt, Leahy, Hogan, Collins, Silver Spring, Md. Two questions with responses by authors:

1. Considering the study's calculations that 25% and 16% reductions in ceiling and duct heat gain rates would reduce cooling load at maximum by 6%, and this saving was found to be within 3 watts of attic fan power consumption under the test conditions, might it not be worthwhile to determine what variations in fan operation would produce actual net energy savings? For example, it was mentioned that more than normal fan operation occurred because the installed thermostats did not perform as designed. What were the actual on-off thermostat temperatures? How would the attic fan running time and impact on cooling load have been affected had the design settings of 100°F on - 85° off been held? Or what might have been the effects at the 105° on - 90° off settings widely used in the South? Or the effects of differences in fan capacity or motor efficiency? Or more air intake? Answers from adequate testing would be useful.

Burch: The thermostatic controls of the power attic ventilators were designed and adjusted at the factory to turn on and turn off the ventilators at 100 and 85°F, respectively. This equipment was tested as installed by the American Ventilation Association; no adjustment in the set points was performed.

After power venting tests were performed, times at which the attic ventilators turned on and turned off were correlated with respect to the attic air temperatures measured below the ridge in the center of the attics. The results of this analysis are given in the table below:

	House 1	House 2	House 3
T_{on}, °F	88.3	97.6	86.8
T_{off}, °F	83.1	75.3	76.1

From the foregoing table, it is seen that both the high and low set points were lower than they should have been. Lower set points caused the ventilators to come on before they should have and to continue to operate past the point when they should have shut off. As a result, more than normal operation of the ventilators occurred.

The results of this paper showed that the addition of power venting to soffit venting had very little measured effect on the daily air conditioner energy consumption. Since the energy consumption of the power ventilator was not included in the analysis of the reduction in daily air conditioner energy consumption, the fact that more than normal operation of the ventilators occurred is not relevant. The important point is that the power ventilators did operate and remove heat from the attic space during the hot part of the day.

The results of this paper can not be used to predict the net reductions in cooling energy consumption achieved by varying the set points of the power attic ventilators. The computer model presented in Brad Peavy's paper could be used for such an analysis.

2. It is not evident that the effects of radiant heat were adequately measured in the study's short test periods and lack of occupancy. Since small differences in radiant heat can affect comfort and air conditioner settings, it would be desirable to know whether ceiling temperatures near outside walls might be significantly higher than at the center of the room; whether over a prolonged heat period low-level attic ventilation might result in buildup of radiant heat in attic insulation, attic storage contents, ceilings and other surfaces; and to what extent increased ventilation would significantly control such heat.

In future study it would be significant to measure over at least an entire season what differences in cooling effect there might be between constant positive ventilation and positive pressure with fan ventilation and non-constant soffit and other non-power ventilation, under varying conditions of outside temperature, wind, orientation, and roof color, pitch and conformation.

Burch: For this study, the ceiling temperatures were measured at the approximate centers of the five major rooms of each of the test houses. An analysis of these ceiling temperatures showed that the various attic ventilation systems compared to soffit venting produced less than a 1°F reduction in the ceiling temperature at maximum-load conditions. The reductions were found to be less during other times of the day. Such reductions would be expected to have very little effect on the indoor comfort condition. Ceiling temperatures were not measured at other locations.

D.T. Harrje, Princeton University:

Would you please clarify the reference temperature for the whole-house fan? There doesn't seem to be the same spread as with the air-conditioning -- 75/82 or 7°F difference vs 65/76 or 11°F difference.

Burch: The reference temperature (or outdoor balance temperature) is defined as the daily-average outdoor temperature at which the daily air conditioner energy consumption vanishes. For the case of Test House 1, cooled entirely with a central air conditioner (see Fig. 5), the outdoor balance temperature is seen to be 65°F when the indoor temperature is maintained at 76°F. The concept of an outdoor balance temperature for the test house cooled with a whole-house fan used in conjunction with a central air conditioner is ill defined because the indoor temperature varies considerably during periods when the whole-house fan is operated.

ANALYSIS OF ATTIC VENTILATION TEST

by

Darrell Brewster; Tom Arkfeld, P.E.
Lincoln Electric System
Suite 300, 1200 "N" Street
Lincoln, Nebraska 68508

In light of the current need to efficiently and effectively manage our existing energy supply, there has been considerable attention focused on powered attic ventilation for homes. At the same time, some disagreement has been voiced as to whether powered ventilation does conserve energy.

The effect of powered attic ventilation in a residence in Lincoln, Nebraska, was investigated and the results presented. The project compared the amount of air conditioning energy used to cool the home with the powered attic ventilator operating to the amount used with only natural ventilation. Energy consumption was adjusted for weather variations through the use of cooling degree hours.

Major conclusions of the study are:

1. With the wind conditions experienced during the test period and the natural ventilation system used in this house (soffit and ridge venting), there were no energy savings realized by using powered attic ventilation.

2. One exception to this - days with more than 150 cooling degree hours exhibited a slight decrease in the amount of cooling energy required per cooling degree hour, when the attic fan was operating.

Key words: Energy consumption; ventilation.

Introduction

A question often asked of utility companies is whether or not a powered attic ventilator will reduce the cost of cooling a home. The research project that was undertaken directs itself toward finding the answer to that question, for the Lincoln, Nebraska, geographical area, and excludes any attempt to evaluate the many other aspects of the subject.

The research project was undertaken as a result of an "attic ventilator - air conditioning research plan" offered to utilities by the Home Ventilating Institute (HVI) of Chicago, in 1976. The Home Ventilating Institute recommended the procedures followed for house selection and date procurement and a member company, the Kool-O-Matic Corporation, assumed the responsibility for selecting and installing the attic ventilator.

Location and Weather Conditions

The test was conducted on a single-family, occupied house in Lincoln, Nebraska, during the summer of 1977. The City of Lincoln is located at a latitude of 40° 5' north and a longitude of 96° 5' west and is approximately 1,150 feet above sea level.

The actual weather data during the test period compared to the thirty-year average for Lincoln, Nebraska for the months indicated is listed below.

Percent Sunshine

	Normal (30 Yr. Avg.)[1]	Actual (1977)
June	71%	74%
July	74%	79%
Aug.	74%	56%

Average Outdoor Temperature in °F

	Normal	Actual (1977)
June	72	74.4
July	77.3	80.7
Aug.	75.6	72.2

Wind Speed

	Average	Actual (1977)
July	9.2	10.7

The highest temperature reading in the upper attic during the first test period with the fan on, was 127.3°F (52.9°C) on June 26. The highest temperature recorded in the lower attic during this test period was 113.7°F (45.4°C) on June 25, while the highest outdoor temperature was 99.3°F (37.4°C) on July 5. The average wind speed on June 25 was 4.4 mph; on June 26 it was 5.8 mph; and on July 5, it was 18.5 mph. This would appear to be the primary reason that, on July 5, the lower attic temperature was only 106.6°F (41.4°C) even with the high outdoor temperature. During the second test period, the highest attic temperatures occurred on July 24,with the lower attic reading of 130.3°F (54.6°C) and the upper attic recording of 137.7°F (58.7°C). The outside temperature on that day was 99.3°F (37.4°C) and the average wind speed was 8.8 mph. On July 18, a high temperature of 99.6°F (37.6°C) was recorded with an average attic temperature of 106°F (41.1°C) and wind speeds of 16.5 mph. The highest temperature during the second test period was 100.4°F (38°C) with attic temperatures of 115.6°F (46.4°C) in the lower attic and 124.2°F (51.2°C) in the upper attic and an average wind speed of 9.9 mph. The yearly average solar energy received in Lincoln, Nebraska is 1354 Btu/ft^2 per day. Our air conditioning design temperature is 95°F (35°C) and the normal July relative humidity is 50% at 1:30 P.M. and 82% at 7:30 P.M. (2)

Test House

The test house is a two-year old, 1070-square foot, one-story, brick and frame dwelling. Occupancy of the house during the test was two adults, both of whom worked five days each week during daytime hours. The house has an attached garage (440 square feet) that is not air conditioned. There is an unfinished basement under the conditioned space in which all of the duct work is located. The house has a dark brown asphalt roof and faces north. The house is not shaded, either from trees or other buildings. The exterior glass comprises approximately 11% of the total sidewall area with 53 ft^2 on the north side, 25 ft^2 on the west and 53 ft^2 on the south side of the

(1) Thirty-year average obtained from National Weather Bureau at Lincoln, Nebraska.
(2) Data from "Handbook of Air Conditioning, Heating and Ventilating," second edition, by Stork and Koral.

house. The attic was insulated with six inches of rockwool with an estimated value of R-19. The sidewalls of the house have three-inch batt insulation with an estimated value of R-11. There is a two-mil polyethylene vapor barrier on exterior walls and under the ceiling insulation. The front door opens to the north (shaded) side of the house and is protected by a two-foot overhang. There is a large, double-glazed, picture window on the north side and glass patio doors on the south side of the house that are shaded by an awning. The calculated heat gain of the house is 25,000 Btu/h, under ASHRAE design conditions of 95/74 degrees. Of this total, approximately 12% is gained through the ceiling.

The house is cooled by a 2½-ton G.E. heat pump. The cooling system was tested by a qualified serviceman before the test began. The indoor air handling unit and the evaporator are located in the basement of the home.

Test Procedures and Data Calculation

The house was designed with seven soffit vents with 56 square inches of free ventilating area each. There is also a 20-foot ridge vent, with 18 square inches of free ventilating area per foot of length, centered along the ridge of the roof running east and west. While this is not the most common type of ventilation used by home-builders in Lincoln, there are many homes that use it. More common in this area is a combination of soffit vents with either roof vents or gable vents. The 752 square inches of ventilating area provide 5.22 square feet of the 1510 square feet of attic area (including the area above the garage). This allows one square foot of ventilation for each 289 square feet of attic area. The recommendation for the Lincoln, Nebraska, area is one square foot of ventilation for each 300 square feet of attic area.

It was necessary to add four more soffit vents for use during the power ventilator test periods to provide the required inlet area for the powered ventilator that was used. These four additional soffit vents were closed during the test periods when the powered ventilator was off to restore the house to its designed condition. Also, the ridge vent was closed off during the test periods when the power ventilator was on. The arrangement of the soffit vents and ridge vent is shown on Figure 2.

The net free ventilation areas during the test periods were as follows:

First test period - June 21 to July 12 -- power ventilator on
(1) Soffit vents - net free area 616 square inches (1,564 cm^2).
(2) Ridge vent - closed.

Second test period - July 12 to August 2 -- power ventilator off
(1) Soffit vents - net free area 393 square inches (998 cm^2).
(2) Ridge vent - net free area of 360 square inches (914 cm^2).

A Kool-O-Matic Model K-64 powered ventilator was specified and furnished by Kool-O-Matic Corporation. The thermostat controlling the power ventilator was factory calibrated to start the fan at 100°F (37.8°C) and turn it off at 85°F (29.4°C). The indoor air conditioner thermostat remained at a constant 78°F (25.6°C) setting during all of the test periods.

The attic area of the test home, including the part that is over the garage, includes a total of 2,960 cubic feet of space. With a ventilator capacity of 1230 cfm, this provided 25 air changes per hour.

The following data were measured and recorded for all parts of the test:

(1) kWh consumption of the air conditioner compressor.
(2) kWh consumption of the attic ventilator.
(3) Temperature inside the house, directly below the thermostat that is located in a center hall.
(4) Temperature in the upper attic at the same level as the attic ventilator. This temperature probe was shielded to protect it from drafts that might give a false indication.
(5) Temperature in the lower attic, just above the ceiling insulation.
(6) Temperature outside the house on the north side.
(7) Wind speed and direction.
(8) Solar intensity.

All electrical usage was measured with Westinghouse electrical meters and continuously recorded on tape with Westinghouse WR-34 magnetic tape recorders. Temperature was measured using platinum RTD's with the changing resistance, brought about by changing temperature, being converted to an analog signal through a Scientific Columbus Model 7306 SC signal conditioner. The analog signal was then converted to a digital signal that could be recorded on magnetic tape by using a Scientific Columbus Model SAF-2 pulse converter. These digital pulses were also recorded continuously by Westinghouse WR-34 recorders. All data were translated and printed by computer at thirty-minute intervals.

The wind speed and direction were measured by a recorder mounted on the roof of the house and the results were recorded on Rustrak chart recorders. The solar intensity was measured by a Science Associates Model 636-2 pyranometer mounted on the roof and the results were also recorded on a chart recorder.

Analysis of Data

The ventilation testing began on June 21, 1977, with the power ventilator on and the ridge vent blocked off to prevent air flow through it. The ventilator was allowed to run as needed for 21 days, being regulated by the thermostat that turned it on at 100°F and off at 85°F. During this 21-day period, the ventilator used 62 kilowatt-hours (kWh) of electrical energy. (See Table 1.) During the same 21-day period, the air conditioner used 424 kWh for a combined usage of 486 kWh of energy for the test period. There were 2,855 cooling degree hours* above 75°F for this 21-day period which gives an average usage of .170 kWh per cooling degree hour.

The second test period ran for 21 consecutive days beginning July 12 and was conducted with the powered ventilator off and the ridge vent open. The total energy used by the air conditioner during this test period was 540 kWh. The number of cooling degree hours for this 21-day period was 3,517, making the average usage .154 kWh per cooling degree hour for this portion of the test. These results are shown on Table 3 and show an increase in electrical use of 10.4% for the 21-day test period during which the fan operated.

If only daytime hours (8 A.M. to 9 P.M.) were used, this increase in consumption becomes smaller as indicated in Table 4. During the daytime hours of the first test period, with the fan on, the energy use per cooling degree hour was .163 kWh. During the second 21-day period, with the fan off and considering only daytime hours, the consumption was .154 kWh/cooling degree hour. This results in an increase of 5.8% in consumption when the fan was on.

The test results show a greater benefit obtained by power ventilation when the temperature increased. When comparing the days with more than 150 cooling degree hours (also shown on Table 4), there was a slight decrease in the energy used per cooling degree hour with the fan on. During the first part of the test there were six days with more than 150 cooling degree hours per day and the electrical usage was .136 KWH per cooling degree hour. During the second part of the test, while the fan was turned off, there were nine days with more than 150 cooling degree hours per day. The average usage during those days was .142 kWh/cooling degree hour. This indicates a 4.2% <u>decrease</u> in energy use with the fan on.

Another important variable appears to be wind velocity. Table 5 shows the difference in energy usage between the days when the wind speed was less than 10 mph and the days when it was greater than 10 mph. While the usage decreased from .177 to .153 kWh per degree hour with the fan on, it decreased from approximately .178 kWh when the average wind speed was less than 10 mph to .142 kWh per cooling degree hour for those hours when the fan was off, the ridge vent open and the average wind speed was greater than 10 mph. Figure 1, a graph of July 18 and 24, helps illustrate this point. The outdoor temperature was nearly the same both days, so the large difference in attic temperatures between the two days was primarily due to wind speeds. Both of the days shown were during the second test period and the fan was off.

The test was conducted for an additional 42 days, but the weather conditions were much cooler than normal and were not favorable for good testing. During the third 21-day period, from August 2 until August 23, the fan was turned on and the total usage was .271 kWh/cooling degree hour. During the final test period, from August 23 to September 13, the usage was .187 kWh/degree hour, a 44.9% decrease with the fan off.

Conclusions

With wind conditions normally experienced in Lincoln, Nebraska, the test results indicate no economic advantage to powered attic ventilation if sufficient natural ventilation is provided.

One exception to this would occur on very hot days. The test shows, as indicated on Table 2, that on days with more than 150 cooling degree hours, it took an average of 4.2% less energy per cooling degree hour during the days when the fan was operating than when it was off.

As previously indicated, a calculated 12% of the heat gain in this house is through the ceiling. During the second test period (fan-off condition), which was the hottest part of the summer, the air conditioner used 540 kWh of electrical energy. Based on this, 12% of the total 540 kWh, or 65 kWh, could have been saved if the attic was cooled to the same temperature as the conditioned space so that no heat transfer occurred. During the first test period, the power ventilator used 62 kWh of energy so that even under the ideal conditions of no heat transfer, the savings would be negligible.

It is recognized that the "degree-hours" method of normalizing outdoor temperature conditions between test periods is not linear over widely varying peak temperatures.

It must be emphasized that the results of this test are only applicable to houses in which similar conditions exist and are not intended to provide a complete assessment of attic ventilation.

TABLE I

RECORDED READINGS FOR A TYPICAL DAY (JUNE 26)

Time	Outdoor Temp. (^{o}F)	Indoor Temp. (^{o}F)	Upper Attic Temp. (^{o}F)	Lower Attic Temp. (^{o}F)	kWh Air Conditioner	kWh Ventilator	Avg Wind Speed (mph)	Solar Intensity (Btu/ft ^{2})
0100	71.4	75.4	72.1	71.4	0.00	0.00	4	0
0200	70.0	75.5	70.0	69.4	0.00	0.00	4	0
0300	69.0	74.9	68.5	68.1	0.00	0.00	4	0
0400	67.7	74.2	67.1	66.7	0.00	0.00	4	0
0500	67.1	73.4	66.0	65.6	0.00	0.00	4	0
0600	65.8	72.7	64.9	64.5	0.00	0.00	4	0
0700	65.2	71.7	64.4	64.0	0.00	0.00	2	30
0800	68.6	71.5	65.9	66.0	0.00	0.00	4	70
0900	73.0	71.8	73.9	73.6	0.00	0.00	4	110
1000	79.9	74.9	88.3	86.7	0.00	0.00	10	150
1100	85.4	78.3	98.3	94.1	0.00	0.03	8	170
1200	88.1	75.7	107.6	101.3	2.90	0.29	10	190
1300	90.3	75.1	116.0	106.1	2.20	0.29	4	200
1400	91.9	74.6	123.1	111.0	2.16	0.29	4	200
1500	92.5	74.2	126.2	112.3	2.60	0.28	6	200
1600	93.9	74.7	127.3	112.3	1.38	0.30	10	180
1700	93.6	75.0	125.2	110.2	2.30	0.29	8	150
1800	93.3	75.8	119.3	106.1	2.51	0.29	10	110
1900	91.9	75.8	111.8	101.4	2.92	0.28	6	50
2000	88.4	75.5	103.7	95.0	1.86	0.30	5	30
2100	84.5	74.9	93.9	88.5	1.41	0.29	4	10
2200	80.8	74.9	85.7	83.0	0.72	0.30	4	0
2300	78.0	74.8	81.1	78.9	0.68	0.00	4	0
2400	75.6	76.5	77.1	75.8	0.00	0.00	4	0

TABLE II

DAILY SUMMARY FOR TOTAL TEST PERIOD

Test No. 1					Test No. 2				
Date	Cooling Degree-Hours	Avg Wind Velocity (mph)	kWh A/C	kWh Fan	Date	Cooling Degree-Hours	Avg Wind Velocity (mph)	kWh A/C	kWh Fan
June 21	0.0	10.6	.56	.02	July 12 PM	87.1	7.8	16.51	0
June 22	33.4	4.8	8.89	2.54	July 13	242.8	12.5	33.17	0
June 23	104.4	5.8	15.23	2.58	July 14	243.6	14.5	36.23	0
June 24	85.5	4.0	16.73	2.77	July 15	122.8	6.0	27.45	0
June 25	124.6	4.2	23.54	3.40	July 16	241.3	9.8	35.90	0
June 26	183.1	5.1	23.66	3.51	July 17	311.7	11.3	45.76	0
June 27	211.1	10.1	25.74	3.66	July 18	317.9	16.5	45.07	0
June 28	26.3	11.7	9.14	1.47	July 19	295.2	16.0	33.58	0
June 29	112.0	12.1	8.14	2.80	July 20	304.1	9.9	35.19	0
June 30	25.5	14.1	9.44	1.97	July 21	43.0	14.3	12.66	0
July 1	118.8	6.7	13.16	3.21	July 22	95.8	6.8	15.44	0
July 2	136.0	8.9	21.54	2.99	July 23	200.7	8.5	34.63	0
July 3	271.0	18.9	35.86	3.82	July 24	238.4	8.8	41.47	0
July 4	266.7	17.3	33.86	3.74	July 25	49.3	10.5	18.14	0
July 5	286.6	18.4	37.12	3.89	July 26	38.5	8.0	12.99	0
July 6	292.3	15.0	40.50	4.08	July 27	92.0	6.4	12.76	0
July 7	104.9	6.8	20.94	2.15	July 28	58.6	8.6	0.00	0
July 8	109.8	9.0	20.61	3.13	July 29	157.8	6.4	26.54	0
July 9	106.3	7.9	17.72	3.13	July 30	183.0	11.2	28.54	0
July 10	151.3	10.3	20.92	3.34	July 31	56.2	8.4	15.34	0
July 11	75.4	10.0	15.57	3.03	Aug. 1	130.6	8.9	12.34	0
July 12 AM	30.4	7.8	4.66	1.09	Aug. 2	4.3	7.5	.66	0
TOTAL	2,855.4		423.53	62.30		3,516.7		540.37	0

Energy Used .170 kWh/Cooling Degree-Hour .154 kWh/Cooling Degree-Hour

TABLE III

COMPARISON OF CONSUMPTION FOR ALL DEGREE HRS. ABOVE 75°F

Test	Average Wind Velocity (mph) at temp. 75°	No. of Days	Cooling Degree-Hours	kWh Compressor	kWh Fan	kWh Per Cooling Degree-Hr.	Percent Increase (Decrease)
Test No. 1 (Fan on)	12.05	21	2,855	424	62	.170	------
Test No. 2 (Fan off)	11.30	21	3,517	540	0	.154	(10.4)

TABLE IV

COMPARISON OF CONSUMPTION FOR DAYTIME HOURS (8 AM - 9 PM) ABOVE 75°F

Test	Average Wind Velocity (mph)	No. of Days	Cooling Degree-Hours	------kWh-------- Compressor	Fan	kWh Per Cooling Degree-Hr.	Percent Increase (Decrease) In Energy Use
Test No. 1							
(Fan on)	11.6	21	2,529	357	55	.163	------
Test No. 2							
(Fan off)	11.1	21	2,892	445	0	.154	(5.8)

COMPARISON OF CONSUMPTION FOR DAYS WITH MORE THAN 150 DEGREE HOURS

Test	Average Wind Velocity (mph)	No. of Days	Cooling Degree-Hours	------ kWh-------- Compressor	Fan	kWh Per Cooling Degree-Hr.	Percent Increase (Decrease) In Energy Use
Test No. 1							
(Fan on)	15.3	6	1,288	157	18	.136	------
Test No. 2							
(Fan off)	13.3	9	2,051	292	0	.142	4.2

TABLE V

EFFECT OF WIND SPEED

Wind Speed Greater Than 10 mph

Test	Cooling Degree-Hours	kWh Air Conditioner	kWh Fan	kWh Per Cooling Degree-Hour
Test No. 1 (Fan on)	1,702	227.7	32.3	.153
Test No. 2 (Fan off)	1,666	237	0	.142

Wind Speed Less Than 10 mph

Test	Cooling Degree-Hours	kWh Air Conditioner	kWh Fan	kWh Per Cooling Degree-Hour
Test No. 1 (Fan on)	814	121.0	23.2	.177
Test No. 2 (Fan off)	1,167	208.2	0	.178

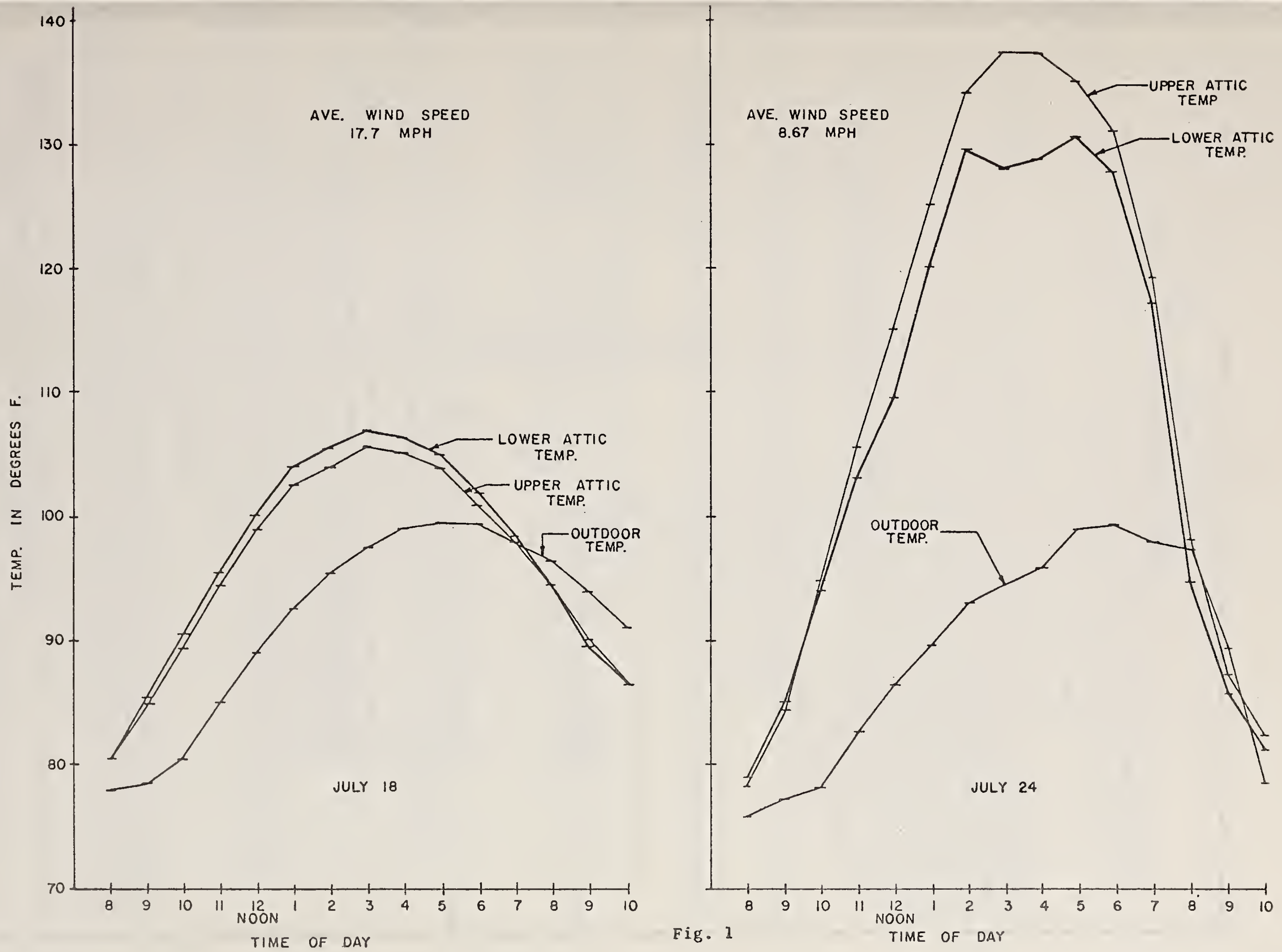

TEMP. IN DEGREES F.
140
130
120
110
100
90
80
70
AVE. WIND SPEED
17.7 MPH
LOWER ATTIC TEMP.
UPPER ATTIC TEMP.
OUTDOOR TEMP.
JULY 18
8 9 10 11 12 NOON 1 2 3 4 5 6 7 8 9 10
TIME OF DAY
AVE. WIND SPEED
8.67 MPH
UPPER ATTIC TEMP
LOWER ATTIC TEMP.
OUTDOOR TEMP.
JULY 24
8 9 10 11 12 NOON 1 2 3 4 5 6 7 8 9 10
TIME OF DAY

Fig. 1

LOCATION OF VENTILATORS
ON TEST HOUSE (PER ORIGINAL DESIGN)

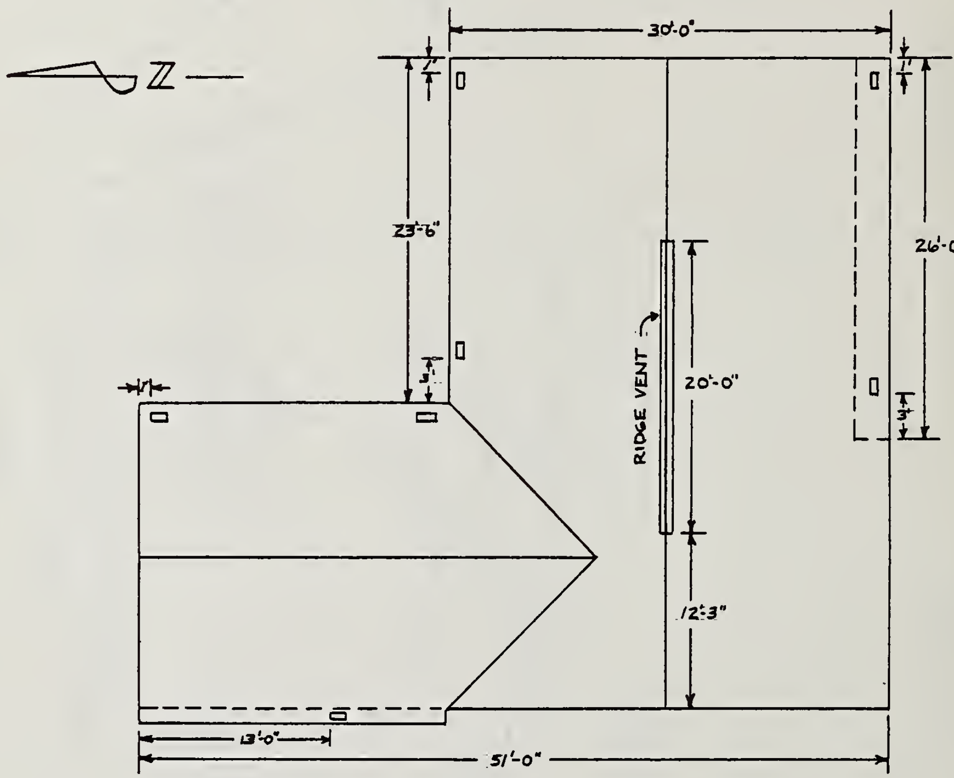


FIG. 2

Questions and Answers

Home Ventilating Institute (HVI). Six questions with Brewster's responses:

1. In relating the results to other areas, the 4.2% less energy per cooling degree hour with powered attic ventilation on days with more than 150 cooling degree hours may prove more important than the finding that in that particular house and location no economic advantage to powered attic ventilation was noted. Few localities in the country have the high average winds of 11.1 to 15.3 mph recorded in this study.

D.R. Brewster: The study house was typical of houses in this area. It was typical in size, style, type of construction and amount of insulation. Our report presented the results of our test, and we do not feel that the results can be generalized to other areas of the country. The Handbook of Air Conditioning, Heating and Ventilating, second edition, by Strock and Koral, reports a survey of average wind speeds for a thirty-year period. The average wind speeds for the month of July in 181 cities located in 47 states are recorded. The average wind speed recorded in all of these cities was 8.06 mph, with 43 of the cities reporting higher average wind speeds than Lincoln, Nebraska. This would indicate that the wind conditions in Lincoln are quite similar to those in other areas of the country.

2. The high prevailing winds are favorable for passive ventilation, as the study shows. Even so, power venting doubtless would have proved more cost effective if the fan had been sized 29% smaller by deducting attic space over the attached garage.

Brewster: The size of the fan was determined by a member company of the Home Ventilating Institute. The house that was tested and installation of the fan was also approved by the same company.

3. Energy reduction also would have increased had the air conditioner thermostat setting of 78°F been held, instead of the 74.2-75.7 recorded.

Brewster: While a thermostat setting of 78°F was maintained, the actual indoor temperature ranged from 74.2 to 75.7°F during the test. The effect of the reduced indoor temperature would have had a small effect on the test results.

4. The 2 1/2-ton heat pump may have been oversized for air conditioning, which by some calculations would call for 1 1/2 to 2 tons of cooling capacity. This factor might have affected efficiency in cooling and potential effectiveness of attic power venting.

Brewster: According to a detailed heat gain analysis of the house, the air conditioner is only slightly oversized. The resulting loss of efficiency should have little, if any, effect on the test results.

5. The measured data compared power venting with ridge venting, which, as stated, is not the most common type of ventilation in the area. The comparison was with the most effective type of passive ventilation but also the least prevalent in most areas-- used mainly in new homes and not a normal retrofit option.

Brewster: We concur with this observation.

6. The authors emphasize that the test results apply only to houses in which similar conditions exist. It may be noted further that the study house was not typical for the locality and far less typical of the millions of existing homes that make up the vast bulk of housing.

Brewster: The study house was typical of houses in this area. It was typical in size, style, type of construction, and amount of insulation.

A MODEL FOR PREDICTING THE THERMAL PERFORMANCE OF VENTILATED ATTICS

B.A. Peavy
Center for Building Technology
National Bureau of Standards
Washington, D.C. 20234

ABSTRACT

A model for predicting the dynamic thermal performance of ventilated attics is presented. A computer program incorporating the heat transfer calculations was validated by the observed data obtained from tests performed in Houston, Texas during the summer of 1977. With the very good agreement between predicted and measured attic thermal performance, it can be expected that the model will adequately predict the thermal performance for a wide variation in operating conditions, climate and geographical location. Employing the weather data and data characteristic to the Houston, Texas attic, some numerical cases are cited for natural ventilation air flow only, and power vent-fan operation when called for, where thermal performance is to be determined for variations in thickness of insulation, variations in natural ventilation air flow rates, and variations in outdoor wind speed. Except for the case with zero outdoor wind speed during a simulated daily weather cycle, all other cases showed that the daily energy consumption of the power vent fan exceeds the daily reduction in the energy required to operate the air conditioning equipment.

Keywords: Attic ventilation; mathematical modeling; predicted and measured attic thermal performance; thermal performance of attic ventilation systems.

1. Introduction

In this paper the dynamic heat transfer occurring in ventilated attic-roof-ceiling combinations of residential buildings is analyzed in order to determine the heat gains to or the heat losses from the habitable spaces below the ceiling. This analysis would be particularly useful for investigating the energy conservation merits of various attic ventilating systems where both summer cooling and winter heating conditions should be considered.

During winter heating conditions, the attic spaces should be ventilated in order to prevent an excessive accumulation of moisture in the insulation and structural elements. Recommended good practice for attic vent openings to provide natural ventilation is given in Chapter 20 of the ASHRAE Handbook of Fundamentals [1]. With these minimum requirements for vent openings, a certain amount of attic ventilation by natural forces of wind and temperature difference will occur at all times of a year. At most times, air flow through an attic space can be characterized as a process of absorption of heat or an increase in enthalpy of the air. In the winter heating condition, heat and moisture flow from the habitable spaces below the attic increase the enthalpy of the ventilating air. At all times of the year, solar radiation upon roof surfaces will increase attic temperatures and ventilating air will increase in temperature as it passes through the attic.

A very comprehensive experimental and analytical study dealing with attic ventilation was performed by Joy [2]. The experimental work was done in the laboratory under controlled steady-state conditions and the experimental results showed good agreement to a predictive model derived from a mathematical analysis. The model of this paper is patterned after the mathematical analysis of Joy, and includes an hour-by-hour determination of temperature and heat flows in the attic affected by dynamic changes in the outdoor weather. Dynamic heat-transfer processes occurring in an attic are extremely complex, where in addition to accounting for the three modes of heat transfer-- conduction, convection and radiation-- the ventilating air flow rates under natural and forced conditions are not uniformly distributed and are dependent upon outdoor wind speed, direction and temperature difference.

During the summer of 1977, attic ventilation research was performed by the National Bureau of Standards at housing sites in Houston, Texas [3]. Weather data from tests involved with and without power ventilation were used as input to the ventilated attic model from which the predicted thermal performance was compared with that measured during the tests. If there is good agreement between predicted and measured attic thermal performance, then it can be reasonably expected that the model would adequately predict the thermal performance for a wide variation in climate and geographical location.

2. Analysis

The mathematical analysis consists of heat balance equations which are to satisfy heat conduction through the solid materials of the roof and ceiling, convection heat transfer between surfaces and the ambient air, and radiation heat transfer between surfaces. Figure 1 is a schematic of an attic-ceiling-roof combination showing the temperatures necessary for determining heat flow quantities. The surface temperatures are:

T_{ro} - roof outside
T_{ri} - roof inside
T_{co} - ceiling outside (attic floor)
T_{ci} - ceiling inside.

The air temperatures are:

T_i - indoor air of habitable space
T_o - outdoor air
T_s - sol-air (eq 16)
T_a - average attic air.

Assumptions made in the analysis are outlined and discussed in the following:

1. Heat flow by conduction through gable ends of pitched roofs is ignored. Generally, the heat flow through the area involved in gables is small in comparison to that for the roof.

2. Radiation within the attic space treats two non-refractory surfaces, the underside of the roof and the attic floor. All other surfaces, such as the gable ends, are assumed refractory, from which there is no net radiant heat flux. When radiant exchange is particularly significant, one gable end may be receiving solar energy while the other, being in the shade, may lose energy, whereby the net exchange is negligible.

3. Coefficients of heat transfer at all surfaces are determined from the two components, namely, radiation and convection-conduction. The definitions for these components are given later in this paper. These coefficients are usually dependent upon temperature and therefore iteration procedures must be established to converge upon the correct temperatures.

4. Entry of outdoor air into attic spaces is usually from more than one position, such as at louvers in gables and soffit vents, such that the air flow path to the points of exit cannot be defined by the length or width of an attic space. For instance, an attic fan located at the midpoint in a roof may draw air principally from the nearest soffit vent, as that may be the position of least resistance to air flow. The path length of air flow will then have to be determined as a matter of judgment.

The heat transfers at the separate surfaces (refer to Figure 1) and at a given time are given by the following relations:

Outside surface of roof $$Q_{ro} = h_o\,(T_s - T_{ro}) \quad (1)$$

Inside surface of roof $$Q_{ri} = h_{r1}\,(T_{ri} - T_{co}) + h_{c1}\,(T_{ri} - T_a) \quad (2)$$

Attic floor $$Q_{co} = h_{r2}\,(T_{ri} - T_{co}) + h_{c2}\,(T_a - T_{co}) \quad (3)$$

Ceiling $$Q_{ci} = h_i\,(T_{ci} - T_i) \quad (4)$$

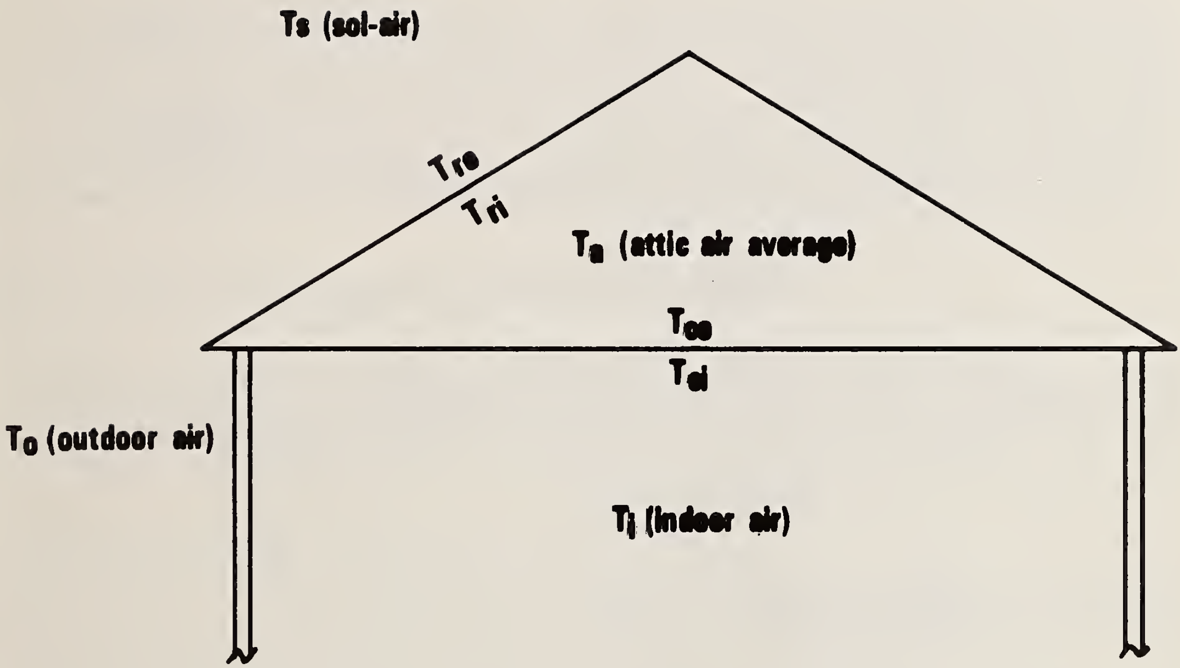


Figure 1. Schematic of ventilated attic-ceiling-roof combination.

For air flowing in the attic from the entry (y=0), where the temperature is the same as the outdoor air temperature, to the point of exit (y=R), the air temperature will then be a function of the distance traveled along the air flow path as heat is either added to or abstracted from the air as given by equations (2) and (3). It is then necessary to determine at any given time, the average attic air temperature, T_a. From elementary heat transfer texts, a heat balance equation satisfied by the temperature in the attic air is found to be

$$(A_{ri} + A_{co})\ Mc\ [\frac{dT_y}{dt} + V\frac{dT_y}{dy}] + A_{co}\ Q_{co} = A_{ri}\ Q_{ri} \quad ,$$

where M is the mass of air in the attic space per unit surface contact area, c is the specific heat of air, V is the air velocity, y is a dimension for the air flow path, A_{co} is the attic floor area, A_{ri} is the inside surface area of the roof and T_y is the air temperature at position y. Because the above equation will be used at discrete time intervals and the change in temperature with respect to time is of much smaller magnitude with respect to the change of temperature with respect to the dimension, y, it is assumed that $dT_y/dt = 0$ for this analysis. For arbitrary attic geometries and from the identity that volume rate of fluid flow is equal to the product of average velocity and average area, a relationship can be derived

$$\frac{(A_{co} + A_{ri})\ McV}{A_{co}} = \rho\ cLV_a$$

where ρ is air density, L is an attic floor dimension parallel to the air flow and V_a is the volume rate of air flow per unit of attic floor area. By appropriate substitutions from equations (2) and (3) where $T_a = T_y$, the differential equation becomes

$$\frac{dT_y}{dy} + B_1\ T_y = B_2$$

where $B_1 = (h_{c2} + \frac{A_{ri}}{A_{co}}\ h_{c1})/\ \rho\ cLV_a$, and

$$B_2 = (h_{c2}T_{co} + \frac{A_{ri}}{A_{co}}\ h_{c1}T_{ri})/\rho\ cLV_a \quad ,$$

from the identity $A_{co}h_{r2} = A_{ri}h_{r1}$. A solution to the differential equation is:

$$T_y = \frac{B_2}{B_1} + (T_o - \frac{B_2}{B_1})\ e^{-B_1 y} \qquad (5)$$

where at y = 0, $T_y = T_o$, the entering outdoor air temperature. The average attic air temperature is found by integrating equation (5) with respect to y over the air flow path from y = 0 to y = R, the distance from air entry to air exit. This integration gives

$$T_a = \frac{B_2}{B_1} + \frac{(T_o - \frac{B_2}{B_1})}{B_1 R}\ (1 - e^{-B_1 R}) \quad ,$$

or

$$T_a = C_1\ T_{co} + C_2\ T_{ri} + C_3\ T_o \quad , \qquad (6)$$

where

$$C_1 = \frac{h_{c2}\,(1 - C_3)}{h_{c2} + h_{c1}\,A_{ri}/A_{co}} \,,$$

$$C_2 = \frac{h_{c1}\,A_{ri}/A_{co}\,(1 - C_3)}{h_{c2} + h_{c1}\,A_{ri}/A_{co}} \,,$$

$$C_3 = \frac{(1 - e^{-B_1R})}{B_1R} \,, \text{ and}$$

$$B_1R = \frac{h_{c2} + h_{c1}\,A_{ri}/A_{co}}{\rho C_p\,V_a L/R} \,.$$

The heat transfer to the air in appropriate units from (5) and y = R is

$$Q_a = \rho c V_a\,(T_R - T_o) \,.$$

Using the response factor notation [4], (1) through (4) become

$$\Sigma Y_{j1}\,(T_{ri_{t-j}} - T_m) - \Sigma Z_{j1}\,(T_{ro_{t-j}} - T_m) + h_o\,(T_{s_t} - T_{ro_t}) + \overline{CR}_1\,Q_{ro_{t-1}} = 0 \qquad (1a)$$

$$\Sigma X_{j1}\,(T_{ri_{t-j}} - T_m) - \Sigma Y_{j1}\,(T_{ro_{t-j}} - T_m) + CR_1\,Q_{ri_{t-1}} + h_{r1}\,(T_{ri_t} - T_{co_t}) + h_{c1}\,(T_{ri_t} - T_{a_t}) = 0 \qquad (2a)$$

$$\Sigma Y_{j2}\,(T_{ci_{t-j}} - T_m) - \Sigma Z_{j2}\,(T_{co_{t-j}} - T_m) + \overline{CR}_2\,Q_{co_{t-1}} + h_{r2}\,(T_{ri_t} - T_{co_t}) + h_{c2}\,(T_{a_t} - T_{co_t}) = 0 \qquad (3a)$$

$$\Sigma X_{j2}\,(T_{ci_{t-j}} - T_m) - \Sigma Y_{j2}\,(T_{co\,t-j} - T_{i_{t-j}}) + \overline{CR}_2\,Q_{ci_{t-1}} + h_i\,(T_{ci_t} - T_{i_t}) = 0 \qquad (4a)$$

Equations (1a), (2a), (3a), (4a), and (6) contain five unknown temperatures T_{ro}, T_{ri}, T_{co}, T_{ci} and T_a at time t. Assuming that the temperatures and heat fluxes are known for time t-1, t-2, etc., the unknown temperatures at time t can be found by suitable substitutions. In the above equations, the summations are for j = 0, 1, 2....N, where N is dependent upon the heat capacity of the roof or ceiling. CR is a factor to be multiplied by the appropriate surface heat flux for the previous hour.

3. Heat Transfer Functions

Convection Heat-Transfer Coefficient

Values for the convection component of surface conductance used to obtain surface conductances given in ASHRAE Handbook of Fundamentals are based on the results of tests made on 12-inch-square samples of different materials. More recent tests [5] show that the surface length significantly affects the convection coefficient, specifically the forced convection coefficient, for which the expression becomes

$$h_f = 0.664\ V^{0.8}\ L^{-0.2} \tag{7}$$

where

V = air velocity, mph

L = surface length, ft.

At low air velocities the forced coefficient must be augmented by a natural convection coefficient. Parmelee [5] states that for low velocities, say under 10 fps (6.82 mph, 3.048 ms^{-1}), it is suggested that a natural convection coefficient be added to the forced convection coefficient. For attic spaces, even with power fans operating, the average air velocity over attic surfaces is well below 10 fps.

The statement of Parmelee is somewhat anomalous, particularly in the region above and below 10 fps, at which there is a sudden discontinuity. For the purpose of this paper, the convection coefficient of heat transfer was computed from the following heuristic relationship

$$h_c = h_f + \frac{(46.512 - V^2)}{46.512}\ h_n, \qquad V \leq 6.82 \text{ mph}, \tag{8}$$

and $h_n = 0$ for $V > 6.82$ mph, where V is the air velocity and h_n is the natural convection coefficient. This relationship was used for computation at all roof and ceiling surfaces.

Natural convection heat-transfer coefficients according to the orientation and direction of heat flow can be defined by the relationships as follows: [1]

1. Horizontal materials with surface facing upward when being heated and facing downward when being cooled

$$h_n = 0.22\ (\Delta T)^{1/3}\ . \tag{9}$$

2. Material with surfaces vertical

$$h_n = 0.19\ (\Delta T)^{1/3}\ . \tag{10}$$

3. Horizontal materials with surface facing upward when being cooled or facing downward when being heated

$$h_n = 0.11\ (\Delta T)^{1/3}\ , \tag{11}$$

where ΔT = temperature difference (°F) between surface and the air.

For the inside roof surface, which is neither vertical or horizontal, the natural convection heat-transfer coefficients are assumed to be of the form:

(heat flow up) $$h_n = \frac{1.393\ \Delta T^{1/3}}{7.333 - \cos\phi} \quad \text{and} \tag{12}$$

(heat flow down) $$h_n = \frac{.2613\ \Delta T^{1/3}}{1.375 + \cos\phi}\ , \tag{13}$$

where ϕ is the roof pitch angle.

Radiation Coefficient of Heat Transfer

The radiation component of heat transfer is defined by the relationship:

$$\begin{aligned} h_r &= \sigma \overline{F}_{12}\ (T_1 + T_2 + 920)\ \ [(T_1 + 460)^2 + (T_2 + 460)^2] \\ &\simeq 4\ \sigma\ \overline{F}_{12}\ (T_m + 460)^3 \end{aligned} \tag{14}$$

where

σ = Stefan-Boltzmann constant = $.1714 \times 10^{-8}$ Btu h^{-1} ft^{-2} R^{-4} (5.67×10^{-8} W m^{-2} K^{-4})

$\overline{F}_{12}$ = Configuration factor to allow radiation exchange between surface 1 and surface 2 with consideration for the emittances of the two surfaces and where all other surfaces are assumed to be refractory surfaces.

$T_m = (T_1 + T_2)/2$.

For radiant energy exchange between the inside roof surface and the attic floor, the radiation exchange factor becomes: [6]

$$\overline{F}_{12} = \frac{1}{\dfrac{1}{\epsilon_1} - 1 + \dfrac{A_{ri}}{A_{co}}\left(\dfrac{1}{\epsilon_2} - 1\right) + \dfrac{1 + (1-2F_{12})\ A_{ri}/A_{co}}{1 - F_{12}^2\ A_{ri}/A_{co}}} \tag{15}$$

where ϵ_1 and ϵ_2 are emissivities of the inside roof and attic floor respectively, and F_{12} is the radiation configuration factor between the two surfaces as a function of the dimensions and the pitch angle between the surfaces.

Sol-air Temperature

Sol-air temperature is defined by the relationship:

$$T_s = T_o + (\alpha I_H - S)/h_o\ , \tag{16}$$

where

T_o = outdoor air temperature

α = roof surface solar absorptance

I_H = total solar radiation incident on horizontal surface

S = $\sigma\ [(T_o + 460)^4 - (T_e + 460)^4]$ = long-wave radiation to the sky

h_o = coefficient of heat transfer by radiation and convection at roof surface (see previous algorithms for h_r and h_c)

T_e = equivalent sky temperature [7].

Attic Ventilation Rates

Air infiltration tests were performed in the attic of house number 2 at Houston. From these tests, an empirical relationship for the attic air ventilation rate under natural conditions was derived:

$$V_a = .45\ W_o\ (.087 + .131|\sin D|^{5/2}) \tag{17}$$

where

V_a = volume air flow rate per ceiling area, $ft^3\ min^{-1}/ft^2$

W_o = wind speed, mph

D = wind direction as measured from the south, degrees,

For power vent fan operation, attic ventilation rates are determined from the relation:

$$V_a' = V_a/4 + 1.188P/100 \tag{18}$$

where P is the percentage of fan operation during a given hour of time.

Equations (17) and (18) are empirical relations for attic ventilation air flow rates developed from the Houston experimental data, and could not be expected to be valid for other attic configurations and ventilation openings. For natural ventilation, the algorithms for air flow rates could be deduced from relations given in p. 22.12 ASHRAE Handbook 1977 [1], for air flow due to combined wind and temperature difference effects.

4. Computer Program

The computer program incorporating the algorithms of the previous section is found in the appendix, and is specifically intended to solve for temperatures and heat flow quantities upon input of weather and operational data from tests performed by the National Bureau of Standards at housing sites in Houston, Texas during the summer of 1977. Hourly time interval data were used as input to the computer program as follows:

1. Outdoor dry-bulb temperature,
2. Indoor air dry-bulb temperature,
3. Wind speed,
4. Wind direction,
5. Solar radiation incident on horizontal surface,
6. Equivalent sky temperature (long-wave radiation),
7. Percent time operation of power vent fan,
8. Percent time operation of air conditioner blower.

For other geographic locations and climates, not all of the above information will be available, and it will be necessary to develop or utilize appropriate algorithms for this determination. If solar radiation data are not available, there are algorithms available for its determination at any geographic location. Equivalent sky temperature may be determined from relations given by Bliss [7]. Item 7 above is not needed for design problems because the power vent fan operation can be started and stopped by noting appropriate attic air temperatures, as discussed later. Item 8 above is mainly a function of heat gain to the spaces below the ceiling-attic combination for which the air conditioning system is turned on to maintain space temperature. For design problems, item 8 would probably have to be estimated.

For conduction heat transfer across roof and ceiling, response factors which include roof rafters for the roof and ceiling joists for the ceiling were input. Because heat flow meters used to record heat flow through the ceilings were installed between the ceiling joists, the conductance of the ceiling without ceiling joists was used to determine ceiling heat flux for each time interval. For the Houston tests, the ceiling consisted of nominal 2 x 4 inch ceiling joists, 16" on centers with 1/2" gypsum plasterboard underneath. The joist cavity was filled with four inches of mineral wool insulation.

Other necessary inputs to the computer program are:

1. Length of attic space,
2. Width of attic space,
3. Pitch angle of roof,
4. Height of roof peak above attic floor,
5. Path length of air flow through attic for natural conditions,
6. Attic air ventilation rate under natural conditions, VA1
7. Attic air ventilation rate under power vent fan operation, VA2
8. Leaving attic air temperature at which power fan operation is to be started, TA1
9. Leaving attic air temperature at which power fan operation is to be stopped, TA2
10. Emittance of inside roof surface,
11. Emittance of attic floor surface.

In the tests performed at Houston, the percent hourly time operation of the power vent fan was known, so that for comparison, the values for TA1 and TA2 were set to unattainably high values and ventilation air flow rates were determined from algorithms given in the previous section. For computer runs with variations in flow rates or ceiling insulation thicknesses, values of percent time operation of power vent fans were set equal to zero, TA1 = 100°F, TA2 = 85°F, and VA1 and VA2 determined from (17) and (18).

Additionally, the test houses at Houston contained air conditioning system air supply ducts in the attic, for which the heat balance incorporated in the computer program was assumed to be:

$$Q_d = J_d(T_a - T_d) ,$$

where T_a = average attic air temperature

T_d = duct air temperature = 55°F (assumed)

$J_d = U_d\, M(t)\, A_d/A_{co}$

U_d = thermal transmittance of duct

$M(t)$ = proportional part of air conditioner operation during hour time interval

A_d = duct surface area

A_{co} = attic floor area.

5. Test Data and Computer Results

During the summer of 1977, three identical houses in Houston, Texas were extensively instrumented for the purpose of determining their attic thermal characteristics under the dynamic weather variations and with variation in the quantity of attic ventilating air, such as by soffit venting, power vent fan, turbine venting and ridge venting. For this paper, input data from house number 2 for attic ventilation by only soffit venting and power vent fan were used in the model computer program.

Pertinent data for a three-day period, September 2, 3 and 4, 1977, represent soffit venting conditions where natural ventilation occurs due to wind and temperature difference forces. Input data for the computer programs are shown in figure 2 for September 4, 1977. A comparison of experimental and computer-predicted results for ceiling heat flux and average attic air temperature is shown in figure 3.

Pertinent data for a three-day period, August 2, 3, 4, 1977, represent power vent fan operation where the fan is started upon attic air temperature attaining a thermostatted temperature and the fan is stopped when the temperature drops to another thermostatted temperature. At other times, attic ventilation occurs due to natural forces. Input data for the computer program are shown in figure 4 for August 4, 1977 and are also shown in sample problem given in the Appendix. A comparison of experimental and computer-predicted results for ceiling heat flux and average attic air temperature is shown in figure 5. The computer-predicted results for August 4, 1977 are shown in Table A-1 of the Appendix.

In figures 3 and 5, the predicted average attic air temperature is always lower than the test values during the nighttime hours. This may be due to methods by which long-wave radiation is determined in relation to the temperature near the ground or roof [6] and to the pitch angle of the roof allowing the roof surface to see things other than the sky. In general, there appears to be very good agreement between predicted and measured values, considering that the experimental values represent the average of measurements taken at four or five locations over a very large area.

6. Prediction of Thermal Performance

As shown in the previous section, the thermal performance of the model gives very good agreement with experimental results and thereby should be a useful tool for predicting thermal performance in a wide variety of operating conditions, climate, and geographical locations.

Results from only a few cases will be presented and discussed. The primary objective will be to determine the effectiveness of power vent fan operation in relation to ventilation only by natural forces such as wind and temperature difference. All cases with power vent fan operation assume the power vent fan is turned on only when the leaving attic air temperature exceeds 100°F and the fan is turned off when the temperature falls below 85°F. Also, the basic weather cycle of figure 4 (August 4, 1977) will be used in all cases.

Of interest is the effect of variations in the thickness of ceiling insulation upon the thermal performance. Assuming a ceiling with one-half inch (1/2") gypsum plasterboard nailed to ceiling joists (16-inches on center) with 1, 4 and 6 1/2-inch thickness of insulation placed between the joists, the calculated ceiling heat flux as a function of time is shown in figure 6. Table 1 gives the daily average ceiling and duct heat flux, and fan on-time. These results show that the use of power venting produced a reduction in the combined heat flow (ceiling and duct) ranging from 2.4 to 6.3 percent as the insulation increased from 0 to 6.5 inches. The decrease in percent reduction as the insulation thicknesses became smaller is explained by the following: The attic floor receives thermal radiation from the heated roof above and is warmer than the attic air. As the amount of ceiling insulation is reduced, conduction heat transfer from the attic floor to the living space below is increased with a subsequent lower attic floor temperature. This creates less temperature difference between the attic floor and the attic air and less heat flow to the attic air, whereby power venting will have less percent effect on reducing ceiling heat gain rates.

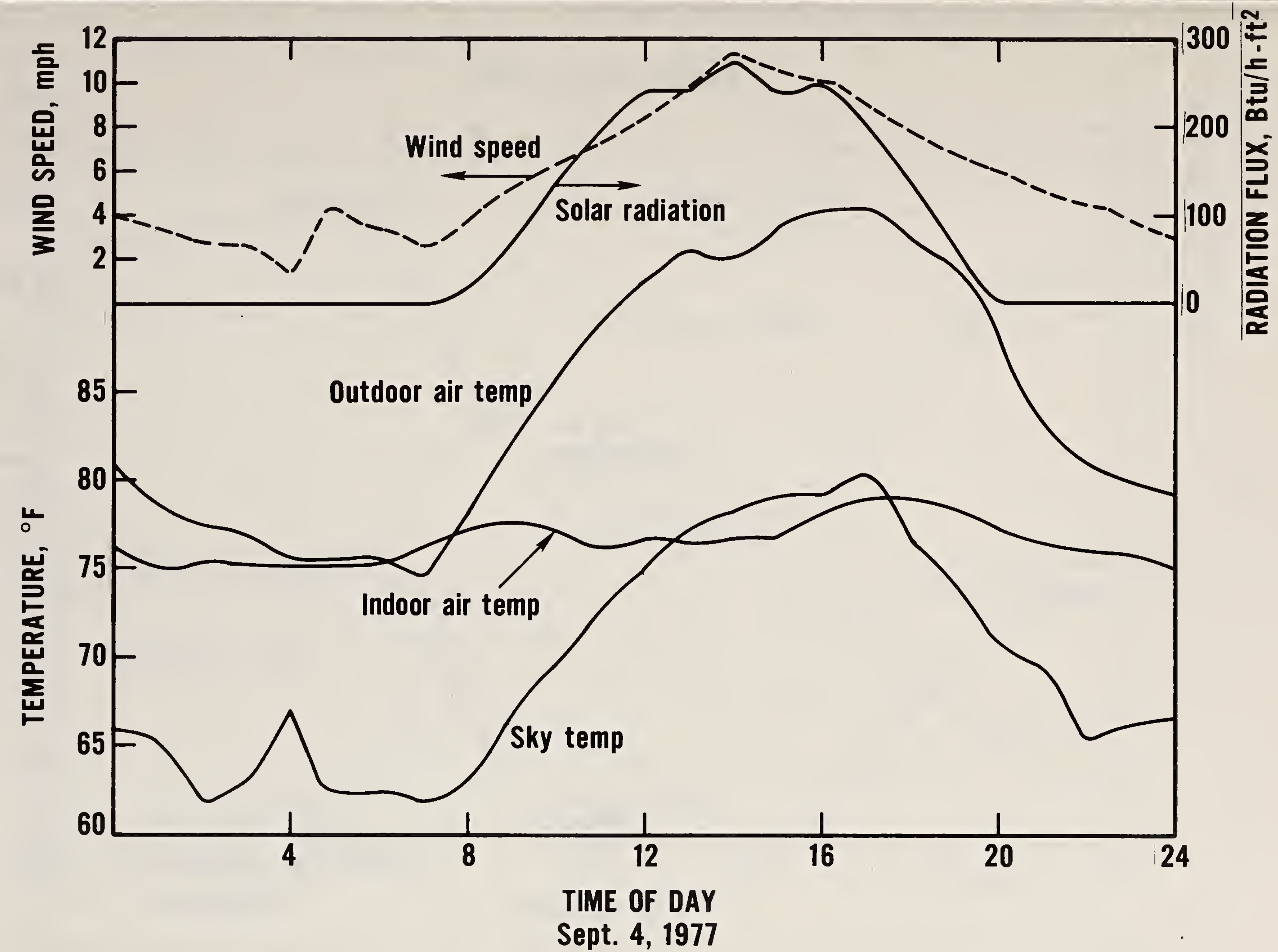


Figure 2. Weather data for September 4, 1977.

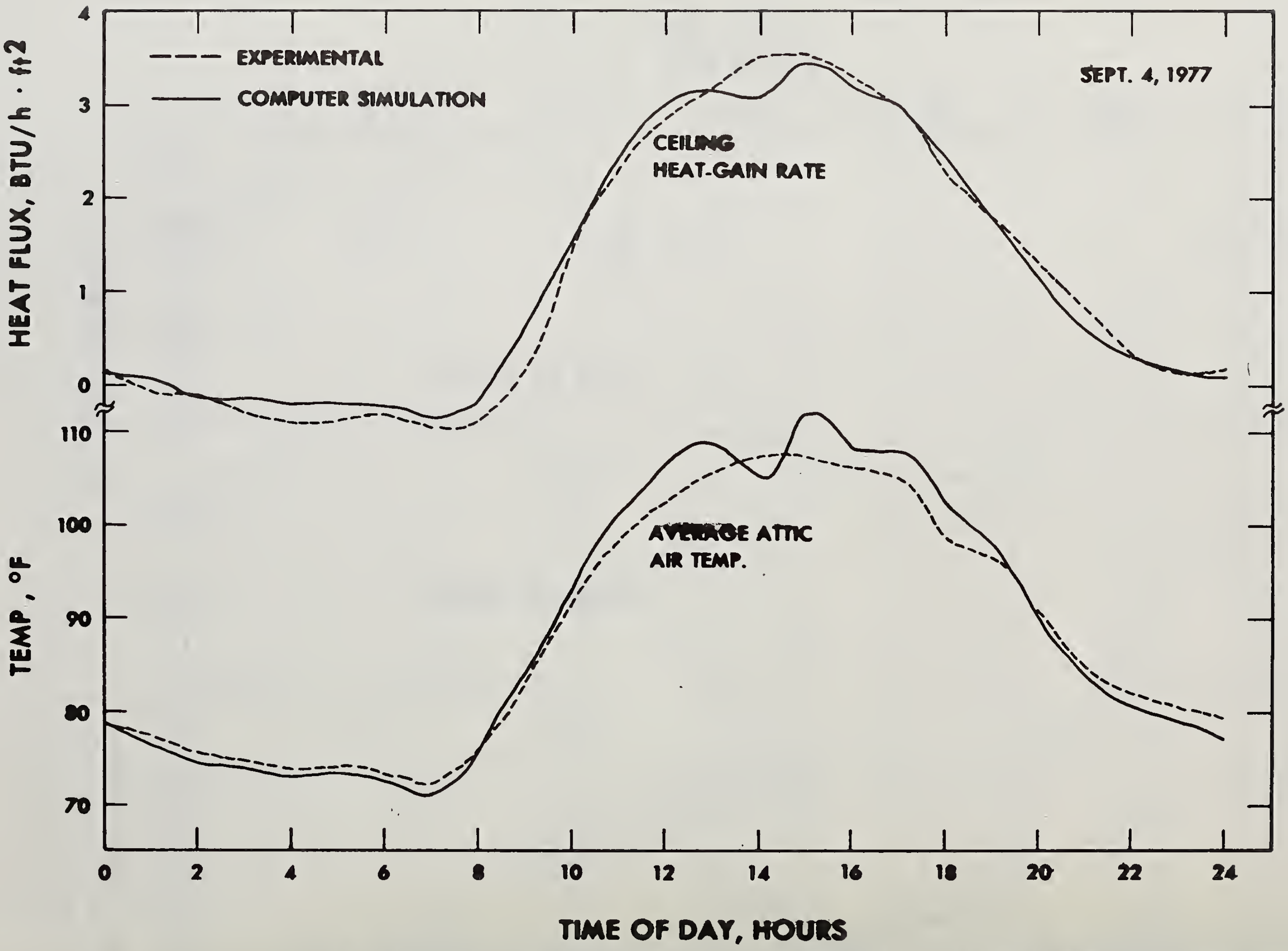


Figure 3. Comparison of ceiling heat flux and average attic air temperature for September 4, 1977.

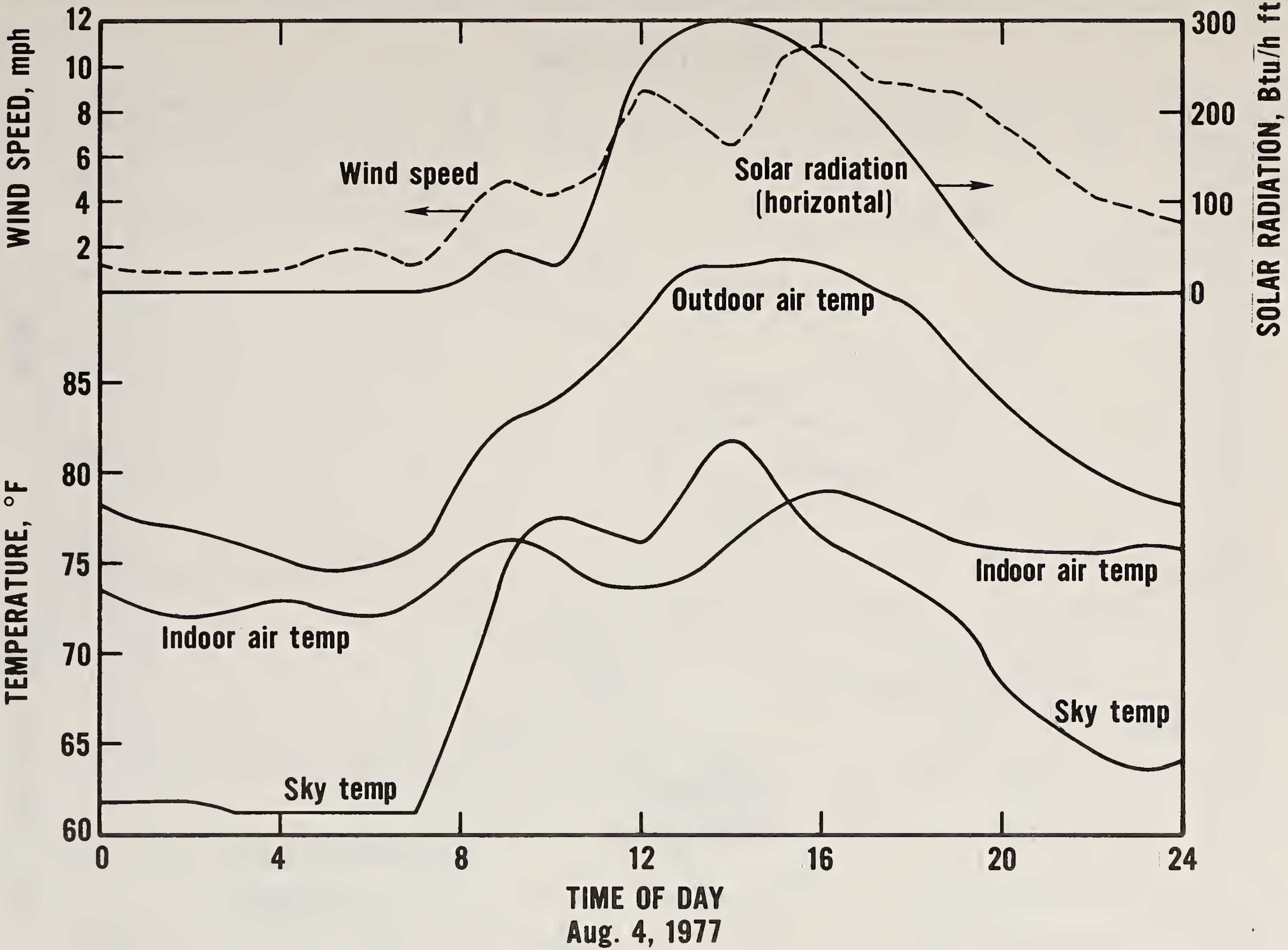


Figure 4. Weather data for August 4, 1977.

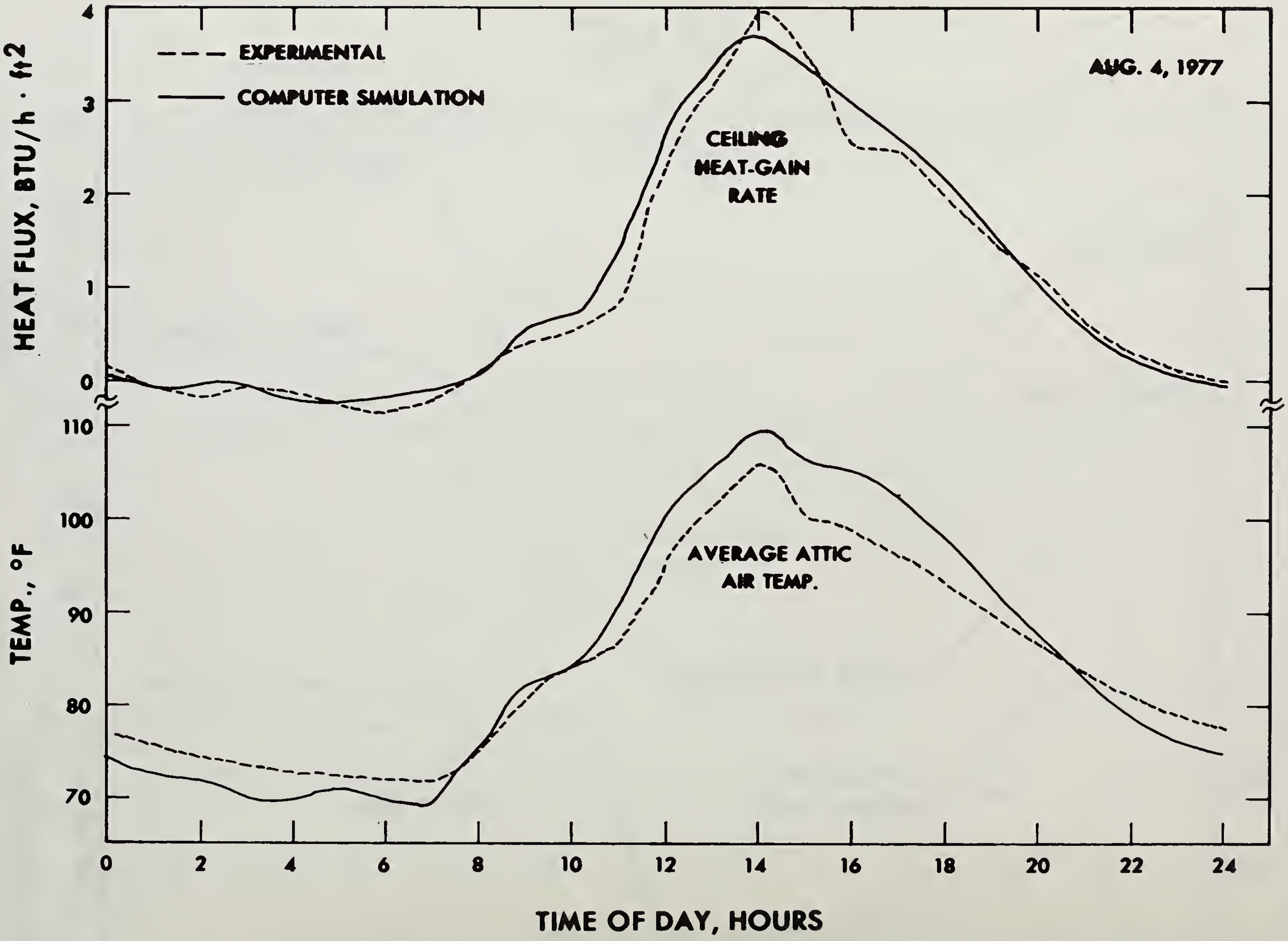


Figure 5. Comparison of ceiling heat flux and average attic air temperature for August 4, 1977.

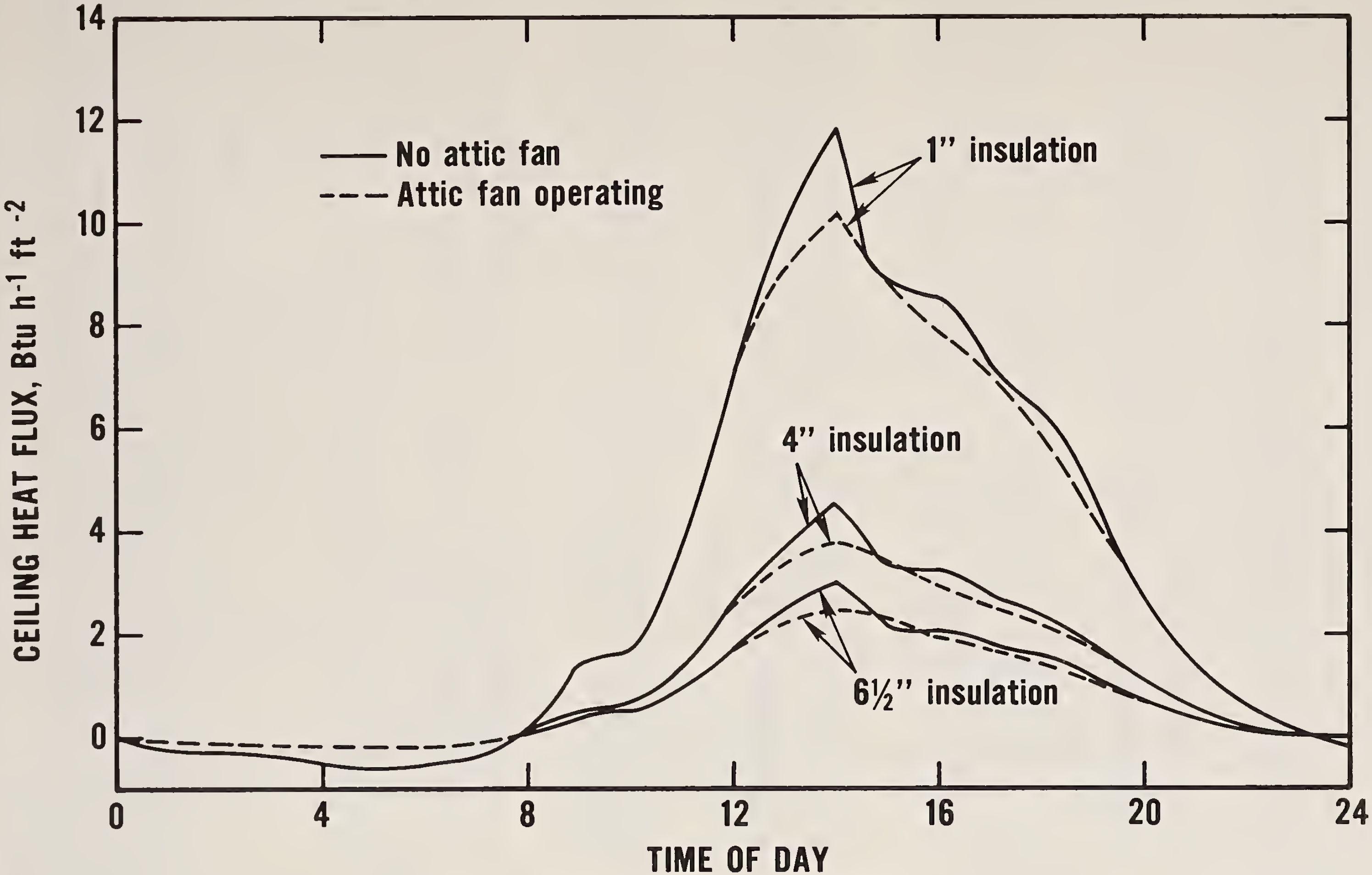


Figure 6. Ceiling heat gains for insulation thicknesses of 1, 4, and 6 1/2 inches.

The rate at which ventilation air flows through the attic by natural forces is heavily dependent upon the area of openings (soffit vents, cracks and openings in roof) through which the air may enter and leave the attic. Of interest is the effect of decreasing the natural attic air ventilation rate as a percentage of that given by equation (17). Table 2 gives the daily average ceiling and air conditioner supply duct heat gains for various percentages of the natural attic air ventilation rates.

Table 1. Effect of Ceiling Insulation Thickness on Computed Daily Average Ceiling and Duct Heat Gain Rates for Cases With and Without Power Vent Fan

	No Fan Operation			Fan-On Operation*		
	Heat Flux Btu $h^{-1}ft^{-2}$		Fan On-Time	Heat Flux Btu $h^{-1}ft^{-2}$		Percent Decrease in
Insulation Thickness, in.	Ceiling	AC Ducts	h	Ceiling	AC Ducts	Heat Flow
None	6.22	.434	8.9	6.04	.451	2.4
1	3.02	.476	8.9	2.85	.466	5.1
4	1.15	.486	9.0	1.07	.472	6.0
6 1/2	.77	.489	9.1	.70	.473	6.3

* Fan-on operation assumes power vent fan is turned on only when the leaving attic air temperature exceeds 100°F, and fan is turned off when attic air temperature goes below 85°F

Table 2. Effect of Restricting the Attic Ventilation Air Flow Rate on Computed Daily Average Ceiling and Duct Heat Flow-Rates for Cases With and Without Power Vent Fan

Percentage of	No Fan Operation			Fan-On Operation*		
Natural Ventilation	Heat Flux Btu $h^{-1}ft^{-2}$		Fan On-Time	Heat Flux Btu $h^{-1}ft^{-2}$		Percent Decrease in
Air Flow Rate Eq. (17)	Ceiling	AC Ducts		Ceiling	AC Ducts	Heat Flow
100	1.154	.486	9.00	1.069	.472	6.0
75	1.208	.493	9.14	1.070	.468	9.6
50	1.275	.505	9.27	1.070	.464	13.8
25	1.379	.524	9.38	1.067	.458	19.9
0	1.509	.557	9.43	1.070	.456	26.1

* Fan-on operation assumes power vent fan is turned on when the leaving attic air temperature exceeds 100°F, and fan is turned off when leaving attic air temperature goes below 85°F.

These results show that the effectiveness of power venting for reducing ceiling and duct heat gain rates improves as the natural ventilation air flow rates are reduced, such as by reducing the area through which air may enter the attic. The increased effectiveness of power venting as the natural ventilation rate is decreased is because the attic temperature becomes hotter and the increase in attic ventilation by power venting allows more heat to be transferred to the air.

Figure 4 shows a considerable variation in the outdoor air wind speed from less than 1 mph during nighttime hours to greater than 10 mph at 1600 (4 pm). For all data taken at Houston, the daytime wind speeds were nearly always greater than those occurring during the nighttime hours or similar to that shown in figure 4. It is quite obvious that when higher wind speeds occur during the time when the heat input to the attic is the highest, natural ventilation will remove more heat from the attic than at lower wind speeds. For this reason, the computer program was also used to analyze the effect of the variation of a constant outdoor wind speed on the daily average ceiling and duct heat gain rates for cases with and without power vent fan, as applied to a ceiling with 4 inches of insulation and wind direction from the west. The results of this analysis are summarized in Table 3.

The results show that as the outdoor wind speed is reduced from 5 to zero mph, the percent decrease in combined heat flow achieved by using a power vent fan increases from 7.2 to 31.5 percent. As the wind speed is reduced, less absorbed solar radiation is transferred from the roof by the wind, less air flows through the attic, and the temperature within the attic space becomes higher. Under such conditions, a power vent fan will produce larger percent reductions in the attic air temperature.

Table 3. Effect of Outdoor Wind Speed on the Computed Daily Average Ceiling and Duct Heat Gain Rates With and Without Power Vent Fan

Wind Speed mph	No Fan Operation — Heat Flux But $h^{-1}ft^{-2}$ Ceiling	No Fan Operation — Heat Flux But $h^{-1}ft^{-2}$ AC Ducts	Fan On-Time h	Fan-On Operation* — Heat Flux Btu $h^{-1}ft^{-2}$ Ceiling	Fan-On Operation* — Heat Flux Btu $h^{-1}ft^{-2}$ AC Ducts	Percent Decrease in Heat Flow
5.0	1.295	.531	9.2	1.201	.493	7.2
3.75	1.374	.542	9.3	1.214	.513	9.9
2.5	1.505	.561	9.5	1.240	.509	15.4
1.25	1.720	.590	9.9	1.280	.499	23.0
0	2.073	.635	10.1	1.378	.478	31.5

* Fan turned on at 100°F and off at 85°F.

7. Effectiveness of Power Vent Fan Operation

The computer program given in the appendix was designed to determine the thermal performance of ventilated attic spaces. In addition to these data, it would be of interest to determine the energy necessary to operate air conditioning equipment in order to absorb the combined heat gains from the ceiling and the attic placed air conditioning air supply ducts. Specifically, the reduction in energy required to operate the air conditioning equipment may be found from the relation

$$q = \frac{24A_{co}\ \varepsilon(Q_{ci} + Q_d)}{100\ (COP)}$$

where ε = percent decrease in heat flow (Tables 1, 2 and 3)

COP = coefficient of performance of the air conditioning equipment,

and the daily average heat flux values for no fan operation are given in Tables 1, 2, and 3. For the Houston test, the attic floor area A_{co} = 1021 ft^2, and the daily average coefficient of performance was determined to be 1.87. In addition, the power vent fan requires 969 Btu h^{-1} (284 W) to operate.

For values given in Tables 1, 2, and 3, the daily reduction in energy required to operate the air conditioning equipment due to power vent fan operation and the daily energy required to operate the fan are given below:

Table 1a

Insulation thickness, in	0	1	4	6 1/2
Daily reduction in energy to operate AC unit, Btu	2093	2336	1289	1039
Daily energy consumption of power vent fan, Btu	8624	8624	8721	8818

Table 2a

Percentage of Natural Ventilation, Eq. (17)	100	75	50	25	0
Daily reduction in energy to operate AC unit, Btu	1289	2140	3219	4962	7066
Daily energy consumption of power vent fan, Btu	8721	8856	8982	9089	9138

Table 3a

Outdoor wind speed, mph	5	3.75	2.5	1.24	0
Daily reduction in energy to operate AC unit, Btu	1723	2486	4169	6962	11178
Daily energy consumption of power vent fan, Btu	8917	9014	9208	9596	9790

8. Conclusion

A computer model for predicting the thermal performance of ventilated attics was developed. This model was used to simulate the performance of the attic in a Houston test house [3]. Computer-predicted attic air temperatures and ceiling heat gain rates were shown to be in close agreement with corresponding measured values. With the very good agreement between predicted and measured attic thermal performance, it can be reasonably expected that the model would adequately predict the thermal performance for a wide variation in operating condition, climate and geographical location. This would of course depend upon the reliability of the input information such as natural ventilation air flow rates for attic configurations, sky temperature, path length of ventilation air flow in attic, and absorptivity of roof surfaces.

Using the computer model, Houston weather, and the characteristics of the Houston residences, predictions of the thermal performance were made to evaluate the effectiveness of power vent fan use compared to attic air flow by natural ventilation, as affected by ceiling insulation thickness, restriction of attic natural ventilation air flow rates, and variations in outdoor wind speed.

When the ceiling insulation thickness was increased from zero to 6.5 inches, the percent reduction in the combined daily average ceiling temperature and direct heat gain by use of a power vent fan increased from 2.4 to 6.3 percent as shown in Table 1.

Tables 2 and 3 show that the percent reduction in the combined daily average ceiling and duct heat gain by use of a power fan increased with a decrease in wind speed.

From the computed thermal performance predictions, employing the weather data and the attic data characteristic to Houston, the coefficient of performance of the air conditioning equipment and the energy required to operate the power vent fan at the Houston test house [3], the daily reduction in the energy required to operate the air conditioning equipment, and the daily energy consumption of the power vent fan were predicted. Except for the case where the outdoor wind speed is reduced to zero (Table 3a), the daily energy consumption of the power vent fan exceeds the daily reduction in the energy required to operate the air conditioning equipment.

References

1. ASHRAE Handbook of Fundamentals, ASHRAE, New York, N.Y. 10017 pp. 20.11, 2.12-13, 22.12, 1977.

2. F. A. Joy, Improving Attic Space Insulating Values, ASHVE Transactions, Vol. 64, pp. 251-263, 1958.

3. D. M. Burch and S. J. Treado, Ventilating Residences and Their Attics for Energy Conservation--An Experimental Study, Proc. Attic Ventilation Workshop, NBS Spec. Publ. 548, 1979.

4. T. Kusuda, NBSLD, The Computer Program for Heating and Cooling Loads in Buildings, NBS Building Science Series 69, U.S. Government Printing Office, Washington, D.C., 1976.

5. G. V. Parmelee and R. G. Huebscher, Forced Convection Heat Transfer from Flat Surfaces, ASHVE Transactions, Vol. 53, pp. 245-284, 1947.

6. W. H. McAdams, Heat Transmission, Second Edition, p. 59, McGraw-Hill Book Co., Inc., New York, 1942.

7. R. W. Bliss, Atmospheric Radiation Near the Surface of the Ground, Solar Energy, Vol. 5, No. 3, p. 103, 1961.

Nomenclature

A_{co} = area of attic floor

A_d = area of attic air supply ducts

A_{ri} = area of inside roof surface

B, C = constants defined for equation (6)

c = specific heat of air, Btu $lb^{-1}F^{-1}$

CR = common ratio for use with response factors

D = Wind direction as measured from south, degrees

h_c = convection conduction component of surface coefficient of heat transfer, Btu h^{-1} $ft^{-2}F^{-1}$

h_i = surface coefficient of heat transfer at ceiling

h_o = surface coefficient of heat transfer at roof outside

h_r = radiation component of surface coefficient of heat transfer

I_H = incident solar radiation on horizontal surface, Btu $h^{-1}ft^{-2}$

L = length of attic, ft

Q_a = heat flux to attic air, Btu $h^{-1}ft^{-2}$

Q_{ci} = heat flux at ceiling

Q_{co} = heat flux at attic floor

Q_d = heat flux at attic air supply duct surface

Q_{ri} = heat flux at inside surface of roof

Q_{ro} = heat flux at outside surface of roof

R = air flow path length through attic, ft

S = long-wave sky radiation, Btu $hr^{-1}ft^{-2}$

T = temperature, F

V = air velocity, mph

V_a = air flow rate per unit attic floor area, ft h^{-1}

W = width of attic (eave to eave), ft

W_o = outdoor wind speed, mph

X,Y,Z= response factors for either roof or ceiling construction

α = absorptance of outside surface of roof

ε = emittance of attic surfaces

ρ = density of air, lb ft^{-3}

CONVERSION FACTORS TO METRIC (SI) UNITS

Physical Quantity	Symbol	To Convert From	To	Multiply By
Length	ℓ	ft	m	3.05×10^{-1}
Area	A	ft^2	m^2	9.29×10^{-2}
Volume	$\underline{V}$	ft^3	m^3	2.83×10^{-2}
Temperature	T	Fahrenheit	Celsius	$t_c = (t_f-32)/1.8$
Temp. Diff.	ΔT	Fahrenheit	Kelvin	$K = (\Delta T_F)/1.8$
Mass		lb	kg	4.54×10^{-1}
Density	ρ	lb/ft^3	kg/m^3	1.602×10^{1}
Thermal Conductivity	k	$Btu \cdot in/h \cdot ft^2 \cdot °F$	$W/m \cdot K$	1.442×10^{-1}
Coefficient of heat transfer	h	$Btu/h \cdot ft^2 \cdot °F$	$W/m^2 \cdot K$	5.68
Heat Flux	Q	$Btu/h \cdot ft^2$	W/m^2	3.15
Heat Flow	q	Btu/h	W	2.93×10^{-1}
Volumetric Flow Rate	v	ft^3/min	m^3/s	4.72×10^{-4}
Velocity	V	ft/min	m/s	5.08×10^{-3}
Specific Heat	c	$Btu/lb \cdot °F$	$J/kg \cdot K$	4.19×10^{3}

APPENDIX

COMPUTER PROGRAM FOR PREDICTING THERMAL PERFORMANCE OF VENTILATED ATTICS

```
      PARAMETER I=72
      DIMENSION A(I),B(I),C(I),D(I),E(I),F(I),G(I),H(I),TS(I),TO(I),TI(I
     A),HO(I),TOA(I),TIA(I),WI(I),SR(I),SK(I),DI(I),CR(2),X(10,2),Y(10,2
     B),Z(10,2),V(I),DM(I)
      COMMON /ATZ/CR1,CR2,VA1,VA2,TA1,TA2,SX,SY,SZ,SW,AL,W,R,J
      COMMON /ATY/CM(I),RE(3,6),M,XT
      KM=6
C   INPUT TO PROGRAM-
C
C     TOA= OUTDOOR AIR TEMP
C     TIA= INDOOR AIR TEMP
C     WI   WIND SPEED
C     DI   WIND DIRESTION
C     SR   SOLAR RADIATIOD-HORIZ SURFACE
C     SK   SKY TEMP (LONG-WAVE RADIATION)
C     DM   PERCENT ATTIC FAN ON TIME IN ONE HOUR
C     CM   PERCENT AIR-CONDITIONER ON TIME
C
C     FOR HEAT CONDUCTION IN ROOF (N=1) AND CEILING (N=2)
C     BA,CR(N)  BA IS NUMBER OF CONDUCTION TRANSFER FUNCTIONS
C               CR(N) IS HEAT FLUX COEFFICIENT
C     X(M,N),Y(M,N),Z(M,N)- CONDUCTION TRANSFER FUNCTIONS M=I,2,  ,BA
C     AL = ATTIC LENGTH
C     W  = ATTIC WIDTH
C     P  = ROOF PITCH ANGLE
C     Q  = ROOF HEIGHT
C     R  = PATH LENGTH OF AIR FLOW
C     VA1= AIR FLOW RATE PER UNIT ATTIC FLOOR AREA - NATURAL AIR FLOW
C     VA2= (FORCED AIR FLOW)
C     TA1= TEMPERATURE AT WHICH FORCED AIR FAN IS TURNED ON
C     TA2= TEMPERATURE AT WHICH FORCED AIR FAN IS TURNED OFF
C     EI = EMITTANCE OF UNDERSIDE OF ROOF
C     E2 = EMITTANCE OF ATTIC FLOOR SURFACE
      READ (5,I) (TOA(N),N=1,I)
      READ (5,I) (TIA(N),N=1,I)
      READ (5,1) (WI(N),N=1,I)
      READ (5,I) (SR(N),N=1,I)
      READ (5,1) (SK(N),N=1,I)
      READ (5,I) (DI(N),N=I,I)
      READ (5,1) (DM(N),N=I,I)
      READ (5,1) (CM(N),N=I,I)
    1 FORMAT (12F6.0)
      DO 2 N=I,I
      K=I-N+1
      HO(K)=2.2+WI(N)*(.32+.001*WI(N))
      TI(K)=TIA(N)
      TO(K)=TOA(N)
      AA=(TOA(N)+460.)/100.
      AB=(SK(N)+460.)/100.
      AA=.I714*(AA**4-AB**4)
      TS(K)=TO(K)+(.9*SR(N)-AA)/HO(K)
      A(K)=(5.*TS(K)+TI(K))/6.
```

```
      B(K)=(4.*TS(K)+TI(K))/5.
      C(K)=(2.*TS(K)+TI(K))/3.
      D(K)=(C(K)+2.*TI(K))/3.
    2 E(K)=(TS(K)+TO(K)+C(K))/3.
      L=I
      DO 3 N=1,2
      READ (5,4) BA,CR(N)
      K=BA
      READ (5,4) (X(M,N),M=1,K)
      READ (5,4) (Y(M,N),M=1,K)
    3 READ (5,4) (Z(M,N),M=1,K)
    4 FORMAT (8F10.0)
      DO 21 N=1,3
   21 READ (5,4) (RE(N,M),M=1,5)
      CR1=CR(1)
      CR2=CR(2)
    5 CALL ATRC
      SZ=1.147
      SJ=.45
      XY=R
      RE(2,6)=3.*SQRT((W/2.)**2+(AL/4.)**2)
   12 M=1
    6 N=1
      J=0
C   A= OUTSIDE ROOF SURFACE TEMP
C   B= INSIDE ROOF SURFACE TEMP
C   C=ATTIC FLOOR TEMP
C   D= CEILING TEMP
C   E= ATTIC AIR TEMP (AVERAGE)
C   F= ATTIC AIR LEAVING TEMP
C   TS= SOL-AIR TEMPERATURE
C   TO= OUTSIDE AIR TEMPERATURE
C   TI= INSIDE AIR TEMPERATURE
C   HO= ROOF SURFACE COEFFICIENT OF HEAT TRANSFER
C   G= CEILING HEAT FLUX
C   H= HEAT FLUX- LEAVING AIR  BTU/HR-SQUARE FOOT OF ATTIC FLOOR AREA
    7 AA=A(L)
      AB=B(L)
      AC=C(L)
      AD=D(L)
      AE=E(L)
      AF=F(L)
      AG=G(L)
      AH=H(L)
      AI=TS(L)
      AJ=TO(L)
      AK=TI(L)
      AM=HO(L)
      AP=CM(L)
      DO 8 IA=2,L
      K=L-IA+2
      A(K)=A(K-1)
      B(K)=B(K-1)
      C(K)=C(K-1)
      D(K)=D(K-1)
      E(K)=E(K-1)
      F(K)=F(K-1)
      G(K)=G(K-1)
      H(K)=H(K-1)
      TS(K)=TS(K-1)
```

```
      TI( K )=TI( K-1 )
      TO( K )=TO( K-1 )
      V( K )=V( K-1 )
      CM( K )=CM( K-1 )
    8 HO( K )=HO( K-1 )
      A( 1 )=AA
      B( 1 )=AB
       C( 1 )=AC
      D( 1 )=AD
      E( 1 )=AE
      F( 1 )=AF
      G( 1 )=AG
      H( 1 )=AH
      TS( 1 )=AI
      TO( 1 )=AJ
      TI( 1 )=AK
      HO( 1 )=AM
      CM( 1 )=AP
      AA=( TO( 1 )+A( 1 ))/2.+460.
      AC=46.5124-WI( N )**2
      AD=.22
      AE=A( 1 )-TOA( N )
      IF (AE.LT..0001 ) AD=.1
      IF (AC.LT..0001 ) AD=.0
      AB=AC*AD*ABS( AE )**.3333/46.5124+.3725*WI( N )**.8
      HO( 1 )=AB+.00617*( AA/100. )**3
      AA=( TOA( N )+460. )/100.
      AB=( SK( N )+460. )/100.
      IF ( SR( N ).LT.2. ) AA=(( TOA( N )+A( 1 ))/2.+460. )/100.
      AA=.1714*( AA**4-AB**4 )
      TS( 1 )=TOA( N )+( .85*SR( N )-AA )/HO( 1 )
      AA=.087+.131*( ABS( SIN( .017453*DI( N )))) )**2.5
      VA1 =SJ*AA*WI( N )
      VA2=VA1/4.+1.188
      RE( 1,6 )=XY
      IF ( DM( N ).GT.7.AND.TA1.GT.150. ) VA1 =VA1/4.+1.188*DM( N )/100.
      IF ( DM( N ).GT.7.AND.TA1.GT.150. ) RE( 1,6 )=RE( 2,6 )
      CALL ATTRC( TS,TO,TI,A,B,C,D,E,F,G,H,HO,X( 1,1 ),Y( 1,1 ),Z( 1,1 ),X( 1,2 )
     A,Y( 1,2 ),Z( 1,2 ),KM )
      NN=1-N+1
      V( 1 )=RE( 3,6 )
      TIA( NN )=XT
      N=N+1
      IF (N.LE.I) GO TO 7
      M=M+1
      IF (M.LE.4) GO TO 6
      WRITE (6,10)
      WRITE (6,9)( TO( K ),TS( K ),A( K ),B( K ),C( K ),D( K ),E( K ),F( K ),TI( K ),G( K ),
     AH( K ),TIA( K ),HO( K ),V( K ),K=33,10,-1 )
    9 FORMAT ( 9F6.1,2F8.2,F8.4,F6.2,F6.3 )
      GO TO 5
   10 FORMAT ( 1H1 )
      END
```

```
      SUBROUTINE ATTRC(TS,T0,TI,A,B,C,D,E,F,G,H,H0,X1,Y1,Z1,X2,Y2,Z2,N)
      DIMENSION A(1),B(1),C(1),D(1),E(1),F(1),G(1),H(1),TS(1),T0(1),TI(1
     1 ),H0(1),X1(1),Y1(1),Z1(1),X2(1),Y2(1),Z2(1)
      COMMON /ATZ/CR1,CR2,VA1,VA2,TA1,TA2,SX,SY,SZ,SW,AL,W,R,J
      DIMENSION TX(6)/6*75./,TY(6)/6*75./
      COMMON /ATY/CM(72),RE(3,6),KK,XT
      ST=.02866
      AX=.1*VA2
      X=75.
      IN=0
    9 IF (J.NE.0) GO TO 2
      IF (F(1).LT.TA1) GO TO 1
      J=1
    1 VA=VA1
      GO TO 3
    2 IF (J.GT.1) GO TO 6
      J=2
      VA=VA2
      GO TO 3
   20 FORMAT(6F12.3)
    6 IF (F(1).GT.TA2) GO TO 21
      J=3
      VA=VA-AX
      IF (VA.GT.VA1) GO TO 3
      J=0
      VA=VA1
      GO TO 3
   21 IF (J.NE.3) GO TO 3
      J=2
    3 BA=(B(1)+C(1))/2.+460.
      IF (J.EQ.2) VA=VA2
      R=RE(1,6)
      IF (J.GT.1) R=RE(2,6)
      HR2=SX*(BA/100.)**3
      HR1=SZ*HR2
      V=W**2*VA/SW
      IF (V.LT..0001) V=.0001
      V=.018475*V*(V*W/2.)**(-.2)
      BA=B(1)-E(1)
      BD=ABS(BA)**.333
      BE=.26125/(1.375+SY)
      IF (BA.LT..0) BE=1.393/(7.333-SY)
      HC1=BE*BD+V
      BA=C(1)-E(1)
      BD=ABS(BA)**.333
      BE=.22
      IF (BA.LT..0) BE=.1
      HC2=BE*BD+V
      BE=(D(1)+TI(1))/2.+460.
      BF=.00617*(BE/100.)**3
      BE=ABS(D(1)-TI(1))**.3333
      HI=.11*BE+BF
      IF (D(1).LT.TI(1)) HI=.22*BE+BF
      BA=HC2+HC1*SZ+ST*CM(1)/100.
      BD=.02044-.0000334*E(1)
      BC=60.*BD*VA
```

```
      BB=BC*W/(R*BA)
      IF (BB.LT..001) BB=.03
      C3=BB*(1.-EXP(-1./BB))
      C2=HC1*SZ*(1.-C3)/BA
      C1=HC2*(1.-C3)/BA
      C4=ST*CM(1)*(1.-C3)/(100.*BA)
      SA=HO(1)*TS(1)+(Z1(1)-Y1(1))*X+CR1*QO
      SB=(X1(1)-Y1(1))*X-CR1*QI
      SC=(X2(1)-Y2(1))*X+HI*TI(1)-CR2*QCI
      SD=(Y2(1)-Z2(1))*X-CR2*QCO
      DO 4 I=2,N
      SA=SA+Y1(I)*(B(I)-X)-Z1(I)*(A(I)-X)
      SB=SB+Y1(I)*(A(I)-X)-X1(I)*(B(I)-X)
      SC=SC+Y2(I)*(C(I)-X)-X2(I)*(D(I)-X)
    4 SD=SD+Z2(I)*(C(I)-X)-Y2(I)*(D(I)-X)
      SB=SB+Y1(1)*SA/(HO(1)+Z1(1))
      SD=SD-Y2(1)*SC/(HI+X2(1))
      RA=X1(1)+HR1+HC1-HC1*C2-Y1(1)**2/(HO(1)+Z1(1))
      RB=HR1+C1*HC1
      RC=HR2+C2*HC2
      RD=Z2(1)+HR2+HC2-HC2*C1-Y2(1)**2/(HI+X2(1))
      RH=RB*RC-RA*RD
      RF=SB+HC1*(C3*TO(1)+C4*55.)
      RG=SD-HC2*(C3*TO(1)+C4*55.)
      B(1)=(RG*RB-RF*RD)/RH
      C(1)=(RG*RA-RF*RC)/RH
      E(1)=C1*C(1)+C2*B(1)+C3*TO(1)+55.*C4
      BA=(HC2*C(1)+HC1*SZ*B(1)+.55*ST*CM(1))/BA
      F(1)=BA+(TO(1)-BA)*EXP(-1./BB)
      A(1)=(SA+Y1(1)*B(1))/(HO(1)+Z1(1))
      D(1)=(SC+Y2(1)*C(1))/(HI+X2(1))
      IN=IN+1
      IF (IN.GT.12)GO TO 7
      IF (J.GT.1) GO TO 8
      IF (F(1).LT.TA1+2.) GO TO 7
      VA=VA+AX
      GO TO 3
    8 IF (J.EQ.3) GO TO 6
    7 QO=QO*CR1
      QI=QI*CR1
      QCO=QCO*CR2
      QCI=QCI*CR2
      DO 5 K=1,N
      QO=QO+Y1(K)*(B(K)-X)-Z1(K)*(A(K)-X)
      QI=QI+X1(K)*(B(K)-X)-Y1(K)*(A(K)-X)
      QCO=QCO+Y2(K)*(D(K)-X)-Z2(K)*(C(K)-X)
    5 QCI=QCI+X2(K)*(D(K)-X)-Y2(K)*(C(K)-X)
      RE(3,6)=VA
      XT=CM(1)*ST*(E(1)-55.)/100.
      H(1)=BC*(F(1)-TO(1))
      SA=HI*TI(1)+(RE(1,1)-RE(2,1))*X
      SB=HC2*E(1)+HR2*B(1)+(RE(3,1)-RE(2,1))*X
      DO 30 K=2,5
      SA=SA+RE(2,K)*(TY(K)-X)-RE(1,K)*(TX(K)-X)
   30 SB=SB+RE(2,K)*(TX(K)-X)-RE(3,K)*(TY(K)-X)
      SC=RE(3,1)+HC2+HR2
      SD=SC*(RE(1,1)+HI)-RE(2,1)**2
      TX(1)=(SA*SC+SB*RE(2,1))/SD
      TY(1)=(SB+RE(2,1)*TX(1))/SC
      IF (KK.GT.3) WRITE (6,32) HC2,HR2,HI,G(1),BB,HC1,HR1,TX(1),TY(1)
```

```
32 FORMAT (10F12.4)
   G(1)=HI*(TX(1)-TI(1))
   DO 31 K=2,6
   I=7-K
   TX(I+1)=TX(I)
31 TY(I+1)=TY(I)
   RETURN
   END

   SUBROUTINE ATRC
   COMMON /ATZ/CR1,CR2,VA1,VA2,TA1,TA2,SX,SY,SZ,SW,AL,W,R,J
   READ (5,3) AL,W,P,Q,R,VA1,VA2,TA1,TA2,E1,E2
 3 FORMAT(12F6.0)
   IF (AL.LT.1.) STOP
   SZ=1.
   P=.017453292*P
   SY=COS(P)
   IF (P.GT..01) GO TO 1
   SW=Q*W
   SA=AL/Q
   SB=W/Q
   F=FP(SA,SB)
   GO TO 2
 1 SA=W/(2.*SY)
   SW=Q*W/2.
   SB=W/AL
   SZ=2.*SA/W
   SA=SA/AL
   F=FMN(SA,SB,P)
   F=2.*F/SZ
 2 SB=(1.-SZ*F*F)/(1.+SZ*(1.-2.*F))
   SA=1./E1-1.+SZ*(1./E2-1.)+1./SB
   SX=.00686/SA
   WRITE (6,5) SX,SY,SZ,SW
 5 FORMAT (6E12.6)
   RETURN
   END

   FUNCTION FP(X,Y)
C  ANGLE FACTORS FOR TWO PARALLEL PLATES - SAME SIZE   DIRECT VIEWING
C    X=A/C, Y=B/C  C=DISTANCE BETWEEN - A AND B DIMENSIONS OF PLATES
   A=SQRT(1.+X*X)
   B=SQRT(1.+Y*Y)
   C=SQRT(1.+X*X+Y*Y)
   D=ALOG(A*B/C)-Y*ATAN(Y)-X*ATAN(X)+Y*A*ATAN(Y/A)+X*B*ATAN(X/B)
   FP=2.*D/(3.14159265*X*Y)
   RETURN
   END

   FUNCTION FMN(X,Y,P)
C  ANGLE FACTORS FOR TWO PLATES MEETING AT AN ANGLE, P    F(1-2)
C   X=A/B, Y=C/B  B=LENGTH OF COMMON SIDE, C=LENGTH SIDE OF PLATE 1
C    A=LENGTH SIDE OF PLATE 2
   DIMENSION Q(65)
   A=SIN(P)
   B=COS(P)
   Z=SQRT(X*X+Y*Y-2.*X*Y*B)
   C=Y/64.
   DO 1 N=1,65
   D=C*(N-1)
```

```
      E=SQRT(1.+D*D*A*A)
      F=X*E/(1.+D*D-D*X*B)
    1 Q(N)=E*ATAN(F)
      D=Q(1)+Q(65)+4.*Q(64)
      DO 2 N=1,31
      K=2*N
    2 D=D+4.*Q(K)+2.*Q(K+1)
      D=B*C*D/3.
      C=SQRT(1.+X*X*A*A)
      E=Y*C/(1.+X*X-X*Y*B)
      D=D+Y*ATAN(1./Y)+X*ATAN(1./X)-Z*ATAN(1./Z)
      C=1.+X*X
      E=1.+Y*Y
      F=1.+Z*Z
      E=(1.+B*B)*ALOG(C*E/F)/(A*A)+Y*Y*ALOG(Y*Y*F/(Z*Z*E))
      E=E+X*X*(2.*ALOG(X/Z)+(2.*B*B-1.)*ALOG(C/F))
      D=D+A*A*E/4.
      E=Y*Y*ATAN((X-Y*B)/(Y*A))+X*X*ATAN((Y-X*B)/(X*A))
      C=X*Y*A+(X*X+Y*Y)*(1.5707963-P)
      FMN=(D-A*B*(C+E)/2.)/(3.1415926*Y)
      RETURN
      END
```

```
*** SAMPLE PROBLEM
    (STATEMENTS IN PARENTHESES ARE FOR INFORMATION PURPOSES ONLY)

(**INPUT DATA - FOR PERIOD FROM 1000, 8/2/77  TO  0900, 8/5/77 )
      (HOURLY DATA FROM HOUSTON TESTS- HOUSE NO 2)
   (OUTDOOR AIR TEMPERATURE- F)
87.4  88.7  90.4  92.2  91.8  96.8  97.7  94.5  88.7  87.7  84.9  82.2
81.6  80.0  79.5  78.4  76.6  76.4  77.0  74.0  75.4  75.0  80.2  86.2
89.1  92.6  92.2  95.2  95.3  93.5  92.6  93.0  88.8  86.9  83.7  81.1
80.1  79.2  78.3  77.2  76.8  76.0  75.2  74.4  75.0  75.7  79.9  82.9
83.7  85.8  88.8  91.3  91.3  91.8  91.5  90.3  89.2  86.3  83.8  81.6
80.2  79.0  78.2  78.2  77.2  77.3  77.2  76.4  76.9  77.0  78.3  83.5
   (INDOOR AIR TEMPERATURE- F)
73.8  74.1  73.6  73.5  73.8  74.4  74.9  75.0  75.1  74.6  73.8  72.8
72.1  71.8  72.3  73.0  73.1  72.0  71.7  72.4  73.4  74.0  73.7  73.5
73.5  73.9  74.1  74.0  74.4  75.0  75.6  76.2  76.4  75.8  74.4  73.0
72.0  72.3  73.2  73.4  72.5  71.9  72.5  73.2  72.5  71.9  72.9  75.3
76.2  75.3  73.8  73.6  74.1  76.3  77.9  79.1  78.2  77.2  76.2  75.9
75.5  75.5  76.1  75.8  74.8  75.7  74.7  75.0  74.8  75.6  74.9  74.5
   (WIND SPEED- MPH)
10.4  9.3   8.2   8.0   8.3   8.1   7.9   7.8   7.4   8.3   5.9   5.0
2.9   3.0   1.7   1.8   2.6   1.1   1.8   1.9   2.4   1.9   1.5   3.1
5.4   4.7   5.9   6.2   6.4   9.4   8.2   10.0  10.0  10.7  9.0   5.1
3.6   3.5   1.3   .96   .96   .99   .96   1.7   1.8   1.2   3.0   4.9
4.3   5.1   9.0   8.0   6.4   10.3  10.9  9.6   9.2   8.9   7.4   5.7
4.2   3.7   3.1   3.3   3.4   2.9   3.1   1.9   1.4   1.2   1.3   5.9
   (INCIDENT SOLAR RADIATION- BTU/H-FT2)
165.  206.  210.  229.  241.  259.  233.  155.  52.   71.   27.1  0.6
                                                      1.9   30.   98.
```

```
                                                                    0.9    10.    47.
31.    97.    254.   291.   302.   285.   255.   214.   153.   85.    23.    0.6
       0.3                                                      0.6    7.2    86.
   (EQUIVALENT SKY TEMPERATURE- F)
57.6   74.6   78.4   80.2   81.2   81.2   81.2   80.2   78.4   71.7   67.8   64.8
63.8   63.8   63.8   62.4   62.9   61.8   62.4   60.7   60.1   60.1   61.8   68.0
72.3   75.5   78.2   80.7   81.8   80.2   79.7   77.6   75.0   71.3   67.4   64.1
62.9   62.4   61.8   61.8   61.8   61.2   61.2   61.2   61.2   61.2   67.4   75.0
77.6   77.1   76.1   79.2   81.8   79.2   76.1   75.0   73.4   72.3   68.0   66.3
64.6   63.5   64.1   63.5   62.9   64.6   63.5   62.9   63.5   64.6   63.5   67.4
   (WIND DIRECTION- DEGREES)
315.   270.   315.   22.5   315.   337.5  292.5  45.    157.5  157.5  202.5  202.5
180.   157.5  247.5  270.   135.   135.   45.    202.5  270.   112.5  337.5  337.5
0.     22.5   22.5   0.     90.    112.5  135.   135.   180.   157.5  157.5  180.   W
180.   202.5  247.5  247.5  247.5  0.     0.     270.   0.     0.     337.5  0.
67.5   135.   135.   157.5  180.   90.    180.   135.   157.5  135.   157.5  157.5
157.5  180.   157.5  135.   112.5  157.5  180.   225.   112.5  67.5   67.5   22.5
   (PERCENT ATTIC POWER VENT FAN ON TIME)
1.1    6.7    97.8   97.8   97.8   97.8   97.8   97.8   100.   100.   88.9   0.
                                                               1.1
27.8   98.9   96.7   96.7   97.8   97.8   97.8   100.   97.8   98.9   55.7   0.

0.     0.     64.4   97.8   98.9   100.   98.9   100.   100.   100.   40.    0.

   (PERCENT AIR CONDITIONER ON TIME)
72.0   94.5   96.2   79.3   99.7   100.   99.7   95.9   93.6   92.4   89.2   61.5
57.5   54.2   41.1   38.5   45.2   41.7   35.9   32.1   31.2   31.2   27.7   57.4
71.1   96.2   97.4   98.5   97.7   97.1   97.1   95.9   93.3   91.5   88.9   74.9
57.7   49.0   40.2   42.3   43.4   35.6   31.5   31.8   37.3   36.7   0.     0.
41.4   92.4   94.2   94.8   95.9   55.7   68.5   95.6   93.3   92.1   67.1   56.6
48.1   42.6   44.6   31.8   44.0   28.3   34.4   34.7   26.8   26.2   17.5   37.3
   (RESPONSE FACTORS FOR ROOF AND ROOF RAFTERS)
6.          .50358
1.66638137-1.0761122.10516096 .000649   .00004146 .0000027
1.2681836 5-0.5289568-.043825  .000674   .00004481 .0000029
1.8359929 -1.3298108.1891261  .0007634  .00004858 .0000032
   (RESPONSE FACTORS FOR CEILING AND CEILING JOISTS)
6.          .476024
.71394474 -.92080076.26117225 .00076361 .00003286 .00000174
.04497909  .01485082-.00523625.00049011 .00002914 .00000156
.25237533 -.24160189.04379619 .00051711 .00002633 .0000014
   (CONDUCTANCE OF CEILING)
.0806665
.0806665
.0806665
   (ATTIC LENGTH,WIDTH,PITCH ANGLE,HEIGHT,AIR FLOW PATH,VA1,VA2,TA1,TA2,E1,E2)
45.58 32.25 22.6  6.7   32.25 .4    .4    250.  250.  .9    .8
45.58 32.25 22.6  6.7   32.25 .4    .4    100.   85.  .9    .8
```

Table A-1. Thermal performance prediction for attic, with attic power vent fan operating (for August 4, 1977, Houston, Texas)

Hr. of Day	Outdoor Temp	Sol-air Temp	Roof Surface Temp Outside	Roof Surface Temp Inside	Attic Floor Temp	Ceiling Temp	Attic Air Temp Avg.	Attic Air Temp Leaving	Inside Air Temp	Ceiling Ht Flux	Attic Air Flow
										$Btu/h \cdot ft^2$	cfm/ft^2
1	77.2	68.9	70.8	71.9	72.6	73.6	72.4	71.0	73.4	-.10	.084
2	76.8	68.9	70.1	71.0	71.6	72.9	71.7	69.9	72.5	-.10	.084
3	75.0	68.3	69.3	70.0	70.5	72.3	70.0	68.9	71.9	-.13	.039
4	75.2	67.9	68.7	69.3	69.8	72.3	69.2	68.2	72.5	-.22	.038
5	74.4	68.2	68.8	69.4	70.0	72.8	70.8	69.1	73.2	-.25	.167
6	75.0	68.6	68.9	69.3	69.8	72.4	69.8	68.1	72.5	-.21	.070
7	75.7	68.9	69.1	69.3	69.7	72.0	69.4	68.2	71.9	-.17	.047
8	79.9	77.6	75.0	73.6	73.1	72.5	74.7	72.6	72.9	.06	.133
9	82.9	95.3	89.3	83.9	81.2	74.4	82.0	81.5	75.3	.58	.192
10	83.7	91.8	89.6	86.7	84.3	75.8	83.9	84.1	76.2	.69	.376
11	85.8	112.9	105.4	97.2	92.3	76.1	90.2	92.7	75.3	1.40	.326
12	88.8	149.7	136.5	117.6	107.2	75.8	100.7	107.4	73.8	2.68	.909
1	91.3	164.9	152.0	130.1	116.7	76.3	106.2	115.3	73.6	3.38	1.251
2	91.3	173.5	160.6	137.6	122.9	77.2	110.1	121.3	74.1	3.79	1.238
3	91.8	154.9	149.6	133.0	119.8	79.0	107.0	116.7	76.3	3.33	1.441
4	91.5	145.5	141.5	128.3	117.0	80.5	105.7	114.4	77.9	2.98	1.282
5	90.3	138.4	134.8	123.4	113.2	81.6	102.5	110.3	79.1	2.58	1.341
6	89.2	123.2	122.4	115.5	107.7	81.2	98.8	104.9	78.2	2.20	1.290
7	86.3	104.2	106.2	104.2	99.3	80.2	92.4	96.5	77.2	1.60	1.330
8	83.8	84.8	89.7	92.6	91.3	78.8	87.9	90.1	76.2	1.05	.558
9	81.6	75.7	80.3	83.8	84.2	77.8	82.4	82.8	75.9	.53	.254
10	80.2	73.8	76.8	79.3	80.0	76.9	79.3	78.8	75.5	.26	.187
11	79.0	72.7	74.8	76.5	77.3	76.4	76.8	75.8	75.5	.07	.145
12	78.2	72.2	73.7	74.9	75.6	76.4	75.4	74.2	76.1	-.08	.138

Table A-2. Thermal performance prediction for attic, without attic fan operating (for August 4, 1977, Houston, Texas)

Hr. of Day	Outdoor Temp	Sol-air Temp	Roof Surface Temp Outside	Roof Surface Temp Inside	Attic Floor Temp	Ceiling Temp	Attic Air Temp Avg.	Attic Air Temp Leaving	Inside Air Temp	Ceiling Ht Flux	Attic Air Flow
1	77.2	68.9	70.8	71.9	72.6	73.6	72.5	71.1	73.4	-.09	.084
2	76.8	68.9	70.1	71.0	71.7	72.9	71.7	69.9	72.5	-.10	.084
3	76.0	68.3	69.3	70.0	70.6	72.3	70.0	68.9	71.9	-.13	.039
4	75.2	67.9	68.7	69.3	69.8	72.3	69.3	68.2	72.5	-.22	.038
5	74.4	68.2	68.8	69.4	70.0	72.8	70.8	69.1	73.2	-.25	.167
6	75.0	68.6	68.9	69.3	69.8	72.4	69.8	68.1	72.5	-.21	.070
7	75.7	68.9	69.1	69.3	69.7	72.0	69.4	68.2	71.9	-.17	.047
8	79.9	77.6	75.6	73.6	73.1	72.5	74.7	72.6	72.9	.06	.133
9	82.9	95.3	89.3	83.9	81.2	74.4	82.0	81.5	75.3	.58	.192
10	83.7	91.8	89.6	86.7	84.3	75.8	83.9	84.1	76.2	.69	.376
11	85.8	112.9	105.4	97.2	92.3	76.1	90.2	92.7	75.3	1.40	.326
12	88.8	149.7	136.4	117.3	106.6	75.8	99.3	105.8	73.8	2.63	.575
1	91.3	164.8	152.7	132.9	120.8	76.3	112.7	122.6	73.6	3.74	.356
2	91.3	173.3	162.2	143.7	131.7	77.5	123.7	134.9	74.1	4.52	.251
3	91.8	154.9	150.0	133.4	119.8	79.3	104.6	113.6	76.3	3.27	1.010
4	91.5	145.5	142.1	130.7	120.4	80.7	110.5	120.1	77.9	3.26	.427
5	90.3	138.4	135.3	124.8	115.0	81.8	104.0	112.3	79.1	2.71	.614
6	89.2	123.2	123.0	117.6	110.6	81.4	102.6	109.6	78.2	2.43	.409
7	86.3	104.1	106.6	105.3	100.7	80.3	93.5	98.1	77.2	1.69	.569
8	83.8	84.8	89.8	93.0	91.7	79.0	87.9	90.1	76.2	1.07	.329
9	81.6	75.7	80.4	84.0	84.4	77.9	82.5	83.0	75.9	.54	.254
10	80.2	73.8	76.8	79.3	80.2	77.0	79.4	78.9	75.5	.27	.187
11	79.0	72.7	74.8	76.6	77.4	76.4	76.8	75.8	75.5	.08	.145
12	78.2	72.2	73.7	74.9	75.7	76.4	75.4	74.2	76.1	-.07	.138

NBS-114A (REV. 9-78)

U.S. DEPT. OF COMM. BIBLIOGRAPHIC DATA SHEET	1. PUBLICATION OR REPORT NO.	2. Gov't. Accession No.	3. Recipient's Accession No.
	Spec. Publ. 548		

4. TITLE AND SUBTITLE

SUMMER ATTIC AND WHOLE-HOUSE VENTILATION

5. Publication Date

July 1979

6. Performing Organization Code

7. AUTHOR(S)

Mary H. Reppert, Editor

8. Performing Organ. Report No.

9. PERFORMING ORGANIZATION NAME AND ADDRESS

NATIONAL BUREAU OF STANDARDS
DEPARTMENT OF COMMERCE
WASHINGTON, DC 20234

10. Project/Task/Work Unit No.

742 2115

11. Contract/Grant No.

12. SPONSORING ORGANIZATION NAME AND COMPLETE ADDRESS (*Street, City, State, ZIP*)

Department of Energy
Washington, D.C. 20585

13. Type of Report & Period Covered

14. Sponsoring Agency Code

15. SUPPLEMENTARY NOTES

Library of Congress Catalog Card Number: 79-600086

☐ Document describes a computer program; SF-185, FIPS Software Summary, is attached.

16. ABSTRACT (*A 200-word or less factual summary of most significant information. If document includes a significant bibliography or literature survey, mention it here.*)

These are the proceedings of the Summer Attic and Whole-House Ventilation Workshop sponsored by the National Bureau of Standards in collaboration with the Department of Energy and the attic ventilation industry. The purpose of the Workshop was to provide a forum for technical discussion to assess summer energy savings that might be achieved from the use of static and powered attic ventilation and whole-house ventilation equipment. Papers on experimental and mathematical model studies relating to attic and whole-house ventilation were presented. In addition, a paper on roof solar absorptance and its effect on the cooling requirement of a residence was presented.

After each paper was presented, participants of the Workshop were given an opportunity to question the speaker concerning his paper. The resulting questions and responses are included at the end of each of the papers.

17. KEY WORDS (*six to twelve entries; alphabetical order; capitalize only the first letter of the first key word unless a proper name; separated by semicolons*)

Attic ventilation; energy conservation; solar absorptance of roofing materials; whole-house ventilation.

18. AVAILABILITY ☒ Unlimited	19. SECURITY CLASS (THIS REPORT)	21. NO. OF PRINTED PAGES
☐ For Official Distribution. Do Not Release to NTIS	UNCLASSIFIED	153
☒ Order From Sup. of Doc., U.S. Government Printing Office, Washington, DC 20402, SD Stock No. SN003-003-02089-3	20. SECURITY CLASS (THIS PAGE)	22. Price
☐ Order From National Technical Information Service (NTIS), Springfield, VA. 22161	UNCLASSIFIED	$4.75

USCOMM-DC

NBS TECHNICAL PUBLICATIONS

PERIODICALS

JOURNAL OF RESEARCH—The Journal of Research of the National Bureau of Standards reports NBS research and development in those disciplines of the physical and engineering sciences in which the Bureau is active. These include physics, chemistry, engineering, mathematics, and computer sciences. Papers cover a broad range of subjects, with major emphasis on measurement methodology, and the basic technology underlying standardization. Also included from time to time are survey articles on topics closely related to the Bureau's technical and scientific programs. As a special service to subscribers each issue contains complete citations to all recent NBS publications in NBS and non-NBS media. Issued six times a year. Annual subscription: domestic $17.00; foreign $21.25. Single copy, $3.00 domestic; $3.75 foreign.

Note: The Journal was formerly published in two sections: Section A "Physics and Chemistry" and Section B "Mathematical Sciences."

DIMENSIONS/NBS

This monthly magazine is published to inform scientists, engineers, businessmen, industry, teachers, students, and consumers of the latest advances in science and technology, with primary emphasis on the work at NBS. The magazine highlights and reviews such issues as energy research, fire protection, building technology, metric conversion, pollution abatement, health and safety, and consumer product performance. In addition, it reports the results of Bureau programs in measurement standards and techniques, properties of matter and materials, engineering standards and services, instrumentation, and automatic data processing.

Annual subscription: Domestic, $11.00; Foreign $13.75

NONPERIODICALS

Monographs—Major contributions to the technical literature on various subjects related to the Bureau's scientific and technical activities.

Handbooks—Recommended codes of engineering and industrial practice (including safety codes) developed in cooperation with interested industries, professional organizations, and regulatory bodies.

Special Publications—Include proceedings of conferences sponsored by NBS, NBS annual reports, and other special publications appropriate to this grouping such as wall charts, pocket cards, and bibliographies.

Applied Mathematics Series—Mathematical tables, manuals, and studies of special interest to physicists, engineers, chemists, biologists, mathematicians, computer programmers, and others engaged in scientific and technical work.

National Standard Reference Data Series—Provides quantitative data on the physical and chemical properties of materials, compiled from the world's literature and critically evaluated. Developed under a world-wide program coordinated by NBS. Program under authority of National Standard Data Act (Public Law 90-396).

NOTE: At present the principal publication outlet for these data is the Journal of Physical and Chemical Reference Data (JPCRD) published quarterly for NBS by the American Chemical Society (ACS) and the American Institute of Physics (AIP). Subscriptions, reprints, and supplements available from ACS, 1155 Sixteenth St. N.W., Wash., D.C. 20056.

Building Science Series—Disseminates technical information developed at the Bureau on building materials, components, systems, and whole structures. The series presents research results, test methods, and performance criteria related to the structural and environmental functions and the durability and safety characteristics of building elements and systems.

Technical Notes—Studies or reports which are complete in themselves but restrictive in their treatment of a subject. Analogous to monographs but not so comprehensive in scope or definitive in treatment of the subject area. Often serve as a vehicle for final reports of work performed at NBS under the sponsorship of other government agencies.

Voluntary Product Standards—Developed under procedures published by the Department of Commerce in Part 10, Title 15, of the Code of Federal Regulations. The purpose of the standards is to establish nationally recognized requirements for products, and to provide all concerned interests with a basis for common understanding of the characteristics of the products. NBS administers this program as a supplement to the activities of the private sector standardizing organizations.

Consumer Information Series—Practical information, based on NBS research and experience, covering areas of interest to the consumer. Easily understandable language and illustrations provide useful background knowledge for shopping in today's technological marketplace.

Order **above** *NBS publications from: Superintendent of Documents, Government Printing Office, Washington, D.C. 20402.*

Order **following** *NBS publications—NBSIR's and FIPS from the National Technical Information Services, Springfield, Va. 22161.*

Federal Information Processing Standards Publications (FIPS PUB)—Publications in this series collectively constitute the Federal Information Processing Standards Register. Register serves as the official source of information in the Federal Government regarding standards issued by NBS pursuant to the Federal Property and Administrative Services Act of 1949 as amended, Public Law 89-306 (79 Stat. 1127), and as implemented by Executive Order 11717 (38 FR 12315, dated May 11, 1973) and Part 6 of Title 15 CFR (Code of Federal Regulations).

NBS Interagency Reports (NBSIR)—A special series of interim or final reports on work performed by NBS for outside sponsors (both government and non-government). In general, initial distribution is handled by the sponsor; public distribution is by the National Technical Information Services (Springfield, Va. 22161) in paper copy or microfiche form.

BIBLIOGRAPHIC SUBSCRIPTION SERVICES

The following current-awareness and literature-survey bibliographies are issued periodically by the Bureau:

Cryogenic Data Center Current Awareness Service. A literature survey issued biweekly. Annual subscription: Domestic, $25.00; Foreign, $30.00.

Liquefied Natural Gas. A literature survey issued quarterly. Annual subscription: $20.00.

Superconducting Devices and Materials. A literature survey issued quarterly. Annual subscription: $30.00. Send subscription orders and remittances for the preceding bibliographic services to National Bureau of Standards, Cryogenic Data Center (275.02) Boulder, Colorado 80302.

U.S. DEPARTMENT OF COMMERCE
National Bureau of Standards
Washington, D.C. 20234

OFFICIAL BUSINESS

Penalty for Private Use, $300

POSTAGE AND FEES PAID
U.S. DEPARTMENT OF COMMERCE
COM-215

U.S.MAIL

SPECIAL FOURTH-CLASS RATE
BOOK

1298