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**Estimated Costs
of Ventilation Systems Complying
with the HUD Ventilation Standard
for Manufactured Homes**

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Summary

At the request of the U.S. Department of Housing and Urban Development (HUD), the Pacific Northwest Laboratory estimated the material, labor, and operating costs for ventilation equipment needed for compliance with HUD's proposed revision to the ventilation standard for manufactured housing. This was intended to bound the financial impacts of the ventilation standard revision.

Researchers evaluated five possible prototype ventilation systems that met the proposed ventilation requirements. Of those five, two systems were determined to be the most likely used by housing manufacturers:

- System 1 combines a fresh air duct with the existing central forced-air system to supply and circulate fresh air to conditioned spaces.
- System 2 uses a separate exhaust fan to remove air from the manufactured home.

The estimated material and labor costs for these two systems range from \$200 to \$300 per home.

Annual operating costs for the two ventilation systems were estimated for 20 U.S. cities. The estimated operating costs for System 1 ranged from \$55/year in Las Vegas, Nevada, to \$83/year in Bismarck, North Dakota. Operating costs for System 2 ranged from a low of \$35/year in Las Vegas to \$63/year in Bismarck. Thus, HUD's proposed increase in ventilation requirements will add less than \$100/year to the energy cost of a manufactured home.

Acknowledgments

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Introduction

In 1987, Congress passed legislation requiring the U.S. Department of Housing and Urban Development (HUD) to revise its energy conservation standard for manufactured housing. On February 24, 1992, HUD published a Notice of Proposed Rulemaking (NOPR) detailing its proposed changes (57 FR 6420-31).

The HUD-proposed rule significantly changed the requirements for ventilation originally specified in the Federal Manufactured Home Construction and Safety Standards (24 CFR Part 3280). Public comments on the NOPR were accepted through May 28, 1992. Subsequent review of the public comments (TenWolde and Burch 1993) led to proposed revisions to the requirements outlined in the NOPR. Basically, the most current proposed revisions require a total ventilation rate of 0.35 air change per hour (ACH) in new manufactured housing. This ventilation rate is to be achieved via a ventilation system (either mechanical or passive) having a continuous capacity of 0.035 cfm per square foot of floorspace. Operable windows are optional except in Region 4 and Alaska.

In light of these proposed revisions, HUD requested the Pacific Northwest Laboratory (PNL)^(a) to evaluate prototype ventilation systems meeting the requirements of the proposed revised standard. Consequently, PNL researchers examined five such systems for manufactured homes.

This report documents that evaluation. The next section presents descriptions of the five systems. Advantages and disadvantages of each are discussed. Material and labor costs^(b) are included for the two systems most likely to be used in manufactured housing.

Estimated operating costs for the two most likely ventilation systems are presented in the final section. Operating costs are defined as the sum of the heating and cooling costs associated with satisfying the increase in thermal load to the house, and the cost of powering the ventilation fan.

An appendix contains a change document that summarizes the differences between the latest proposed revision, the HUD proposed rule, and the original standard.

^(a)Pacific Northwest Laboratory is operated by Battelle Memorial Institute for the U.S. Department of Energy under Contract DE-AC06-76RLO 1830.

^(b)All costs presented in this report should be treated as estimates. Builders of manufactured homes may get lower prices than those quoted here because of volume buying.

Prototype System Descriptions

Five possible system approaches that meet the new ventilation requirements for manufactured homes are described here. These approaches are not meant to be exhaustive, but to serve as examples of possible systems. The ventilation systems are discussed relative to the proposed revised standard.

System 1: Fresh Air Duct to Furnace

The System 1 approach uses the existing central forced-air system to supply and circulate fresh air to all the conditioned rooms in the house. A fresh air duct is connected to the return side of the furnace fan for the fresh air supply. The system includes controls (parallel to the thermostat) to run the fan, even when heating or cooling is not needed. Either a fixed flow restrictor in the fresh air duct or an adjustable mechanical volume damper (for seasonal changes in ventilation needs) is used to control ventilation rates. Trickle ventilators (window vents) or envelope vents are needed to minimize positive pressure and associated exfiltration through the envelope (see Figure 1).

Advantages

System 1 offers three primary advantages:

- No additional fan is needed.
- The potential exists for filtering fresh air.
- Ventilation air is distributed evenly throughout the house.

Figure 1. Fresh Air Duct to Furnace

Disadvantages

- In northern climates, slight positive pressure may lead to moisture problems in the envelope. However, in humid southern climates where the dominant moisture flow is inward, positive pressure would be an advantage.
- The use of the large furnace fan to induce a relatively small amount of ventilation air results in relatively high yearly fan energy costs (\$45) compared to the costs incurred when a small exhaust fan is used (\$26).^(a)
- If ventilating only when heating or cooling, the system may underventilate in the spring and fall and overventilate in the winter. Target ventilation levels may be difficult to maintain unless the fan runs continuously or is on a prescribed schedule.
- Circulation of unconditioned air with the furnace fan may lead to occupant discomfort (feel drafty).
- Furnace fan noise levels may be higher than the noise produced by a small dedicated exhaust fan.

Comment

This system appears to be very low-cost because it is essentially a duct from outside the envelope to the return side of the furnace fan. However, to meet the ventilation requirement in all seasons, a parallel control device must be used. Also, the requirement to avoid positive pressure may mandate the use of a mechanical exhaust fan (not passive vent) that would be controlled to run when the furnace fan is ventilating. These additions contribute substantially to the initial cost. The long-term operating cost is approximately 50% higher than a dedicated exhaust fan.

Cost Estimate

Jin Stewart at Evcon Industries^(b) (represents Coleman) in Vancouver, Washington, was contacted concerning the Coleman POS system. This system is as described above except for the vented windows and timer controls. He estimated the cost of the POS system at \$32. In its normal configuration (10 ft of 5-in. insulated duct), the POS system typically delivers 30 to 40 cfm. This is lower than what is needed to meet the HUD requirement (continuous 45.5 cfm for a 1300-ft² house) when thermostat-driven operation is taken into consideration. Increasing the duct diameter to 7 or 8 in. would provide roughly the flow needed. The cost of the POS system (or equivalent) would not change significantly if a larger duct were substituted.

^(a)The derivation of these dollar amounts is explained on p. 11.

^(b) Mention of specific manufacturers and equipment does not imply endorsement of their product, but rather is done to document the cost estimates as realistic values.

POS system parts include

- fresh air supply duct
- hardware for mounting to furnace
- roof cap and grille
- automatic gravity backdraft damper (when not ventilating).

POS system	\$32
Timer control	\$20
Vented windows (4)	\$100
Labor (1.5 h)	\$60 ^(a)
-----	-----
Total	\$212

System 2: House Exhaust Fan (Single Duct)

This approach uses a dedicated exhaust fan to remove air from the house. Fresh air is recovered from passive vents built into the windows or installed in the envelope. The fan is mounted remotely (most likely on the roof) and ducted to the bathroom. Remote mounting minimizes noise levels. The duct is insulated to minimize condensation of exhausted moisture. Exhaust fan operation can be either scheduled from a timer control or turned on from a wall switch. Spot exhaust fans are still used in any additional bathrooms and the kitchen (see Figure 2).

Advantages

- simple to control
- low operating costs
- quiet operation when remotely mounted
- no positive pressure effects (disadvantage in humid southern climates).

Disadvantages

- Negative pressure must be below furnace vent backdrafting point.
- Continuous fan noise in bathroom may motivate the occupant to disconnect the unit or not use it frequently enough.
- Slight negative pressure may be a drawback for manufacturers supplying to both northern and southern climates.
- Does not filter dust from fresh air.

^(a)The 1.5-h estimate is based on a discussion with Mike Snorsky at Kanalfakt, as well as on the estimate for labor (2 to 3 h) in the *Builder Connection - DETAILS 6* guide for site-built houses (Puget Power 1991). It was assumed that installation in a manufactured house would entail about one-half the labor time for installation in a site-built house.

Figure 2. House Exhaust Fan

Comment

The house exhaust fan system is possibly the best overall approach. With slight negative pressure, use of this system would avoid moisture problems in northern climates. It is simple to control, making it easier for the manufacturers to guarantee that the ventilation requirements are being satisfied.

Cost Estimate

Several local manufacturers' representatives were called for pricing information on house ventilation fans. Among them were NuTone, Kanalflokt, and Broan, all of Seattle, Washington; and FANTECH, Inc., of Burton, Washington. It was assumed that an average manufactured home size is 1300 ft², which would require an average background ventilation rate of 50 cfm to meet the criterion level of 0.035 cfm/ft².

It was assumed that most occupants would desire to run their fans less than 12 h/day. The less than 50% duty cycle implies at least a 100-cfm capacity to meet the requirement. This implication is compatible with the available market of fans because the lowest flow capacity for high-quality (centrifugal or squirrel-cage) fans with remote mounting is approximately 100 cfm at 0.25 in. of static pressure. One example is the QT130 offered by NuTone. Mr. Paul Stanton quoted a price of \$104. This unit is rated at 1.0 sones and 103 cfm at 0.25 in. of static pressure. (Note that the cost quotation for this [and the other four systems] reflects basically a retail, low-volume price from the representative; most representatives were reluctant to quote a high-volume price that would be charged to a manufacturer.)

System parts include

- high-quality (long-life/ squirrel-cage) exhaust with automatic backdraft damper
- roof cap and grille
- interior wall cap
- timer controls
- standard wall switch control
- vented windows.

Because this system replaces the normal bath fan, only new costs are considered:

Exhaust fan	\$90 to \$150
{less spot fan}	-\$40
Timer control ^(a)	\$20
Vented windows (4)	\$100
Labor (1.5 h)	\$60
<hr/>	
Total	\$230 to \$280

System 3: House Exhaust Fan (Multiple Ducts)

This system is similar to the single-duct approach except that the exhaust fan is multiplexed (ducted) to additional rooms such as a second bathroom and the kitchen. The multiplexing replaces all spot fans. This option was not considered further because the HUD standard requires individual spot fans in bathrooms and the kitchen. However, this approach would satisfy the intent of the standard so long as independent controls were available in each room ducted to the central fan. The cost of this approach is comparable to that of the single-duct approach because spot fans are being replaced.

System 4: Distributed Balanced System

The distributed balanced system would have a whole-house exhaust and also a whole-house supply fan, each with multiple duct runs to several rooms (bathrooms, kitchen, and laundry room). This system offers the highest potential for operating under balanced pressure. However, this approach was not considered further because of high initial costs.

System 5: Local Balanced System

The approach using the local balanced system involves the installation of several local ventilation units, each with small supply and exhaust fans (no heat recovery). This system was not considered further because of its high initial cost and potential maintenance problems.

^(a) The estimated costs for the timer control and vented windows are based on costs reported in the *Builder Connection - DETAILS 6* (Puget Power 1991) and in Baylon, Lubliner, and Davis (1990).

System Operating Costs

Annual operating costs were estimated for the two prototype ventilation systems that meet HUD requirements for ventilation as described in the new ventilation standard for manufactured homes. Operating costs for a ventilation system are defined as the sum of the heating and cooling costs associated with satisfying the net increase in thermal load, plus the cost of powering the ventilation fan.

Technical Approach

We first estimated the energy consumed in increasing the total effective air change rate by 0.1 air change per hour (ACH) above the assumed natural infiltration rate of 0.25 ACH. Using the assumptions described by TenWolde and Burch (1993), a 0.1-ACH increase requires a rated continuous-ventilation capacity of 0.035 cfm/ft² of total floor area. The effective increase in air change rate (0.1 ACH) is used to estimate the thermal impact; the 0.035 cfm/ft² continuous capacity requirement is used to estimate the fan energy consumption. A typical floor area of 1300 ft² was assumed for a manufactured home.

The energy consumption associated with a ventilation system is composed of three components:

- sensible load
- latent load
- drive power.

Sensible load is the energy required to heat (or cool) the 0.1 ACH of fresh air to the indoor control temperature.

$$\dot{Q}_s = \dot{V} \rho c_p \Delta T \quad (1)$$

where $\dot{Q}_s$ = sensible load, Btu/hr
 $\dot{V}$ = ventilation rate, ft³/hr
 ρ = air density, 0.0745 lb/ft³
 c_p = specific heat of air, 0.240 Btu/lbm°F
 ΔT = $T_{\text{room}} - T_{\text{outside}}$, °F.

Latent load is the energy required to condense and cool the net flux of water vapor in the ventilation air from room temperature to the temperature of the condensate leaving the air conditioner cooling coils. The humidity ratio-hour approach for estimating latent loads, shown in Equation (2) (and analogous to the degree-day approach for estimating sensible loads), should be considered a rough approximation because inside humidity conditions are not controlled. The results that follow indicate latent ventilation loads higher than the sensible ventilation loads. This runs slightly against intuition as one generally thinks in terms of the total latent load on the air conditioner as being substantially less than the total sensible load. The relatively high latent to sensible ratio in the ventilation load is as expected when one considers that ventilation is only a small part of the total sensible load from the whole structure and yet one of the largest sources of latent load.

$$\dot{Q}_l = \dot{V} \rho \Delta h \Delta W \quad (2)$$

where $\dot{Q}_l$ = latent load, Btu/hr
 $\dot{V}$ = ventilation rate, ft³/hr
 ρ = air density, 0.0745 lb/ft³
 $\Delta h = h_{g78} - h_{f50}$, 1077 Btu/lb water
 $\Delta W = W_{room} - W_{outside}$, lb_w/lb_{da}.

Drive power is the electricity consumed in driving the fan motor. In addition to consuming electricity, fans can affect the sensible component by dumping fan heat and fluid kinetic energy into the envelope. In System 2 (house exhaust fan), the fan heat and fluid kinetic energy is exhausted from the envelope and does not add or subtract from the sensible load component. For System 1 (fresh air duct to furnace), the usual operation mode is to ventilate when the furnace or air conditioner is running. For this reason, System 1 has no direct drive power costs (except for some small increase in power use because the fresh return air will be pulled through a run of duct and pushed out through the envelope as compared to the normal recirculation and unducted return). However, when System 1 ventilates at times outside of heating and cooling demand, nearly all of the furnace fan energy will be consumed in recirculating air. Consequently, as this fan energy dissipates within the envelope, an internal gain affecting the sensible component will result. To simplify the calculations for System 1, it was assumed that 1) the time the unit ventilates outside of calls from the thermostat is small (12.5% of the year) and 2) any *net* effects on the yearly load from ventilating in this mode will be small because the internal gain will decrease the thermal load in the winter and increase it during the summer. Any net effects will slightly reduce System 1 operating costs in cold climates and increase them in warm climates.

Assumptions

The calculations performed to estimate the operating costs were based on the following assumptions:

- U.S. average electricity cost = \$0.083/kWh (*Energy Design Update* 1993)
- Air-conditioner coefficient of performance (COP) = 2.93
(seasonal energy efficiency ratio [SEER] = 10)
- U.S. average natural gas cost = \$5.95/MBtu (*Energy Design Update* 1993)
- Furnace efficiency = 75%
- System 1 furnace fan consumes 500 W of electricity and runs as a pure ventilator (no heating or cooling) for 25% of the year at a 50% duty cycle.
- System 2 fan, rated for 100 cfm, consumes 70 W of power and runs all year at a 50% duty cycle.

- Winter indoor temperature = 68°F. Summer indoor conditions are 78°F, 50% relative humidity (0.010 humidity ratio).
- Total floor area = 1,300 ft²; volume = 10,400 ft³.

Results

Table 1 shows weather characteristics, annual thermal load, and annual costs for fuel consumed in satisfying the load. The results for 20 cities are sorted in order of decreasing operating costs. The annual cost for electricity to drive the fan is \$45.40 for System 1, the prototype with the fresh air duct to the furnace return. For System 2—the house exhaust fan prototype—the annual operating cost is \$25.50. The total for each system includes the fuel costs for satisfying the annual heating and cooling loads as well as the cost of fan motor electricity associated with ventilation.

Table 1. Energy Required to Provide 0.10 ACH Above Natural Ventilation Level of 0.25 ACH

City	HDD ^(a)	CDD ^(b)	HRH ^(c)	HL ^(d)	SCL ^(e)	LCL ^(f)	HL\$ ^(g)	CL\$ ^(h)	S1\$ ⁽ⁱ⁾	S2\$ ^(j)
Bismarck, ND	10,179	150	1	4.54	0.07	0.08	36.0	1.3	83	63
Minneapolis, MN	9,271	200	2	4.14	0.09	0.17	32.8	2.1	80	60
Cheyenne, WY	8,300	79	0	3.70	0.04	0.00	29.4	0.3	75	55
Chicago, IL	7,121	192	1	3.18	0.09	0.08	25.2	1.4	72	52
Cleveland, OH	7,154	117	1	3.19	0.05	0.08	25.3	1.1	72	52
Indianapolis, IN	6,808	162	2	3.04	0.07	0.17	24.1	2.0	72	52
Boise, ID	7,046	258	0	3.14	0.12	0.00	24.9	1.0	71	51
Boston, MA	6,738	108	1	3.01	0.05	0.08	23.9	1.1	70	50
Seattle, WA	6,533	21	0	2.92	0.01	0.00	23.1	0.1	69	49
Washington, DC	5,096	238	3	2.27	0.11	0.25	18.0	3.0	66	46
Albuquerque, NM	5,342	358	0	2.38	0.16	0.00	18.9	1.3	66	46
Raleigh, NC	4,512	217	3	2.01	0.10	0.25	16.0	2.9	64	44
Phoenix, AZ	2,400	1,579	4	1.07	0.70	0.33	8.5	8.6	63	43
Houston, TX	2,217	550	10	0.99	0.25	0.83	7.8	9.0	62	42
San Francisco, CA	4,721	17	0	2.11	0.01	0.00	16.7	0.1	62	42
Lake Charles, LA	2,167	521	10	0.97	0.23	0.83	7.7	8.9	62	42
Charleston, SC	2,958	346	6	1.32	0.15	0.50	10.5	5.5	61	41
Miami, FL	429	642	15	0.19	0.29	1.25	1.5	12.8	60	40
Los Angeles, CA	2,829	33	0	1.26	0.01	0.00	10.0	0.1	56	36
Las Vegas, NV	2,167	521	0	0.97	0.23	0.00	7.7	1.9	55	35

^(a)HDD = heating degree-days (Olsen et al. 1984a,b,c,d)

^(b)CDD = cooling degree-days (Olsen et al. 1984a,b,c,d)

^(c)HRH = humidity ratio hours (Olsen et al. 1984a,b,c,d)

^(d)HL = sensible heating load (MBtu/yr)

^(e)SCL = sensible cooling load (MBtu/yr)

^(f)LCL = latent cooling load (MBtu/yr)

^(g)HL\$ = cost to satisfy heating load (\$)

^(h)CL\$ = cost to satisfy cooling load (\$)

⁽ⁱ⁾S1\$ = System 1 operating costs (\$)

^(j)S2\$ = System 2 operating costs (\$).

Conclusions

Of the five prototype ventilation systems evaluated, two were considered most likely to be used by manufacturers in their efforts to meet the new requirements of the HUD standard. System 1 (fresh air duct to furnace) and System 2 (house exhaust fan) were chosen because they offer a low-cost, long-life, easy-to-operate approach.

The estimated material and labor costs for System 1 was \$212 per home. For System 2, the estimated cost ranged from \$230 to \$280 per home. The two most expensive elements in System 1 were the vented windows and installation labor. The exhaust fan and the vented windows proved to be the most costly items for System 2.

Annual operating costs for the two ventilation systems were estimated for 20 U.S. cities. The operating costs for System 1 ranged from \$55/year in Las Vegas, Nevada, to \$83/year in Bismarck, North Dakota. The operating costs for System 2 ranged from a low of \$35/year in Las Vegas to \$63/year in Bismarck. On average, for the 20 cities considered, fan motor electricity accounted for more than half of the annual operating cost. Cities dominated by heating loads had the highest operating costs.

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Appendix

HUD Ventilation Standard for Manufactured Homes: Change Document

Appendix

HUD Ventilation Standard for Manufactured Homes: Change Document

3280.103 Interior Light and Ventilation

General

Latest Revision:

- A total ventilation rate of 0.35 air changes per hour (ACH) (includes both induced ventilation and natural infiltration). Use any of the five following ventilation systems that will have a minimum continuous capacity of 0.035 cfm/ft² of floor area.
 - i Fresh air inlet on the return side of the existing air handler. Sufficient exhaust vents to avoid positive pressure.
 - ii Mechanical ventilation system.
 - iii A passive ventilation system other than windows.
 - iv Combined passive and mechanical
 - v Operable windows except in Region 4 and Alaska.
- Systems shall be balanced or have adequate air inlets or exhausts. Adverse negative or positive pressure is to be avoided.
- No exhausting into the attic; only directly outside.
- Intermittent exhaust vents shall have an automatic or manual damper.
- Capacity of mechanical ventilation systems shall be certified.
- At least half of the glazed area required by 3280.103(a) shall be openable.

Proposal in *Federal Register*:

- Sufficient ventilation to provide a total, including natural infiltration, of 0.35 ACH.
- Independent mechanical system that would intake 75 cfm of air.
- Exhaust system capable of exhausting 50 cfm of air per minute. Exhaust system can be either mechanical or passive. It must be in addition to the flows required for the bathroom and kitchen.

- Any mechanical systems must have manual controls (additional automatic controls are optional).
- Each habitable room must have ventilation area (windows) equal to 5% of floor space.
- All rooms (other than bathrooms, toilet compartments, and kitchens) that do not have exterior walls must have 10 cfm of mechanical ventilation. The mechanical ventilation system may be integral with the whole-house supply/exhaust system.
- Open windows cannot be used to meet these new flow requirements except in the toilet compartment.

Current:

- Each habitable room must have ventilation area equal to 4% of floor space.
- Windows need not be openable if there is a mechanical system producing an air change every 30 minutes in the rooms with not less than one-fifth of the air being taken from the outside of the envelope.

Kitchen

Latest Revision: no change.

Proposal in *Federal Register*:

- Exhaust rates of 100 cfm. Fan must be within 10 ft of the range or cooktop.

Current:

- If no outside walls in kitchen, one air change every 30 minutes from mechanical system.

Bathroom and Toilet Compartment

Latest Revision:

- 50 cfm of mechanical ventilation. A separate toilet compartment may alternately use window ventilation (minimum of 1.5 ft²) in place of the mechanical ventilation except in Region 4 and Alaska.

Proposal in *Federal Register*:

- 50 cfm of mechanical ventilation. A separate toilet compartment may use window ventilation in place of the mechanical.

Current:

- Must have windows (minimum of 1.5 ft²) or 1 air change every 12 minutes from mechanical system that exhausts directly to the outside.

3280.710 (g) Ventilation (under: Venting, ventilation and combustion air) improvement options to improve indoor air quality

Latest Revision: no changes.

Proposal in *Federal Register*: no changes.

Current:

- In addition to the minimum ventilation requirement in Section 3280.103, the manufacturer shall make available one of the following options to improve indoor air quality:
 - passive
 - mechanical
 - combination
 - fresh air intake to the HVAC system (at least 25 cfm).

3280.504 Condensation Control and Installation of Vapor Retarders

Vapor Barriers

Latest Revision:

- Regions 3 and 4 and Alaska: wall, roofs (ceilings), and floors use a vapor barrier (1 perm) on the living side of the construction.
- Regions 1 and 2: do either A or B for all sides of the envelope:
 - A. Keep permeance on the living space side of the insulation greater than 5 perms.
 - B. Use a 1-perm barrier exterior to the insulation.

Proposal in *Federal Register*:

- In Southern climate (zone II), ceiling vapor barriers may be omitted when the attic vent area is 1/150 of the roof cavity floor area.

Current:

- 1-perm barriers for all ceilings (living space side)

- 1-perm barriers for all walls (living space side) along with a high permeance covering or sheathing on the outside.

Attic Ventilation

Latest Revision:

- Attic ventilation (of any type, mechanical or passive) is optional.

Proposal in *Federal Register*:

- Ventilation area equal to 1/300 of the floor space or a mechanical system capable of 10 ACH or 0.7 cfm/ft² of attic floor area (at 0.03 in. static pressure).
- Single-section homes with metal roofs are excused from the above requirement, provided that
 - Ceiling vapor retarder and air leakage paths to the attic are sealed.
 - Conditioned space mechanical ventilation system is as described above.
 - The system is switch-controlled and has an automatic humidity sensor control.
- 50% to 60% of ventilation area to be provided by ventilators in the upper portion of the space, with the rest coming from eave, soffit, or low gable vents. The vent locations are chosen to provide cross-ventilation of the entire attic.
- A minimum of 1-in. gap between the top of the insulation and the roof sheathing or roof covering.
- Free ventilation areas should be designed to prevent the entrance of bugs, rain, and snow.

Current: Nothing.

3280.505 Air Infiltration

General

Latest Revision:

- Continuous air barrier on walls through either use of a continuous sheet (the poly moisture barrier would satisfy this requirement) or special efforts at sealing cracks and joints and general air leakage paths such as with electrical outlets.
- Ceiling: all leakage paths are to be sealed. No venting into the attic.
- General joints and penetrations: all to be sealed, weatherstripped, or gasketed.
- Ventilated walls are not permitted.

- Vapor-retarder groundcover over the bare earth of a crawlspace.

Proposal in *Federal Register*: No change.

Current:

- Limit air infiltration without sacrificing air quality. Limit the impact of penetrations (through good design) and joints (through good design or sealing after construction).

Envelope Penetrations

Latest Revision: See above under **Air Infiltration**.

Proposal in *Federal Register*: No change.

Current:

- Plumbing and electrical penetrations, windows, and door frames are considered. Outlet boxes are considered exempt.

Joints Between Major Elements

Latest Revision: See above under **Air Infiltration**.

Proposal in *Federal Register*: No change.

Current:

- Caulking and sealing of joints.

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