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Field Measurements of Heating System Efficiency in Nine Electrically-Heated Manufactured Homes

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Ecotope, Inc.
Seattle, WA

Revised Report Prepared for the Bonneville Power Administration
under Contract #DE-AM79-91BP13330
Task Order #61668

Disclaimer

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(revision of 'Field Measurements of Heating Efficiency of Electric Forced-Air Furnaces in Six Manufactured Homes', report #DOE/BP-2458, published October 1994)

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Executive Summary

This report presents the results of field measurements of heating efficiency performed on nine manufactured homes sited in the Pacific Northwest. The testing procedure collects real-time data on heating system energy use and heating zone temperatures, allowing direct calculation of heating system efficiency.

The test homes are factory-built units constructed to thermal specifications approximately 35% more energy-efficient than those recently revised (October 1994) specifications mandated by the United States Department of Housing and Urban Development. The homes are heated with conventional electric furnaces and forced-air distribution systems.

The test protocol combines real-time measurements of furnace energy usage with energy usage during periods when zonal heaters heat the home to the same internal temperature. By alternating between furnace and zonal heaters on 2 hour cycles, a "short-term co-heat test" is performed. Additional measurements, including blower door and duct tightness tests, are conducted to measure and characterize the home's tightness and duct leakage so that co-heat test results might be linked to other measures of building performance.

Manufactured housing represents an affordable housing type for a large percentage of the population in the United States, and is the dominant housing type in many rural areas. Nationally, manufactured homes account for about 15% of the existing housing stock, and this proportion is expected to rise. In the Pacific Northwest, this percentage is about 25% of the single-family housing stock. At current production levels, manufactured homes represent about 30% of new houses being built in this region. Furthermore, more than 90% of manufactured homes sited in the Pacific Northwest are heated with central forced-air electric furnaces. Discovering the impacts of forced-air heating systems on the overall building energy load of manufactured homes is therefore an important undertaking.

Thermal standards which purport to improve the efficiency of these homes might easily be undermined by duct air leakage and poor heating system efficiency. No comparable work has been done to measure the heating system efficiency of manufactured homes built to Model Conservation Standards (MCS), although some detailed tests have been performed on manufactured homes heated with natural gas in an attempt to validate earlier HUD thermal standards.

The heating system efficiency is measured in a number of ways. As defined in Chapter 29 of the 1992 ASHRAE *HVAC Systems and Equipment Handbook*, the **heat delivery efficiency** is the amount of total useful heat delivered through the supply registers while the furnace fan is on, divided by the power input to the furnace (including the air handler fan power). The **system efficiency** is the amount of total useful heat delivered to the conditioned space during the entire period of furnace cycling, divided by the power input to the furnace (including fan power). System efficiency includes heat which is recovered from buffer spaces and home structural members during furnace off-cycle periods. Both of these efficiency measures are reported here.

The steady-state efficiency is also measured and compared to modeled steady-state efficiency results.

The average measured system efficiency for these homes is 83%, meaning that a home with ducted heat delivery uses 1.20 times as much input energy to maintain a given thermostat setpoint as a home heated to the same setpoint with zonal electric heaters (assuming the zonal heaters are 100% efficient).

Manufactured home duct systems differ from those in most site-built homes because they lack a ducted return system and because the ducts run in a thermal zone which is not as isolated from the home's interior as a crawlspace or attic. The measurement protocol used in this study does not disaggregate conductive and convective energy losses, nor does it suggest design changes which could improve system efficiency. A mathematical model is employed which does offer some insight into the relative size of loss factors.

Given the atypical energy specifications used in the production of these homes, the results should not be generalized to older manufactured homes or even new manufactured homes built in other regions.

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1. INTRODUCTION

During the past fifteen years, organized energy conservation efforts have focused primarily on building shell and air sealing measures, including improved insulation levels and better windows. Researchers and sponsors of conservation programs have only very recently shifted their attention to heating systems and their effect on overall energy usage in the home. There have been only a handful of reports published on the effects of distribution systems on home energy usage, and these reports have largely ignored manufactured homes.

Various researchers (for example, Olson et al. 1993, Cummings et al. 1990, Modera 1989, Parker 1989, Robison and Lambert 1988) have reported decreases of between 10% and 30% in the heating system efficiency in groups of site-built homes due to conduction losses, direct duct air leakage, and associated effects. These researchers used a variety of measurement and estimating techniques on relatively small groups of homes, so results should be viewed cautiously; however, there is now general recognition that forced-air distribution systems can add significantly to annual heating costs.

It is the purpose of this report to examine the distribution systems in new manufactured homes built to the Bonneville Power Administration's (BPA) Manufactured Home Acquisition Program (MAP) construction standards. Through a contractual agreement between BPA, participating utilities and home manufacturers, all electrically heated manufactured homes in the Pacific Northwest built from 1992 until 1995 were constructed to these Model Conservation Standards. The specifications included alterations in floor construction and insulation strategies, and duct sealing requirements which were intended to improve heating distribution efficiency.

The primary findings on heating system efficiency are discussed in the following sections of this report. For clarity, the findings are presented in a series of tables with accompanying explanatory text. These tables present site characteristics, air and duct leakage results, average temperatures in and outside of the home during tests, heating efficiencies, heating system pressures and flows, furnace cycling and power loss summaries, and the effect of efficiency losses on annual heating load and cost.

The testing was conducted on nine homes built to MAP specifications in 1993 and 1994. With the limitations of such a small sample, this report does not determine which type of floor construction and insulation technique will consistently result in the least amount of energy waste. These conclusions would require an expanded protocol and paired and sequential testing of different floor and duct system designs. Observations and mathematical modeling are offered, however, which suggest the relative magnitude of efficiency loss factors in these homes.

2. FIELD PROTOCOL AND TESTING METHODOLOGY

This section details several different tests performed to evaluate the heating system efficiency and related parameters. These include a thorough house audit, an alternating co-heat test, a furnace steady-state temperature test, tracer gas decay tests, a two-point blower door test, duct tester measurements, and other miscellaneous measurements. The protocol takes roughly 45 technician-hours over a two day period to complete. The first day is spent mostly on setting up the alternating co-heat test (placing thermocouple wires and power meters, etc.). This test runs overnight under automated control. The next half-day is spent completing testing and removing the test equipment.

The data collected using this protocol is carefully reviewed and analyzed to estimate heating system efficiency. The analysis includes a modeled heat delivery efficiency based on audit measurements.

2.1 House Audit

We performed an audit of the house and heating system prior to running the tests. The audit includes drawing a floor plan of the house, complete with elevations, from which we calculate the house volume. The heating system is described in detail, including furnace and air handler fan specifications, duct length and make-up air system. Trunk ducts are checked with a mirror for crushing. Crossover ducts are also inspected. Any peculiarities or flaws in the furnace or duct system are noted.

2.2 Co-heat Test

This test is the method by which we obtain the overall heating ***system efficiency***. This is defined as the ratio of the power required to heat the home with electric resistance baseboard heating to the furnace power used during normal cycling to provide the same average room temperatures. For the purposes of this test, portable space heaters are placed in every room with a supply register, and the house is alternately heated with these heaters and with the furnace during two hour intervals. Individual rooms are maintained at the same average temperature (with very limited fluctuation) during space heater operation as during furnace cycling, thus allowing us to separate the furnace and space heater energy demands. This methodology accounts for all heat delivered to the living space, including heat recovered from the ducts and crawl space, as well as uneven loading of buffer zones.

The space heaters and furnace are controlled with Campbell 21X dataloggers, which are commanded by batch files from a laptop computer. The furnace is controlled to cycle between six and eight times per hour, and the space heaters maintain room temperatures to ± 0.25° C of those produced during the previous furnace period. Furnace and co-heat power consumption are monitored with a pair of clamp-on true power meters attached to the house electrical mains. Temperatures are measured with copper-constantan (Type T) thermocouple wires.

Supply register air temperatures and flows are also measured. The flows are corrected to standard air density and multiplied by the register temperatures to calculate the total useful heat delivered to the conditioned space during furnace cycling periods. The ***cycling heat delivery efficiency*** is then calculated by dividing the heat delivered through the registers by the power consumed by the furnace. This efficiency is calculated for the same furnace cycling period as the system efficiency.

2.3 Tracer Decay Test

The tracer decay tests serve two purposes: they quantify the infiltration rate of the home with the fan running and they provide a measure of the additional infiltration added by the furnace air handler fan. The test is performed by injecting a quantity of sulfur hexafluoride (SF_6) into the home and measuring concentrations with a Brüel & Kjær 1302 gas monitor. The furnace air handler is used to initially mix the house air, and then small portable oscillating fans are used to mix air continually during the decay tests.

During the first part of the test, the air handler fan is on. When the concentration drops by at least 10%, the air handler is turned off and the second decay begins, continuing until an additional 10% of the SF_6 leaves the building. The time constants of the two decays are the air handler fan induced air-change rate, and the natural infiltration rate of the house plus ducts, respectively. For some houses, a tracer decay test was also performed with exhaust fans operating.

2.4 Blower Door and Duct Blaster™ Measurements

Envelope leakage is measured with a depressurization blower door test at 50 Pa and 25 Pa. Pressures are measured with a two channel digital micromanometer.

A Minneapolis Duct Blaster™ measures duct leakage. Duct leakage tests are done with supply grilles blocked. The duct system is pressurized to 50 Pa and the flow through the Duct Blaster™ fan recorded. The test is repeated with the house pressurized to 50 Pa in order to isolate and quantify duct leakage to outdoors. This test is repeated with the ducts pressurized to 25 Pa.

2.5 Additional Measurements

The pressures across closed bedroom doors are measured with the furnace on. This gives an indication of the magnitude of increased infiltration due to differential pressurization when interior doors are closed. Pressures across the house envelope are also measured with the furnace both on and off to give an idea of the heating system leakage balance. The Duct Blaster™ is attached to the furnace cabinet and used to measure the flow through the air handler fan with the furnace at its normal operating static pressure.

3. SITE CHARACTERISTICS

Basic descriptive data for the test homes are shown in Table 3.1. Five homes are sited in western Washington. Home M01 is sited at an elevation of 1750 ft^3 just east of the Cascade mountain crest. Homes M07, M08, and M09 are sited in Boise, ID.

The study homes are of basic manufactured home design: mostly rectangular structures with moderate-angle gable roofs. The average home size of 1434 ft^2 is close to the regional average size for MAP homes: 1406 ft^2 (Baylon et al. 1995). Double-width homes are by far the most common type of home built to MAP specifications, making up 75% of all MAP homes. The average size of a MAP double-width home (based on manufacturers' records) is 1424 ft^2 (Baylon et al. 1995).

TABLE 3.1: TEST HOME CHARACTERISTICS

Site ID	Location	Width	Floor Area [ft^2]	House Volume [ft^3]	Site Altitude [ft]	Duct Length [ft]	Trunk Duct Material	Furnace Capacity [kW]
M01	Blewett, WA	Double	960	7561	1750	88	Duct board	11.6
M02	Graham, WA	Double	1716	14586	530	135	Metal	15.2
M03	Langley, WA	Triple[1]	2038	18530	50	145	Metal	19.2
M04	Vashon, WA	Double	1709	14813	250	150	Metal	11.2
M05	Snoqualmie, WA	Double	1699	14144	425	142	Metal	15.2
M06	Everett, WA	Double	1739	14900	350	135	Metal	15.2[2]
M07a/b[3]	Boise, ID	Double	1340	11334	2830	115	Metal	21.6
M08	Boise, ID	Single	858	6280	2710	56	Metal	16
M09	Boise, ID	Single	846	6551	2710	52	Metal	16
Average	-	-	**1434**	**12078**	**1289**	**113**	-	-

[1] Has additional section containing family room and master bedroom.

[2] Nominal rating; only two elements (supplying about 8 kW) connected when tests run.

[3] Two tests were run at this site.

The heating plant for these homes consists of a downflow electric furnace installed in a louvered cabinet inside the home. There is no ducted return system, although some furnaces receive ducted outside air through passive or ducted make-up air systems; metal trunk ducts on each side of double width homes are connected by a large-diameter (usually 12") round insulated flex duct called the cross-over duct. In most cases, the trunk ducts, boot risers, and boots are constructed of 18-gauge aluminum, and are fabricated on-site.

A manufactured home's floor system is built up on top of a steel undercarriage which serves also as the home's transportation platform. Fiberglass insulation blankets are draped over the steel undercarriage. Wooden floor joists, usually running perpendicular to the steel I-beams, are eventually placed on top of this blanket. In the most common floor insulation strategy, the belly insulation is slit so that it can be brought into the joist cavities outboard of the main steel I-beams. In a notable exception to this construction strategy, two manufacturers frame the floor joists parallel to the steel understructure and run each trunk duct inside a single joist cavity. This "longitudinal" floor offers some theoretical advantages in terms of sequestering duct losses. Site M04 is the only home in this study with a longitudinal floor. Refer to the appendix for diagrams and more discussion of floor construction details.

In all homes, trunk ducts are wrapped with fiberglass insulation and run above the R-33 belly insulation. Underneath the entire floor structure, there is a continuous nylon barrier called the belly board which protects the insulation and framing members. Several plumbing lines penetrate this barrier; holes in the barrier may or may not be repaired after the home is transported and set up.

The remainder of the house is insulated to standards equivalent to Pacific Northwest site-built codes, namely, R-21 walls with insulated headers and minimized framing lumber, R-38 vaulted ceilings or R-49 attics. The windows used perform on average to a U-value of 0.40 or better, and overall glazing area averages about 12% of the heated floor area.

At Site M07, two tests were run. The first test was run in the "as-found" condition. The second test was run with a piece of sheet metal installed in the trunk duct near the furnace so that a substantial amount of the crossover duct inlet was blocked. The second test condition was created in order to investigate the effect of crossover duct blockage on system efficiency. The impetus for performing this test had to do with Site M05. At this home, the air flow differential between the furnace and non-furnace trunk ducts was dramatic, suggesting a possible crossover blockage. The blockage could not be confirmed, but the idea for the intentional blockage test was born.

4. HOUSE AIR LEAKAGE

This section discusses house and duct air leakage. Table 4.1 presents blower door and tracer gas results. The Minneapolis Blower Door (Model 3) was used for the house tightness test. The house was depressurized to approximately 50 Pa relative to outside, with the ducts open to the house. Air leakage figures are expressed in standard cubic feet per minute (SCFM), which is the flow rate that would be required at a density of 0.075 lb/ft^3 to give the same mass flow rate as was actually measured at the site.

TABLE 4.1: HOUSE AIR LEAKAGE SUMMARY

	Blower Door		Tracer Decay[3] (ACH)				Δ Pressure [Pa]	
Site ID	Q50[1] [CFM]	ACH_{50}[2] [vol/hr]	Fan Off	Fan On	Difference	Exhaust Fans[4]	Envelope[5]	Bedroom Door[6]
M01	504	4.00	0.10	0.22	0.12	0.58	0.8	9.4
M02	1257	5.17	0.16	0.20	0.04	0.40	0.4	2.0
M03	989	3.20	0.18	0.28	0.10		0.4	4.6
M04	840	3.40	0.16	0.16	0.00		0.8	*[7]
M05	766	3.25	0.10	0.13	0.03	0.76	0.5	5.8
M06	1050	4.23	0.07	0.13	0.06		0.3	3.0
M07a/b	1039	5.50	0.10	0.23	0.13		0.9	8.0
M08	533	5.07	0.12	0.64	0.52		3.6	25.6
M09	820	7.50	0.11	0.39	0.28		1.5	16.0
Average	**866**	**4.59**	**0.12**	**0.26**	**0.14**	**0.58**	**1.0**	**9.0**
Stand. Dev.	**247**	**1.39**	**0.04**	**0.16**	**0.16**	**0.18**	**1.0**	**7.5**

1 Total leakage in standard ft^3/min (SCFM) with ducts unsealed, all interior doors open, and house depressurized to 50 Pa. Furnace and exhaust fans off.

2 Same conditions as above but leakage expressed in air changes per hour.

3 Total leakage expressed in air changes per hour based on tracer gas decay test. Tracer tests done with all interior doors open.

4 Both bathroom ventilation fans on. Air handler fan off. Due to time constraints this test was not performed at every site.

5 Change in pressure across house envelope due to air handler fan (fan on - fan off). In all cases the houses are depressurized with respect to outside. (Numbers shown are the absolute value of the ΔP.) Test performed with all interior doors open.

6 Change in pressure across the master bedroom door due to air handler (fan on - fan off). In all cases the bedrooms are pressurized with respect to the main part of the house. Most master bedrooms have over-door fixed-blade louvers. All interior doors open except for the door to the tested room.

7 Interior doors were not yet installed at this site.

4.1 Comparative Leakage Rates

These homes have an average tightness at 50 Pa (ACH_{50}) of 4.59 ACH, which is very tight by any standard applied to site-built or manufactured homes. The MAP air sealing specifications assume a maximum measured tightness of 7.0 ACH_{50}. For purposes of comparison, the ACH_{50} values for other research conducted in the past several years are shown below:

TABLE 4.2: COMPARATIVE WHOLE-HOUSE LEAKAGE RATES

Study	Energy Efficient Homes[1]	Number of Homes	House Type	Year Built	ACH_{50}
Palmiter & Brown (1989)	No	134	Site-built	1980 - 86	9.28
Palmiter et al. (1990)[2]	Yes	49	Site-built	1987 - 88	7.18
Palmiter et al. (1990)[2]	Yes	129	Site-built	1989	5.55
Kennedy et al. (1994)	No	21	Manufactured	1965 - 80	14.3
Palmiter et al. (1992)	No	29	Manufactured	late 1980s	8.75
Palmiter et al. (1992)[2]	Yes	131	Manufactured	1988 - 89	6.10
Baylon et al. (1995)[3]	Yes	157	Manufactured	1992 - 93	5.50

1 Affirmative entry means these homes were built to a more energy efficient specification than the most common practice at the time.

2 Built to BPA energy efficiency standards.

3 Random sample of MAP homes; these homes are built to the same thermal specifications as the homes tested in this study.

The ACH_{50} is a useful comparison measurement; however, a house very rarely experiences a pressure gradient of 50 Pa. Taking measurements at this relatively high reference pressure is done to reduce measurement error. Homes sited at many locations in the Pacific Northwest generally experience pressure differentials of between 0-2 Pa during the heating season. This is the time of the year when infiltration driving forces are at their greatest, since temperature differences are greatest and the furnace fan operates frequently.

4.2 Tracer Gas Test Results

Tracer gas tests were performed to better estimate the effective whole-house ventilation rate under normal winter conditions with the furnace air handler fan on and off. These results are also summarized in Table 4.1. The effective ventilation rate describes the actual rate at which pollutants are removed from the home by introduction of outside air and removal of stale indoor air. The effective ventilation rate is generally less than the time-weighted average ventilation rate, which is the rate commonly used for heat loss calculations. A full discussion of the difference between the two rates can be found in the appendix.

The tracer gas decay test was performed by injecting sulfur hexafluoride (SF_6) into the home's air handler until the indoor concentration (measured at a central sampling point) reached about 5 parts per million (ppm) at a central sampling point. The air handler fan was left on, (mixing fans were placed in all rooms to circulate the air) and the concentration was allowed to drop by approximately 10%. The air handler fan was then turned off. The gas concentration was allowed to drop by an additional 10%. The house air change rate for both cases was determined from the slope of a linear regression of the concentration versus time. An infrared photoacoustic gas analyzer (the Bruel and Kjær Model 1302) was used to measure the SF_6 concentration in real-time. This analyzer is calibrated to work in the range of concentrations commonly encountered (1-5 ppm). Tests were performed during the early morning, with the ΔT between inside and outside usually 20°F or more during the testing period.

Table 4.1 shows the results of the decay tests with the air handler fan on and off. With the fan off, the natural ventilation rate ranged from 0.07 ACH to 0.18 ACH, averaging 0.12 ACH. Under these testing conditions of stack-dominated infiltration/exfiltration (which is the norm in much of the Pacific Northwest, since sustained wind-driven infiltration/exfiltration is generally limited), none of the houses meet the ASHRAE Standard 62 recommended minimum effective ventilation rate of 0.35 air changes per hour (ASHRAE 1989).

The difference between fan-on and fan-off house air change rate represents duct leakage and induced infiltration caused by operation of the air handler fan. Depending on the natural infiltration rate and the amount of duct leakage, the amount of induced infiltration can be considerable (Palmiter and Bond 1991). With the furnace running (air handler fan on), the effective ventilation rate increased to an average of 0.26 ACH. Site M04 showed no change, which reflects the very limited duct leakage (about 20 SCFM at furnace operating conditions) and tight building shell. Even though the building envelope was depressurized by 0.8 Pa, there was very limited incursion of extra outside air when the air handler was running. The average increase in home infiltration/exfiltration rate from the operation of the air handler fan is 0.14 ACH.

The tracer test introduces gas into the interstitial belly area and crawlspace when the air handler is on, and some of this tracer re-enters the house during both decay tests. This means that the air change rates measured with a tracer decay are appropriate for air quality concerns, but they are biased low for purposes of evaluating efficiency impacts of heat loss due to induced infiltration.

The single-section homes tested had significantly greater fan-on infiltration rates than multi-section homes. Envelope depressurization during furnace operation was significantly greater in single wides than in multi-section units. Relatively high duct leakage at Site M08 also is a major contributor to its dramatic increase in infiltration rate when the furnace air handler is on.

We also made tracer measurements with the exhaust fans running in three homes. In each case, both bathroom fans (delivering a nominal exhaust capacity of 50 CFM per fan) were switched on for the duration of the test. The average air change rate with the exhaust fans running was 0.58 ACH. The exhaust fans in these homes were set to run, on average, for a total of 8 hours per day.

The daily effective ventilation rate in these homes is therefore considerably below the minimum ASHRAE performance standard.

For the purpose of estimating the combined effect of unintentional leaks and exhaust fan operation, we can take a weighted average of the tracer decay air change rates for the natural leakage case ("Furnace Fan Off" in Table 4.1) and the three exhaust fan on cases ("Exhaust Fans" in Table 4.1). The tracer decay test is, again, a measure of *effective* ventilation rate and therefore describes how quickly pollutants are removed from the home's interior. Note that there are only three homes where this comparison can be made (sites M01, M02, and M05).

If two 50 CFM fans are timer-controlled to run a total of 8 hours per day, the weighted average effective ventilation rate for the three homes is 0.29 ACH. While much closer to meeting ASHRAE Standard 62, the fan-assisted ventilation rate still fails the standard. Because MAP homes have been found to be relatively tight (5.5 ACH_{50} for a regional sample, according to Baylon et al. 1995), and because occupants often do not use the fans for more than intermittent spot ventilation, many occupants live in indoor environments with insufficient pollutant removal rates. Appendix C summarizes the calculation procedure used to extend ventilation system operations to effective air changes. For purposes of extending the results of fan flow testing, this procedure will generally result in estimates comparable to tracer gas tests.

The make-up air systems for these homes were disabled for all tests because no reliable procedure has been developed to describe their contribution to overall heating system efficiency and house tightness. During normal furnace operation the make-up air duct behaves like an intentional return leak. The impact of various make-up air systems on ventilation in SGC manufactured homes (Palmiter et al. 1992) showed average flow during furnace operation of 33.8 cfm. These measurements were considered somewhat unreliable because the protocol involved a field technician installing a flow grid in the make-up air duct. Usually, the access to the make-up air port in manufactured homes precludes a correct installation of this device, resulting in an underestimation of the flow rate. Further research should be conducted on the magnitude of the make-up air system on the effective ventilation rate of manufactured homes.

Pressure Diagnostics

Table 4.1 also shows the measurement of the pressure change across the envelope of the house when the air handler is turned on. This measurement averaged –1.0 Pa for these homes, indicating that the house was depressurized when the air handler was running versus when the air handler was off. This is not unexpected, since there is no ducted return system running in an exterior or buffer space which could bring in extra air via leaks to pressurize the house. A pressure difference of zero across the envelope with the air handler running would indicate that either there were no return leaks at all or that supply and return leaks were balanced. This relatively small net depressurization in most homes when the air handler is running is not as sizable as the larger values found in site-built homes, where it was not uncommon to find a +/- 2 Pa gradient due to air handler operation (Olson et al. 1993).

An exception to this is the single-section homes. These homes were manufactured in Idaho and were equipped with large furnaces and blowers relative to their floor area. Retail dealers in Idaho do not stock homes equipped with furnaces with less than 15 kW capacity, even though the house will likely have a design heating load of less than 5 kW in all but the harshest climates in the Pacific Northwest region. Dealers (and consumers) have not had enough experience with super-insulated manufactured homes to appreciate the difference in performance of these homes versus the "trailers" of old. The combination of a large air handler fan, limited envelope infiltration, and moderate duct leakage result in relatively high envelope depressurization in the single-section homes: -3.6 Pa at Site M08 and -1.5 Pa at Site M09, or more than twice the remaining homes.

Table 4.1 also shows the average of the absolute values of pressures measured across interior doors when closed one by one. Previous studies of infiltration in homes with central forced-air furnaces (e. g. Palmiter and Bond 1991) show that the closing of even a single bedroom door can more than double the infiltration rate into the rest of the house. This effect is especially pronounced if the master suite (generally containing multiple supply registers) is adjacent to the furnace closet. We ran all of our efficiency tests with all interior doors open, and no adjustment to the system efficiency was made to account for door closures. The pressurization of rooms with doors closed can be offset slightly by the installation of over-door or ceiling pass-through ventilation louvers. The air flow through these louvers has been measured by Ecotope and is usually only 5 to 10 CFM; pressure differences on the order of 3 Pa still exist between louvered master bedroom suites and the rest of the house when the master bedroom door is closed.

5. DUCT LEAKAGE

Table 5.1 summarizes duct leakage measurements taken with the Minneapolis Duct Blaster™. For the total duct leakage test, registers and the furnace cabinet are temporarily sealed and the Duct Blaster™ fan connected to the supply system and adjusted so that the duct system is pressurized to two reference pressures: 25 Pa and 50 Pa. Duct external leakage is done similarly, except the home's interior is pressurized in turn to 25 and 50 Pa with the blower door so that the pressure differential between duct system and home interior is reduced to around zero. At this point, any measured duct leakage is assumed to be to outside the home's interior and is classified as "exterior leakage."

TABLE 5.1: DUCT AIR LEAKAGE SUMMARY

Site ID	Total Leakage at 50 Pa [SCFM]	Exterior Leakage at 50 Pa [SCFM]	Exterior Leakage at 50 Pa as a % of total at 50 Pa	Total Leakage at 25 Pa [SCFM]	Exterior Leakage at 25 Pa [SCFM]	Exterior Leakage at 25 Pa as a % of total at 25 Pa
M01	118	51	43	76	36	47
M02	179	126	70	117	86	74
M03	232	122	53	150	96	64
M04	122	34	28	77	24	31
M05	201	105	52	123	67	54
M06	158	82	52	102	54	53
M07a	308	103	33	203	63	31
M07b	279	103	37	185	72	39
M08	234	88	38	157	52	33
M09	191	74	39	122	44	36
Average[1]	**194**	**87**	**45**	**125**	**58**	**47**
Std. Dev.	**60**	**31**	**13**	**40**	**23**	**15**

[1] Site M07b not included in summary statistics.

5.1 Duct Air Leakage Results

The average exterior duct leakage at 50 Pa is 87 CFM, with the standard deviation of about one-third of the mean. Site M01, with trunk ducts made of fiberglass duct board rather than sheet metal and the smallest double wide floor area (960 ft^2), and Site M04, with a longitudinal floor, had the lowest duct leakage to outside. The single-section homes have the next smallest exterior leakage. Exterior leakage at 25 Pa averages 58 CFM. This is a better estimate of actual exterior leakage, since 25 Pa is close to the average static pressure measured in these homes when the furnace is operating normally.

Table 5.2 summarizes heating system flows and pressures. Supply register flows are measured along with the furnace air handler flow and combined with delivery temperature data so that the rate of energy delivery at the registers can be compared with the rate of energy delivery at the furnace. Register and air handler flows are corrected to standard temperature and altitude conditions and expressed in standard cubic feet per minute (SCFM).

5.2 Duct Operating Characteristics

Duct system static pressure is measured so that duct leakage at normal system operating pressure can be calculated. The duct system static pressure is measured with a static pressure tip (usually a compact Pitot tube) inserted into the supply plenum or supply register close to the furnace. Rather than using the measured exterior duct leakage at 25 Pa or 50 Pa, a reference pressure equal to 80% of the measured system static pressure was used to represent the average leak driving force operating when the air handler is running. This 80% factor is based on our experience with manufactured home supply ducts, which, because of their shorter runs and lack of traditional supply plenum, generally maintain relatively high static pressures when the air handler operates. Duct leakage is calculated using the basic volumetric flow equation, $\mathbf{Q = Cp^n}$, where $\boldsymbol{Q}$ is the flow (leakage), $\boldsymbol{C}$ is found empirically from the two point total duct leakage test (reported in Table 5.1), $\boldsymbol{p}$ is 80% of the measured duct static pressure and $\boldsymbol{n}$ is assumed to be 0.65 (a common assumption for a flow exponent generally associated with the leaks found in residential building materials and ducts).

TABLE 5.2: HEATING SYSTEM FLOWS & PRESSURES

Site ID	Exterior Duct Leakage at 25 Pa [SCFM]	Duct Leakage at 25 Pa per foot of ductwork [SCFM/ft]	Exterior Duct Leakage at 50 Pa [SCFM]	Measured Duct System Static Pressure [Pa]	Reference Duct System Static Pressure[1] [Pa]	Calculated Duct Leakage at Reference Pressure[2] [SCFM]	Sum of Register Flows[3] [SCFM]	Number of Supply Registers	Calculated AH Flow[4] [SCFM]	Supply Leakage Fraction[5] [%]
M01	36	0.41	51	17.0	13.6	23.1	732	7	755.1	3.1
M02	86	0.64	126	21.1	16.9	64.4	715	10	779.4	8.3
M03	96	0.66	122	34.0	27.2	91.8	955	11	1046.8	8.8
M04	24	0.16	34	26.5	21.2	20.5	948	12	968.5	2.1
M05	67	0.47	105	27.0	21.6	60.9	621	10	681.9	8.9
M06	54	0.40	82	25.5	20.4	46.6	600	8	646.6	7.2
M07a	63	0.55	103	13.2	10.6	36.7	917	11	953.7	3.9
M07b	72	0.63	103	16.1	12.9	44.7	888	11	932.7	4.8
M08	52	0.93	88	52.0	41.6	75.2	770	6	845.2	8.9
M09	44	0.85	74	28.2	22.6	42.6	820	7	862.6	4.9
Avg.[6]	**58.0**	**0.56**	**87**	**27.2**	**21.7**	**51.3**	**786**		**838**	**6.2**
S. Dev.	**23.0**	**0.24**	**31**	**11.2**	**9.0**	**23.8**	**134**		**135**	**2.7**

[1] 80% of the measured supply plenum pressure. This is assumed to be the average pressure in the duct system when the air handler fan is on.

[2] Calculated using the reference duct pressure, the flow coefficient calculated from the 25 Pa and 50 Pa total duct leakage tests, and a flow exponent of 0.65.

[3] As measured with either the Lambert FH250 or the Pacific Science Technology Fast-1 Flow Hood.

[4] Sum of register flows plus the calculated duct leakage at the reference pressure.

[5] Calculated duct leakage divided by air handler flow.

[6] Site M07b not included in summary statistics.

Just as blower door results are normalized by house size and expressed in air changes per hour (ACH), duct leakage can be normalized by air handler size and expressed as a percent of the air handler flow. The last column of Table 5.2 contains the supply leakage fraction for these homes. The average supply leakage fraction is just under 9%, with the lowest value calculated for Site M04 (2.1%). The supply leakage fraction is a necessary input for Ecotope's duct efficiency model, discussed in Section 6.4.

Measurement of supply register and furnace air handler flows is subject to uncertainty. Ecotope has spent many hours experimenting with the flow hoods used in this research and has determined the flow measurements are sensitive to register type, the position of the flow hood, and the position of the register relative to the furnace. In single-section units, very high flows in some registers close to the furnace will force the hood outside of its normal calibration range. Measuring the air handler flow by using the Duct Blaster™ as a helper fan can produce unreliable results, as can depending on the furnace temperature rise method. Because we have relatively more confidence in the supply register flow measurements and the exterior duct leakage measurements, we combined them to estimate air handler flow.

This has less impact than may first be assumed since only one measure of heating system efficiency (the cycling heat delivery efficiency) depends on the flow measurements, whereas the overall system efficiency does not. System efficiency is the figure of merit in this study, since it includes the net effect of heat delivered through supply registers and heat recovered from buffer spaces during off-cycles. The next section discusses heating system efficiency results in more detail.

6. HEATING SYSTEM EFFICIENCY

This section discusses the measurement of the real-time heat delivery efficiency of the furnace and duct systems installed in new manufactured homes. Duct heat conduction and air leakage are the main contributors to decreased heating efficiency. Conduction losses are a significant factor in heating system efficiency, since heating ducts are at temperatures considerably higher than their surroundings when the furnace is running. The testing protocol does not measure conduction directly, but temperature sensors are placed in the belly interstitial region (where the ducts are located) and in the crawlspace so that temperatures in these regions can be compared with temperatures in the furnace supply plenum and at the supply registers.

6.1 Testing Conditions

Table 6.1 summarizes thermal testing conditions at the nine homes. Average room temperatures, crawlspace temperature, outside temperature, and inside-outside temperature difference during the short-term coheat test are reported. The data are averaged over at least six hours in all cases. The efficiency tests were conducted during the night in all cases, so solar effects are not included in any of the data.

The thermostat is set to a high enough temperature to ensure a significant temperature difference between the home interior and ambient while efficiency tests are conducted. A temperature difference of between 30° F and 40° F is typical during the heating season in many parts of the Pacific Northwest. The heating system efficiencies measured under this loading condition can thus be considered representative of typical winter conditions. A possible exception is Site M06, where the average temperature gradient was 25.7° F.

TABLE 6.1: TEMPERATURES DURING TESTING (°F)

Site ID	Avg. Room[1]	Avg. Crawl	Avg. Out[2]	Δ Out[3]	Supply Registers[4]	Δ Supply[5]
M01	75.6	48.2	44.1	31.5	104.2	28.7
M02	76.3	46.6	40.7	35.6	114.9	38.7
M03	74.8	47.5	45.6	29.2	102.6	27.7
M04	74.1	44.1	33.4	40.7	102.4	28.3
M05	75.7	48.4	42.4	33.3	115.1	39.3
M06	74.1	*[6]	48.4	25.7	102.2	28.2
M07a	72.1	49.3	28.9	43.2	102.4	30.3
M07b	72.0	49.6	31.9	40.1	102.6	30.6
M08	72.5	*[6]	33.9	38.6	111.8	39.3
M09	73.8	*[6]	39.5	34.3	101.3	27.5
Average	**74.1**	**47.7**	**38.9**	**35.2**	**105.95**	**31.9**

[1] Average of heating zone control temperatures (6-11 control temperature measurement points per home).
[2] Outside thermocouple shielded from night sky.
[3] Average of heating zone control temperatures minus outside temperature.
[4] Flow-weighted average of register temperatures during furnace cycling.
[5] Average flow-weighted supply temperature minus average inside temperature. When multiplied by the sum of supply register flows, this gives the heat delivered to the home through the supply registers.
[6] Crawl temperature not measured at these sites. These sites had no skirting; therefore, crawl temperature can be assumed to be the outside temperature.

6.2 Efficiency Definitions

Three independent standard measures of heating efficiency are of primary interest in this project. The first two of these efficiency measures are defined in Chapter 29 of the 1992 ASHRAE *HVAC Systems and Equipment* volume. The first measure is the cycling heat delivery efficiency, which is defined as the total useful heat delivered to the supply registers while the fan is on, divided by the power input to the furnace (including fan power). The total useful heat delivered is determined by comparing the rate of energy delivery through supply registers (based on temperature and flow measurements) to the power input to the furnace. This measure of efficiency does not take into account any supply leaks back to the home, nor any heat recovered from the ducts when the air handler fan is off. It also does not take into account the conduction and radiation of heat back into the home from the floor's structural members. The usefulness of this efficiency measure is somewhat limited because it relies on supply register flows, which are subject to considerable measurement error.

The second measure is overall system efficiency. System efficiency is defined as the total useful heat delivered to the conditioned space during the entire period of furnace cycling, divided by the power input to the furnace (including fan power). "Total useful heat" here refers to the electricity that non-ducted electric heaters (such as baseboards) would use to maintain the same average

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Another measure of efficiency, the steady-state heat delivery efficiency, is also calculated. The steady-state efficiency is similar to the cycling heat delivery efficiency, but the measurement is restricted to a smaller time period. The thermostat is set to a high temperature and the furnace is run full-on until the register temperatures reach a maximum. At this point, we assume the ducts and surrounding air and structural materials are completely charged with heat and thus the overall efficiency of heat delivery is maximized. The steady-state efficiency is not as useful as the system efficiency in predicting the seasonal energy penalty associated with duct losses. However, steady-state efficiency can be measured in the field without special equipment and is also the only efficiency figure which can be compared directly with Ecotope's mathematical model of heating system performance.

6.4 Efficiency Measurement & Analysis

All measurements of heating system efficiency rely on some comparison of the energy delivered through supply registers (found through temperature and flow measurements) with the energy delivered at the furnace (found through direct measurement of electrical power and air handler flow measurement). The steady-state heat delivery efficiency relies on limited measurements taken after a relatively short time, whereas cycling heat delivery efficiency and system efficiency are based on a co-heat test which lasts overnight.

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The co-heat protocol measures temperatures and energy usage during alternating heating periods. The furnace and portable heaters ("co-heaters") are alternated on two-hour cycles with an automated control system. The thermostat setpoint is determined based on outdoor conditions; generally, a temperature difference of at least 30° F is desired so that the furnace responds to normal winter ambient conditions. During the co-heat period, the portable heaters in each heating zone are operated automatically by the control system to maintain the temperature in each zone to within ±0.25° C of the average temperature measured during the furnace heating period. In this report, a "heating zone" is usually any room with a heating register. Larger rooms sometimes have two registers, and so co-heaters are sometimes ganged together for simultaneous operation. Power is measured directly during these alternating periods with true power meters clamped on the electrical mains. Room and supply register temperatures are measured with Type T (copper-constantan) thermocouple wires. Readings are taken every second and averaged every 10 seconds.

For purposes of calculating system efficiency, we compared the second hour of the furnace and co-heat periods. Power measurements cannot always be 100% accurate during the transition periods from co-heat to furnace heating modes, due to short-term thermal mass effects. The furnace will stay on longer to heat the duct and underfloor members which cool during the co-heat period. Conversely, the first part of the co-heating energy cycle requires less heating input energy than later parts in the cycle. This is because the furnace has been cycling and has heated up the floor thermal mass, reducing the overall heating load (a combination of the thermal mass load and the load due to the temperature difference between the thermostat setpoint and the outside temperature).

6.5 Efficiency Results

System efficiency for these homes averages 83%. The average contains results from two single-section homes (Site M08 and Site M09), a double-section home with an additional "pod" unit (Site M03) and six double section homes. Site M07 was tested twice in order to assess the effect of artificially diverting furnace flow away from the crossover duct entry point. As the table shows, the diversion of flow did not produce much difference in the system efficiency in Site M07b. This result was surprising; however, mathematical modeling of the duct system after the efficiency data were analyzed actually predicted this outcome. It turned out the flow diversion was not large enough to produce a significant change in system efficiency.

TABLE 6.2: MEASURED HEATING EFFICIENCIES

Site ID	Width Class	Floor Type	System Efficiency[1] [%]	Cycling Heat Delivery Efficiency[2] [%]	Heat Recovery Fraction[3]
M01	Double	Transverse	83	64	0.53
M02	Double	Transverse	89	74	0.58
M03	Triple	Transverse	81	67	0.42
M04	Double	Longitudinal	87	85	0.14
M05	Double	Transverse	74	61	0.34
M06	Double	Transverse	79	71	0.28
M07a	Double	Transverse	78	67	0.34
M07b	Double	Transverse	76	65	0.32
M08	Single	Transverse	86[4]	77	0.39
M09	Single	Transverse	89	76	0.54
Average[5]			**83**	**71**	**0.39**
Median			**83**	**71**	**0.38**
Stan. Dev.			**5.3**	**7.5**	**0.15**

1 System efficiency is the total heat delivered to the conditioned space divided by the energy output of the heating system, as measured by the co-heat method.

2 This efficiency is the heat delivered to the home though supply registers during the time the air handler fan is running divided by the energy output of the heating system. It does not account for factors such as supply leaks to the conditioned space, heat recovered from ducts during the off-cycle, or heat recovered from buffer zones.

3 (System efficiency - cycling heat delivery efficiency) / (1 - cycling heat delivery efficiency).

4 System efficiency for Site M08 is determined based on the home's measured cycling efficiency and the average heat recovery fraction for all homes but Site M04 (longitudinal floor). The heat recovery fraction reported for Site M08 is the average for all sites but M04.

5 Summary statistics do not include Site M07b. Summary statistics for heat recovery fraction also do not include Site M08.

The last column in Table 6.2, the heat recovery fraction, shows the relationship between the cycling heat delivery efficiency and system efficiency. The ratio is not indexed to a common point, so a home with very similar cycling heat delivery efficiency and system efficiency (e. g. Site M04) may have a small heat recovery fraction even though its system efficiency is relatively high. The heat recovery fraction says that, on average, about 40% of the heat "lost" during furnace operation (to the belly region, to floor structural members, and to other thermal bypasses) is recovered as useful heat at some point during the furnace off-cycle.

The homes with lower system efficiency warrant some mention, since Ecotope was able to identify some possible reasons for their relatively poor performance. Site M05 appeared to have some sort of blockage in the crossover duct that we could not positively identify, even on a return visit. Register flows on the side of the home containing the furnace (the "A" side) were markedly larger than on the other side of the home (the "B" side), which can decrease heat delivery efficiency since the rate of energy delivery through registers decreases as supply flow drops. Ecotope attempted to replicate this condition at Site M07 but was unable to restrict the flow as dramatically as was measured at Site M05.

The single-wide homes, Sites M08 and M09, performed relatively well. These units have lower-than-average duct leakage, fewer riser takeoff joints than the multi-section homes, and no crossover ducts. There were some problems with the system efficiency data for Site M08, so its system efficiency was derived from the measured cycling efficiency and the average heat recovery fraction for the other homes (excluding Site M04, which has a longitudinal floor). This procedure probably underestimates Site M08's system efficiency somewhat, since the calculation is based mostly on double-section homes with crossover ducts. However, the calculated system efficiency of 86% is reasonable, given the similarity in floor design, cycling heat delivery efficiency, and testing conditions between Site M08 and M09. (These units were set up side-side on a dealer lot and were tested on successive nights with very similar ambient temperature profiles.)

Site M04 has a longitudinal floor and therefore less separation of ducts from the home's interior and no supply register risers (which reduces duct surface area and therefore reduces conductive losses). Almost all of the energy delivered to the air stream at the furnace finds its way into the conditioned space as useful heat. The manufacturer of this home installs an air flow splitter, so occupants benefit from more even distribution of supply air than in other homes.

6.5 Efficiency Modeling Compared With Measured Results

In order to verify and extend the measured results, Ecotope conducted a series of modeling exercises. The model employs a mathematical procedure under development by Larry Palmiter at Ecotope (Palmiter and Francisco 1996) and first presented (in part) in Palmiter and Bond (1991). The model is designed to predict steady-state heat delivery efficiency given information about duct system surface area, duct insulation, furnace plenum and supply register delivery temperatures at the end of the steady-state test, and air flows into the furnace and out of the supply registers. The model is not designed to predict system efficiency, but it does provide information about the changes in steady-state efficiency which might be expected given different amounts of duct leakage, duct insulation, and furnaces of various capacity. The basic form of the model is discussed in Appendix D.

The steady-state heat delivery efficiency is defined as the ratio of energy delivered through the supply registers to the energy delivered at the furnace after the furnace has been operating for a long enough period to warm the duct material and floor structure. The steady-state heat delivery efficiency thus should be a "best-case" efficiency, since it is measured when the thermal mass of the floor system is charged with heat and conductive losses are minimized. The steady-state heat delivery efficiency is the easiest efficiency to measure in this protocol. However, it is subject to some uncertainty, since it relies on accurate measurements of the supply plenum temperature, supply register air flows, and the air handler flow. All of these measurements are subject to error, as discussed above. Also, Ecotope has observed that some steady-state tests never seem to reach a true "plateau"; that is, even after nearly an hour of warm-up, the supply register air temperatures are still changing slightly and thus the efficiency ratio is not at its maximum. On average, the model results agree quite well with the steady-state heat delivery efficiency measurements.

The table also suggests a (probably) coincidental agreement between average system efficiency and either modeled or measured steady-state efficiency. Limited duct air leakage (averaging less than or equal to 5% of the air handler flow) and elevated levels of duct and belly insulation would suggest this, but at this preliminary stage of the research, it is unwise to draw grandiose conclusions. Agreement is found only between the <u>average</u> values for system efficiency and steady-state efficiency. On an individual basis, it is unwise to predict system efficiency based on steady-state heat delivery efficiency. Errors in prediction using this approach do appear to be unbiased; that is, in some cases, the steady-state heat delivery efficiency is less than the measured system efficiency, whereas system efficiency is less than steady-state heat delivery efficiency in other cases. Much more work needs to be done on separation of efficiency loss factors and on their interaction to allow the modeled result to substitute for a full test of system efficiency.

TABLE 6.3: MEASURED VS. MODELED EFFICIENCY RESULTS

Site ID	Measured System Efficiency [%]	Measured Steady-State Heat Delivery Efficiency [%]	Modeled Steady-State Heat Delivery Efficiency [%]
M01	83	85	88
M02	89	80	84
M03	81	88	80
M04	87	89	91
M05	74	73	74
M06	79	80	80
M07a	78	79	85
M07b	76	80	80
M08	86	85	83
M09	89	85	87
Average	83	83	84
Stan. Dev.	**5.3**	**5.1**	**5.1**

As mentioned earlier, Ecotope attempted to duplicate a crossover blockage at Site M07. The first test at this site (labeled M07a) was run with the ducts in as-found condition. For the second test (M07b), the furnace was partially disassembled and a sheet metal flow diverter was attached to the trunk duct immediately under the furnace blast plate so that it blocked over half of the entry into the crossover duct run. The furnace was reassembled and the co-heat test re-run. Flows changed considerably to the B side of the home; however, the overall change in flow distribution was not dramatic enough to change the system efficiency by more than a couple of percentage points.

The steady-state efficiency model predicts this result. The conductive efficiency of the B side of the home decreases, but this decrease is offset by an increase in the conductive efficiency on the A side. The amount of duct air leakage is essentially unchanged. The overall effect of the modeling is a prediction that steady-state efficiency will not change much, and this is what was measured.

7. ANNUAL HEATING ENERGY IMPACTS

Table 7.1 summarizes furnace power measurements and expresses efficiency penalties in absolute terms rather than in percentages. The system efficiency results from Section 6 are applied to the actual power information to calculate power loss when the furnace is running.

TABLE 7.1: FURNACE POWER SUMMARY

Site ID	Full Power[1] [W]	Fan-On Power[2] [W]	Fan-On as % of Full[3]	Percent Ontime[4]	Average Cyc. Pwr[5] [W]	Efficiency Loss [%][6]	Efficiency Loss [W][7]
M01	11967	10727	83	16	1761	17	299
M02	15887	11909	74	33	3987	11	439
M03	20500	12430	60	33	4109	19	781
M04	12520	9983	81	50	5006	15	751
M05	15236	12197	79	29	3558	26	925
M06	8245[8]	7554	88	37	2773	21	582
M07a	22033	13248	60	23	3047	22	670
M08	16376	12471	76	24	2993	14	419
M09	15786	9471	60	28	2652	11	292
Average	15394	11110	75	30	3321	17	573
Std. Deviation	4223	1828	12	10	955	5.1	225

1 Total power of all electric resistance elements and fan.
2 Average power consumption of elements and fan while fan is on during furnace cycle.
3 Ratio of fan-on power to full power as percent.
4 Percent of time the air handler was on during a complete furnace cycle.
5 Fan-on power times percent on-time. This is the average power consumption during a complete furnace cycle.
6 100 minus system efficiency.
7 Efficiency loss (%) times average cycling power.
8 This furnace was on a dealer lot and operating with only two heating elements connected.

The first number in the table is the full-power measurement, which is taken during the steady-state heat delivery efficiency test. In this test, the furnace is run continuously for a sufficiently long period to level out the temperatures in the ducts; this consumption figure includes the air handler fan power. The next column of the table is the average fan-on power. This is the average power consumption of the furnace while the air handler fan is on, and is taken over all the furnace cycles used in the furnace efficiency analysis. This number is smaller than the full-power measurement because of furnace heating element sequencing and because the elements shut off before the fan goes off so that the fan can extract heat from the furnace cabinet and ducts according to the thermostat anticipator setting.

Sequencing impacts system efficiency; efficiency decreases when elements first come on because the temperature of the return air is colder than later in the furnace's operating cycle. The colder return air is due to a combination of lower temperature inside air (which causes the thermostat to call for heat) and incursion of cold outside air when the air handler comes on and depressurizes the building envelope. Some of the efficiency loss due to sequencing is recovered through the recapture of heated air in the ducts after the air handler turns off.

The last column in this table is the power loss, which is the product of efficiency loss and average cycling power. Power loss is the most important statistic in estimating the energy impacts of efficiency losses in manufactured homes. The average cycling power loss of these homes is 573 watts, ranging from 292 watts at Site M09 to 925 watts in Site M05. The power loss is a clearer method of expressing system efficiency losses than percentages. This is because different houses have different loads. Houses with large loads and small system efficiency losses can have larger power losses than houses with small loads and large system efficiency losses.

Table 7.2 predicts annual cost impacts of varying levels of heating system efficiency. The minimum, maximum, and average system efficiencies reported in Table 6.2 are applied to a 1400 ft^2 MAP home sited in Portland, OR and Boise, ID. This prototype home was used as the basis for calculating the final cost-effectiveness of MAP (Baylon et al. 1995). The energy use of the prototype home (excluding duct losses) is simulated with *SUNDAY 3.0* (Palmiter et al. 1987). System efficiency losses are then applied to the baseline energy use to calculate annual added costs.

Added yearly costs are generally modest for homes insulated to MAP specifications. This is especially true for a home with above-average system efficiency sited in a relatively mild climate such as Portland (4520 heating degree days, base 65° F, based on 1961-1990 data). A less efficient home sited in a colder climate such as Boise (5871 heating degree days, base 65° F, based on 1961-1990 data) can cost the homeowner considerably more over the course of a heating season: around $100 for the minimum efficiency case. As the home ages and duct air sealing products fail, annual costs of duct inefficiency will increase. Costs will also be higher in private utility service territories and in more severe climates.

TABLE 7.2: ANNUAL HEATING ENERGY AND COST ADDERS

Measured System Efficiency	value	Annual increase in Portland heating energy (kWh)	Annual increase in Portland cost	Annual increase in Boise heating energy (kWh)	Annual increase in Boise cost
Average	83%	773	$39	1161	$58
Minimum	74%	1326	$66	1991	$100
Maximum	89%	466	$23	700	$35

Assumptions

1400 ft^2 prototype home is built to MAP specs ($U_o = 0.0532$ Btu/hr °F ft^2).

Glazing percentage is 12% of heated floor area (168 ft^2)

Combined (natural + mechanical) infiltration rate of Portland home is 0.24 ACH; combined infiltration rate for Boise home is 0.29 ACH (based on Baylon et al. (1995) and location of homes).

The resulting UA of the Portland home is 261.3 Btu/hr °F; the UA of the Boise home is 270.9 Btu/hr °F

Thermostat is set to 67° F throughout the heating season with no setback.

Internal gains are set to 2500 Btu/hr.

Solar multiplier is set to 0.45 (combination of low-e coating on windows and intentional shading devices) and window area.

System efficiencies are assumed to be typical for the heating season and are applied to the base heating load to estimate added energy requirements.

Electricity cost is $0.05/kWh.

8. SUMMARY OF FINDINGS

This report presents the results of field measurements conducted on nine electrically-heated manufactured homes built to Model Conservation energy-efficiency standards and sited in the Pacific Northwest. The purpose of the research was to estimate the effects of forced-air heating distribution systems on heating energy requirements under typical winter conditions. The research was not conducted on a large enough sample to draw definitive conclusions; however, it is an important preliminary effort towards understanding the operation of forced-air systems in new manufactured homes.

The average and median system efficiency as defined by ASHRAE (1992), including heat recovered from buffer spaces and bypasses, is 83%. This means that manufactured homes built with high levels of underfloor insulation (R-33), duct insulation, and displaying limited levels of duct leakage (about 6% of the air handler flow, on average), use on average 1.20 times as much heating energy as they would if heated with zonal electric baseboard heaters.

Average measured system efficiency for these homes is considerably better than that found during a study of 24 Pacific Northwest site-built homes tested with a very similar protocol during the 1992 and 1993 heating seasons. That study found an average system efficiency 71% for the 22 homes which had at least half of the ductwork located outside of the thermal envelope. The site-built homes had ducted return systems and much longer and leakier supply systems than the manufactured homes in this study. Two homes in the site-built study had all ducts and the furnace located inside the thermal envelope of the home; these homes had an average system efficiency of 98%. Even though the manufactured homes in this study have furnaces located inside the home's thermal envelope, duct conductive and air leakage losses and increased whole-house infiltration induced by the operation of the air handler fan reduce overall system efficiency considerably.

During 1995, a review of 14 homes built to 1994 HUD standards and sited in New York and North Carolina was undertaken. These homes had lower levels of belly insulation (R-19) than MAP homes and a larger average supply leakage fraction (about 11%). A steady-state heat delivery efficiency test was conducted in these homes and averaged 63%. This result suggests that the impact of duct sealing and insulation standards in MAP homes has a large effect on overall duct efficiency when compared to the national minimum standard of manufactured home construction and duct installation.

The cross-over duct used in multi-section homes almost certainly results in a significant fraction of the duct losses in some cases. The mechanism for problems with the cross-over varies somewhat. In one test case, this mechanism appears to be a function of interaction between the cross-over and the air handler, resulting in imbalanced air flow. Unless the imbalance is very large, however, overall system efficiency effects seem limited to only a few percentage points.

Installation shortcomings in manufactured homes can result in drastic changes in system efficiency. Ducting systems in manufactured homes are not designed with standard HVAC engineering and therefore it is crucial that air leakage is kept to a minimum and insulation standards are upheld if acceptable system efficiencies are to be maintained.

These houses were found to be very tight, with natural effective air change rates averaging only 0.12 ACH. This air change rate is much lower than that recommended by indoor air quality performance standards such as ASHRAE Standard 62. The exhaust-only ventilation systems for these homes are not designed for continuous operation. If recommended air exchange rates are to be maintained throughout the year, alternative ventilation systems should be installed in homes of this type.

The protocol used in this study provides a reliable estimate of overall heating system efficiency. However, disaggregation of conductive and air leakage losses is still poorly understood and will require additional protocol development, field testing, and analysis.

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Mark Jackson of the Bonneville Power Administration managed the testing contract for the first six homes. Eileen Jensen of Bonneville managed the testing contract for the final three homes.

APPENDICES

A. MAP Heating System Protocol
B. Floor System Construction Details
C. Ventilation in Energy-Efficient Manufactured Homes
D. Efficiency Model

APPENDIX A: MAP HEATING SYSTEM PROTOCOL

Date(s) of Visit________________________________

Field technicians________________________________

Audit form filled out by__________________________

Occupant Information

Name

____

Address

____

City ________________ State ____________

Phone# Home

____

Work

____

Occupant Questions

Be sure to get the letter of agreement signed.
Do you have any paperwork on the furnace? (Get it from the homeowner. Take a copy or get the document itself, promising to send it back).
If the thermostat is digital, get the operations manual. (Important!)
Do you have floor plans for this home? May we borrow them, please?
How many people live in this home?__________________________
Have you noticed any odor or moisture problems?____________________________
What manual setbacks do you normally use?________________________
During which seasons is the air handler operated continuously?____________________

____
Do you know where your system filter is? ______________________
How often do you change it? ______________________________
When was the last time you changed it? ______________________
Window vent customary settings ____________________________

MAP HEATING SYSTEM PROTOCOL

Initial Diagnostics and Setup

Synchronize technician watches, dataloggers and computers.
Pressures across the envelope (reference house):

Measurement Location	ΔP AH Fan Off	ΔP AH Fan On

Sketches

_______________ Draw a rough floor plan of each level of the home, noting locations of supplies, returns, air handler, and exhaust fans. For heat pumps, include location of outside unit and refrigerant lines.

_______________ Number each supply register with a different number.

_______________ Draw a rough elevation view of the home, noting the height of ceilings, cathedral peaks, etc.

_______________ Take pictures at the four main directions of the house. There should be shots looking toward and away from the home.

_______________ Take pictures of air handler and any noteworthy interior or exterior details.

Home Description - Exterior

Skirting vents: Number open__________ Number closed__________
Orientation of long axis of house

____

Local topography

____

Altitude _____________________
Describe shielding and terrain for each of the four main directions

Other notes (condition of skirting, unusual obstacles, etc.):

MAP HEATING SYSTEM PROTOCOL

Home Description - Interior

Make/model __________________
Floor Area __________________
Volume __________________
Floor type __________________

Number of bedrooms __________________
Number of bathrooms __________________
Number of other rooms __________________
HUD #s __________________

Other notes

Ventilation System

Intake (include number, location and height of intake vents)

Exhaust

Controls (include humidistat or timer settings)

Makeup air system

Deactivate ventilation controls. Note disabling done:

Position of window vents

MAP HEATING SYSTEM PROTOCOL

Thermostat

Thermostat location

____

Thermostat make and model

____

Fan-only switch ________Yes ________No

Anticipator setting ________amps

Measure control circuit amperage:
1) Remove thermostat
2) Put Fluke ammeter between terminals R and W on sub-base
3) Let furnace run for one minute and read amperage ________

Set anticipator to amperage read
List all thermostat programs as found:

List programs as tested:

Quick Air Flow Measurement

1) Let the air handler run for 5 minutes.
2) Measure the supply temperature ____________
3) Measure the return temperature ____________
4) Measure the power through both hearing circuits with the power meter: ______watts
5) If supply air temperature is greater than 140°F and airflow is low, check whether the air handler is cycling on the limit switch.

Air flow calculation:

ΔT = Supply T - Return T = ______________
(______ Watts/ ______ ΔT (F)) x 3.16 = ____________ SCFM

or
(______ Watts/ ______ ΔT (C)) x 1.75 = ____________ SCFM

MAP HEATING SYSTEM PROTOCOL

Heating System

Flow type: ______ Upflow ______ Downflow ______ Sideflow
Locate breakers for furnace on each leg of the circuit. Note breaker amperage.

Nameplate Information
Record all information on all name plates (furnace, fan motor, electronic air cleaner, etc.)
Make and model ____

Maximum outlet temperature ____
Additional nameplate information:

Resistance heater make and model ____
Resistance heater power rating ____
Electronic air cleaner make and model ____

Filters

Describe filters

____
____
____
(Number of filters, physical arrangement in air handler, etc.)

Filter condition ____
(Clean, OK, dirty, very dirty, impenetrable [inspect by holding up to light])

MAP HEATING SYSTEM PROTOCOL

Air Handler Fan
From nameplate:

Blower make/model ______________________ Fan power rating ______________
Fan flow rating(s) ______________________ Full load amps ______________

Measure fan motor with air handler cover plate closed by installing Sperry on furnace mains (or separate circuit for fan, if available).

Amps (rms) ______________________ Volts (rms) ______________________
Watts (rms) ______________________ Power factor ______________________

Number of taps connected to fan motor __________

If more than one tap is connected, determine whether the fan operates at different speeds with the heat on and off, and note below:

Make-Up Air System
From nameplate:

System type ______________________
Fan flow rating ______________________

Measure flow under normal system operation with flow grid installed just above return side of system.

Flow ____________________________________

MAP HEATING SYSTEM PROTOCOL

Air Handler Pressures

Measure the pressures across the air handler cabinet with the fan running. Use the static pressure tap and measure through a crack in the air handler cabinet, through a hole created by removing a screw, or through a drilled hole. The positive side of the manometer should measure into the air handler. Measure with make-up air system enabled and disabled.

	Before filter cleaning	**After filter cleaning**
Supply pressure with make-up air system operating		
Supply pressure with make-up air system operating		

Cabinet and Plenums

Description of furnace cabinet & plenums (give dimensions):

Describe insulation in cabinet, if any:

Additional details:

MAP HEATING SYSTEM PROTOCOL

Supply Registers

Position: Record the position of each register (open, partially open, closed). Open the register fully. Check the "restored" column when you have put the register back to its initial position.

Flows: Measure supply flows with the air handler running *and with the heat on.*

Measurement start time: ____________
Measurement stop time: ____________

Register #	Location	Position	Flow hood used	Flow (cfm)	Restored
				Total Flow	

MAP HEATING SYSTEM PROTOCOL

Ductwork

Total ductwork footage___________

Supply Ducts

Pull off supply registers and inspect for cracks. Use a flashlight and a mirror. Notes:

Description of crossover duct connections (if accessible):

Other supply duct notes

Pressures Across Bedroom Doors

Measure pressures across bedroom doors (positive tap inside bedroom, negative outside). Only one interior door closed at a time!

Room	P, AH off	P, AH on	Undercut

MAP HEATING SYSTEM PROTOCOL

Tracer Test
Injection points:

Sampling points:

Background

Turn on B & K and measure all six gases (with compensation for H_2O) for about one hour.

Time started_________________
Time finished_________________

Decay

Decay tests required are:
- Fan on
- Testing registers unsealed; fan off

Turn on B & K and measure SF_6 with compensation for H_2O.

Time	Conc (ppm)	ΔT	Description & Notes
			Start injecting, fan on
			Stop injecting; leave fan on until concentration drops 10%
			Turn fan off
			Wait until conc drops additional 10%
			End of test

Datalogger Setup

Select temperature control zones and install thermocouples and co-heaters. Put TCs in all registers, in the supply plenum, the furnace closet, and outdoors. Also measure temperatures in the crawlspace and belly blanket (center section and outrigger section).
List channels on each datalogger in the tables on this and the next page.

Master Datalogger

Channel #	TC#	Description

Slave Datalogger

Channel #	TC#	Description

MAP HEATING SYSTEM PROTOCOL

Furnace and Co-heat Tests

After running the LOAD program to set up the computer, and after all the equipment is set up, experiment with setting the setpoint and deadband.

Make sure all non-essential loads are disconnected: domestic hot water, refrigerator, make-up air, freezer well, sump pump, and exterior lighting. When the furnace is cycling about six times per hour, start averaging zone temperatures. **Make sure all co-heaters come on and oscillate properly**. Then load the HEATAG program, which will automatically run furnace and co-heaters for alternating 2-hour periods throughout the night.

Time	Setpoint	Deadband	Furnace Cycles/Hour

Co-heat Control Points

TC Location	Heater Setting (W)	Notes

MAP HEATING SYSTEM PROTOCOL

BLOWER DOOR AND DUCT BLASTER TESTS

1) BLOWER DOOR TEST

Set-up: Close all windows and doors to the outside. Open all interior doors and close all dampers and doors on wood stoves and fireplaces. Make sure that furnace and water heater can not come on during test. Make sure all fans are off.

Reference Pressure______ Plug fan hole and record pressure difference across the door.

Turn blower door on and depressurize house to 25 Pa from reference pressure.

Negative house pressure: _______ Pa
Ring size (O,A,B,C): _______
Flow pressure: _______ Pa
CFM: _______

Turn blower door on and depressurize house to 50 Pa from reference pressure.

Negative house pressure: ______ Pa
Ring size (O,A,B,C): ______
Flow pressure: ______ Pa
CFM: ______

2) AIR HANDLER FLOW MEASUREMENT

Set-up: All zones with duct work opened to outdoors and each other, where possible. Turn on air handler with fan switch.

Reference static supply pressure: ________ (from Page 7)

Use Duct Blaster as helper fan into supply cabinet. Use Duct Blaster to match reference static supply pressure with air handler running. Record Duct Blaster flow pressure and CFM.

Supply pressure: ______ Pa
Ring # ______
Flow pressure: ______ Pa
Table CFM: ______

3) SUPPLY AIR LEAKAGE TEST

Set-up: Return opening sealed in blower compartment
Duct Blaster attached to blower access opening as in Test 1
All zones open to outside or each other
All registers sealed with tape and paper
Remove blower from furnace (make sure breakers off) or fit cover over fan plenum and attach Duct Blaster.

Total leakage (@50 Pa and 25 Pa)

Pressurize supply system to about 50 Pascals; measure near supply plenum with smallest ring possible using the pressure tap previously installed. (use 10 second average)

Supply pressure ____________
Ring # ________________
Flow pressure ________________
CFM ________________

Pressurize supply system to about 25 Pascals, measure near supply plenum with smallest ring possible (use 10 second average)

Supply pressure ____________
Ring # ________________
Flow pressure ________________
CFM ____________________

Supply Air Leakage Test, con't.

Supply Leakage to outside (@ 50 Pa and 25 Pa)

Set-up: Blower door set to pressurize heated space
Internal doors of heated space open
Connections to outdoors from heated space closed

Pressurize house to about 50 Pascals. Pressurize ducts to about 50 Pa WRT outside. Check pressure between ducts and house; adjust BD and DB speed controllers until this pressure differential is near 0. Recheck house pressure; re-adjust as necessary.

Supply pressure ______________
Ring # ____________________
Flow pressure ____________________
CFM __________________________

Repeat preceding test at 25 Pascals.

Supply pressure ______________
Ring # ____________________
Flow pressure ____________________
CFM __________________________

APPENDIX B: FLOOR SYSTEM CONSTRUCTION DETAILS

Construction Techniques:

Manufactured home floor systems differ from site-built home floors in two important respects. First, the floor is framed on top of a steel undercarriage made of steel I-beams (which run the length of the home) and outriggers (placed several feet apart, which run from the outer I-beam to the perimeter of the home). This undercarriage is the means by which the structure is towed from the factory to the home site. In the most commonly found construction techniques, the crushing of insulation batts between the floor joists and the undercarriage reduces the performance of insulation. Second, heating system ducts run in the floor system. The ducts compromise the thermal performance of the floor.

Three floor system construction techniques are described here. The transverse floor, in which the floor is framed upside down and is perpendicular to the steel undercarriage, is the most common technique found in the Northwest. The second framing scheme, the upright transverse floor system, is similar, but the joists are placed on top of the steel undercarriage after the belly blanket is laid out and the rest of the insulation is placed from above. The longitudinal floor is framed so that the floor joists run parallel to the steel I-beams and are supported with steel stays which run orthogonally to the I-beams.

Ducts are generally made from thin gauge sheet metal which is shaped and crimped at the manufacturing plant. Duct insulation (fiberglass) is usually nominally R-5, although it is crushed somewhat during installation.

Transverse Floor System:

Most Northwest manufacturers construct a floor structure which is called "transverse." In this configuration, 2 x 6 (or 2 x 8) floor joists are placed at right angles to the steel I-beams and parallel to steel outriggers (which extend from the I-beams to the rim joists) that make up the undercarriage of the home. The floor is assembled upside down, with the heating ducts, plumbing lines, and electrical service located in the center ("belly") portion of the floor system. A "belly blanket" (one or more layers of insulation) is placed over the floor framing and the various utility conduits, then covered with a reinforced plastic sheet called the "belly board" (or "bottom board"). The steel undercarriage is placed on top of this layer and strapped and bolted to the joist assembly. The entire system is then flipped back over and the interior flooring and heating registers are installed.

The belly blanket insulation is brought up into the joist cavity in the outrigger region (the outer 3 feet of each side of the double-wide) by cutting the batts where they come up against the joists and pushing the batts into the joist cavity. This strategy reduces the compression which normally occurs when the belly blanket is crushed between the steel undercarriage and the joists. (The compression is not totally eliminated, because the bottom layer or layers of belly insulation are usually still crushed between the floor joists and I-beams.)

The belly blanket is crushed at the edges of the home and between the steel I-beams and the floor joists. Normal density joist insulation may be compressed depending on the framing depth and the insulation thickness. For example, an R-19 batt is usually 6" thick. If an R-19 batt is placed in a 2 x 6 frame, where the cavity space is only 5-1/2", compression reduces the performance of the insulation by 5.3%. The R-21 high-density batt has an uncompressed thickness of 5 ½", so it will retain full R-value in the 5 ½" joist cavity. At the home site, home sections are joined mechanically, and an insulated flexible crossover duct, which runs below the belly board, connects the two supply trunk ducts in each section.

Upright Transverse Floor System:

The upright transverse floor is framed similarly to the more common transverse floor, except the floor is framed on top of the undercarriage, rather than the other way around.

Typically, the belly blanket insulation is laid on top of the belly board before framing the ends. Then the space between the joists is insulated with insulation that extends across the floor structure and under the supply duct. Since the belly blanket is continuous across the floor, additional layers are placed in the center section of the steel undercarriage (between the I-beams) before the duct is placed in the frame. Then, the floor framing is put in place, compressing the belly blanket where joists intersect with the steel carriage. A partial layer of insulation is placed in the floor cavity space from above before the flooring is installed. This last layer of insulation is cut so that the insulation does not extend over the duct but falls down into the cavity space above the belly blanket. In order to assist heat recovery in the event of duct leakage, the joist insulation should not cover the duct at all. Conductive losses from ducts are then more easily recovered into the home's interior.

Longitudinal Floor System:

In the longitudinal configuration, the floor structure is constructed similarly to the transverse floor; however, the floor joists are placed <u>parallel</u> to the steel undercarriage I-beams. Each heating duct is located inside one joist cavity (rather than being strapped to the bottom of the joists and perpendicular to them).

In this case the floor is framed upright, similar to the upright transverse floor. The belly blanket is draped over the steel carriage and framing is placed on top of the belly blanket, compressing it. Batt insulation is placed in all joist cavities except the one containing the duct. In the center of each house section, a single joist cavity is given over to the duct.

Thermal Analysis

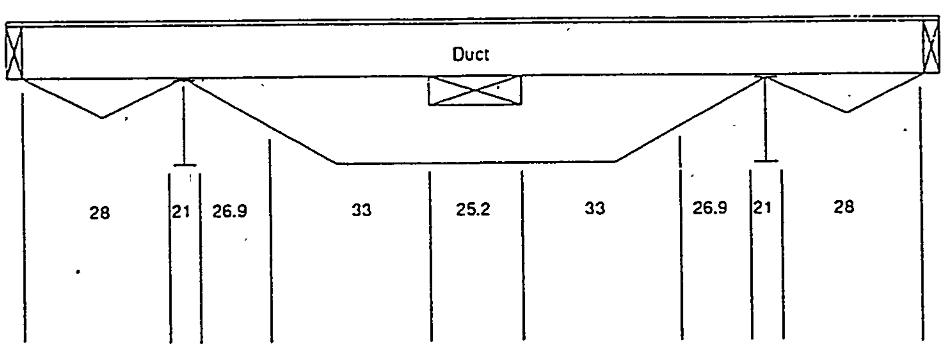
To analyze thermal performance of floor systems in manufactured homes, we used a one-dimensional parallel heat flow analysis. Compression (and subsequent reduction in performance) of belly blanket and joist insulation is factored into the calculation.

A framing correction is made, based on average dimensions measured in the plants. These average measurements are an out-to-out floor box width of 156" and a distance of 30" from the outside of the rim joist to the middle of the steel I-beam. The buffering effect of the crawlspace is also factored into the U-value calculation, assuming a ventilation rate of 3 ACH.

Insulation Compression and Conductance:

The belly blanket conductance varies due to compression of the insulation. There are several distinct compression regions for the belly blanket, each corresponding to an area of the floor, and each with a different conductance. Belly blanket insulation is compressed between the belly board and the underside of the joists in the outrigger region, between the I-beams and the joists, and between the belly board and the bottom of the heating duct (in some cases).

The compression for different insulation configurations was determined from site visits and some simplifications were made (for example, assuming a uniform slope of the insulation in compression zones, as shown in the following figure). Figure 1 shows the compression regions with corrected R-values for an R-33 transverse floor. This figure extends from the rim joist (at left) to the other side of one half of a double-wide home. The R-values are a combination of belly and (where applicable) joist insulation.

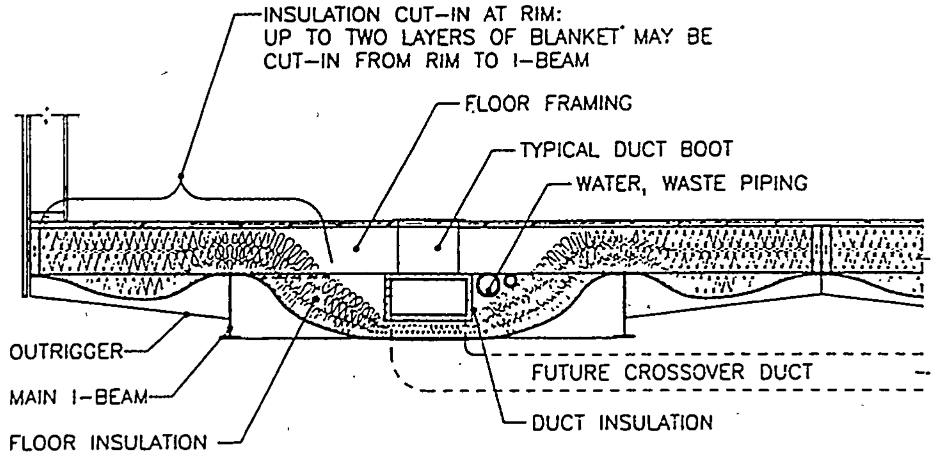


Numbers are the average R-value calculated for each compression zone

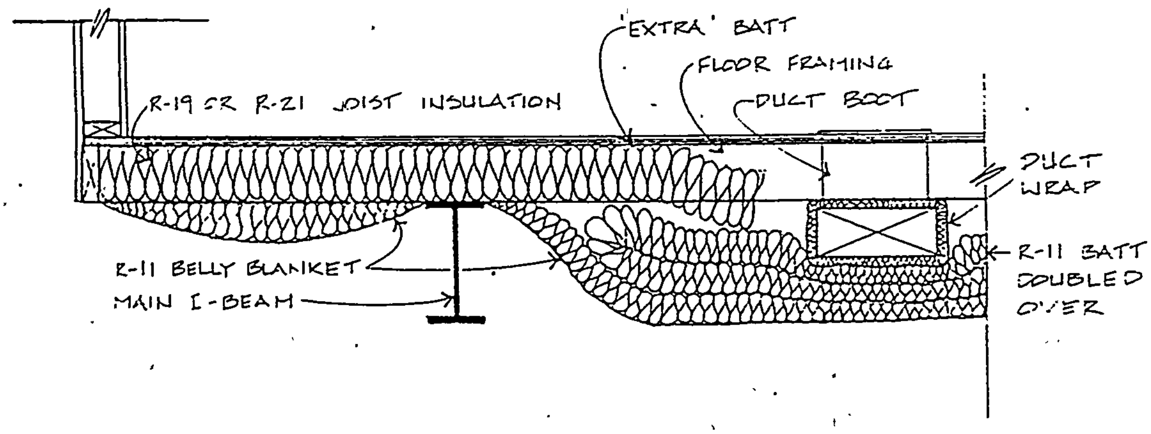
INSULATION COMPRESSION REGIONS IN R-33 TRANSVERSE FLOOR SYSTEM

In cases where the insulation compression in the region was less than or equal to 50% of the nominal thickness of the batt or batts, an equation developed at Ecotope was used to find an R-value. In cases where compression exceeds 50% (where insulation is crushed between joist and I-beam, for example), information from manufacturers was used to estimate the R-value.

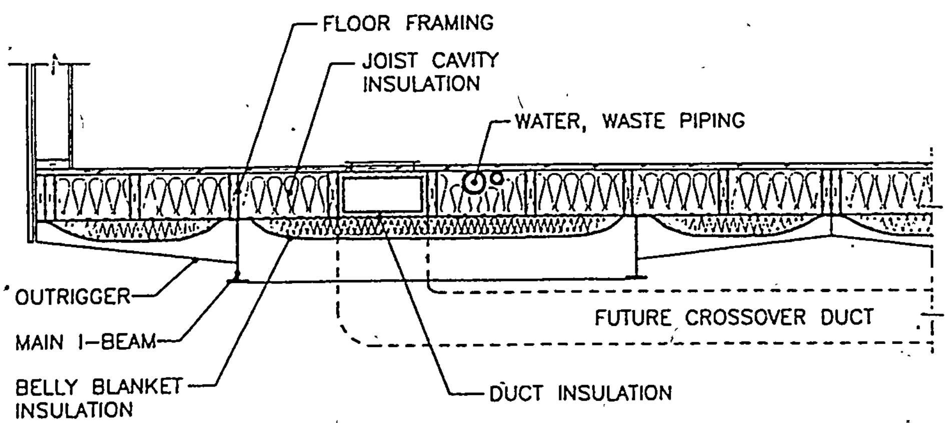
The R-values were calculated for the joist and belly regions in each compression zone. The reciprocals of the values were area-weighted by each compression zone to find the overall steady-state floor U-value.



TRANSVERSE FLOOR SCHEMATIC



UPRIGHT TRANSVERSE FLOOR SCHEMATIC



LONGITUDINAL FLOOR SCHEMATIC

APPENDIX C: VENTILATION IN ENERGY-EFFICIENT MANUFACTURED HOMES

Background

Ventilation of energy-efficient manufactured homes in the Pacific Northwest has followed a tortuous path. This is because of changing specifications, cost and physical limitations issues, and the fundamentally confusing nature of residential ventilation.

Effective dilution and removal of indoor air pollutants is the intent of air quality specifications and standards such as ASHRAE 62, but this intent is not explicitly stated in HUD or MAP specifications. The systems which have most often been installed to meet these specifications do not provide long-term effective ventilation to manufactured home dwellers.

Energy codes began mandating increased insulation levels and decreased unintentional air leakage starting in the late 1970s. As codes have become stricter, more attention has been paid to measuring the air tightness of buildings, both through the use of blower doors and tracer gas tests. These studies have shown tighter buildings (with or without ventilation systems) do have the potential to cause health and comfort problems because increased tightness of the building shell and predominantly intermittent operation of ventilation systems can lead to indoor air quality concerns.

To mitigate potential health problems which could be associated with reduced indoor ventilation rates, regulators decided that new codes and specifications should not reduce the capacity of people to ventilation their homes below the average level of ventilation of typical homes being built at that time. After a series of field tests, the average ventilation rate of 0.35 air changes per hour (ACH) was settled on as a representative value. The American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) adopted the same figure as a recommended ventilation standard (Standard 62) for maintaining adequate residential indoor air quality (ASHRAE 1989).

When the original Super Good Cents site-built specifications (BPA, 1987) were developed, specific procedures were mandated to tighten the buildings' construction, thereby reducing the natural infiltration rate. These specifications were transferred more or less intact into the ventilation specifications for the Super Good Cents manufactured home program, with the addition of some language for set-up standards. The SGC specifications also mandated ventilation systems which were designed to have the capacity to ventilate at 0.35 ACH and were to be controlled automatically. These specifications, in turn, carried over into the MAP when it began in April, 1992. The specifications implied that air sealing procedures would be such that homes would be sufficiently air tight that additional ventilation would have to be provided at some times with mechanical systems to both remove occasional pollutants and maintain healthy ambient air conditions..

Many indoor air pollutants are associated with the level of occupancy. If we are concerned with CO_2 levels, or biocontaminants which can transmit disease from one person to another, our ventilation specifications should be governed by the number of occupants in a building. Relative humidity levels are also highly correlated with occupancy levels, as more people will bathe, cook, wash, and emit more water vapor.

The value of 0.35 ACH, however, is based on the volume of the building. The volume of the building has little or no correlation with the number occupants living there. The size of a home tends to be strongly correlated with family income but only weakly correlated with the number of occupants. Under a volume-based specification, a very large home with two occupants must be ventilated at a much higher volume than a small manufactured home with four occupants.

The SGC and MAP ventilation specifications for manufactured homes are not directly home-volume based. They are, in fact, based on the number of bedrooms in the home, which is more closely correlated with number of occupants. This type of specification is still somewhat inexact, in that bedrooms can vary in size; however, in manufactured homes this variation is not at large as in site-built homes. The fact that minimum prescriptive or performance specifications are based on number of bedrooms does not change the fact that system sizing is still based on the figure of 0.35 ACH.

The MAP ventilation specifications do not mandate that a ventilation level of 0.35 ACH is actually met on a day-to-day basis, only that a home's ventilation system have the capacity to reach this level. Therefore, a single large fan, set to operate for 15 minutes a day, could meet the specification even though the fan will have a negligible impact on the total ventilation rate and the home may have a natural ventilation rate far below 0.35 ACH.

Different types of indoor air pollutants require different methods for their removal, since they are produced by different sources. The ideal ventilation system design depends on which pollutants are to be removed. If the primary concern is with pollutants associated with human occupancy, such as CO_2, biocontaminants, or cigarette smoke, then ventilation should coincide with occupancy, and the ventilation rate should be increased depending on the number of people present. If we are concerned with high humidity levels which can lead to fungi (mold) growth or dust mite proliferation, air should be exhausted at high volumes from areas where most of the moisture is generated (e.g. bathrooms during and after bathing and from the kitchen during meal preparation). If the primary concern is with carcinogens or irritants which are emitted continuously by the environment (radon), the building (formaldehyde) or occupants who smoke and who are home most of the time, then the home should be ventilated continuously at low volumes. This concern extends also to tight homes where the occupants complain of "stuffiness," even though no contaminants may be present. What constitutes stuffiness is probably a matter or personal preference, but it has to do in part with elevated CO_2 levels.

Average Ventilation and Effective Ventilation

Continuous low volume ventilation and intermittent high volume ventilation have very different effects on indoor air quality, and should be used for different purposes. The physical explanation for this was first discussed by Larry Palmiter and Ian Brown of Ecotope, Inc. (Palmiter and Brown, 1989). To understand the difference, we need to distinguish between two different types of ventilation rates: the *average ventilation rate* and the *effective ventilation rate*.

It is easiest to explain the difference with an example. Consider two identical manufactured homes. One house is ventilated continuously (24 hours/day) at a rate of 0.35 ACH. The other is ventilated at 0.20 ACH for 23 hours/day and then for one hour every day a large exhaust fan comes on and ventilates the building at 3.8 ACH.. In terms of average ventilation rate, both of these buildings are equivalent.

<u>Average Ventilation Rate:</u>

Home 1	Home 2
0.35 ACH for 24 hr/day	0.2 ACH for 23 hr; 3.8 ACH for 1 hr
$\frac{(0.35ACH)24hr}{24hr} = 0.35$ ACH	$\frac{(0.2ACH)23hr + (3.8ACH)1hr}{24hr} = 0.35$ ACH

The average ventilation rate is what is measured by a flowhood and governs the amount of heat loss associated with air leakage. The average leakage rate is typically the value discussed when we speak generally of ventilation rates.

There is another measure of ventilation known as the *effective ventilation rate*, which is defined as the inverse of the steady state concentration of a pollutant with unit source strength. Although it may not be clear from the language of air ventilation specifications, the effective ventilation rate is what is referred to in these specifications. The effective ventilation rate is that which is measured by tracer gas tests; this rate governs the removal of pollutants from the air.

Assume that our two example homes each have a continuous pollution source which is emitting pollution at a rate of one cubit per hour. (The cubit is an imaginary unit which is used for purposes of illustration only.) The ability of the two ventilation schemes described above to remove this air pollution is very different.

<u>Average Pollutant Concentration (assuming a pollutant source strength of 1 cubit):</u>

Home 1	Home 2
$24hr\frac{(1cubit / 0.35ACH)}{24hr} = 2.857$ cubits per air volume	$\frac{23hr(1cubit / 0.2ACH + 1hr(cubit / 3.8ACH)}{24hr} = 4.803$ cubits per air volume

So, the average daily pollution concentration in the intermittent case nearly twice as high. The effective ventilation rates for the two cases are:

Effective Ventilation Rate:

Home 1	Home 2
$\frac{1}{2.857} = 0.35$ ACH	$\frac{1}{4.803} = = 0.21$ ACH

Tracer gas (PFT or SF_6) testing in these manufactured homes would reveal an air change rate of 0.35 ACH in Case 1 and 0.21 ACH in Case 2. However, if we were to calculate the heat loss associated with the ventilation, both cases would be equivalent. Note this analysis assumes perfect air mixing inside of the building. If all of the indoor air pollution were to occur during the one hour of high volume ventilation, then Home 2 would achieve superior air quality; however, if the pollution is distributed over time, the ventilation in Home 1 is far superior.

This discussion has defined *average* and *effective* ventilation. Two other terms which should be defined, generally implicitly assumed to be understood are *natural* and *mechanical* ventilation. *Natural* ventilation refers to ventilation which is caused by wind and stack (buoyancy) effects only; this type of ventilation occurs through unintentional penetrations in the building envelope and neglects any contribution from mechanical equipment. This type of ventilation is what is estimated using blower door and/or tracer gas tests. *Mechanical* ventilation is ventilation added by exhaust fans, furnace make-up air systems, balanced flow (two fan) ventilation systems and the like.

Another aspect of ventilation in buildings which is often overlooked is the interaction of natural and mechanical ventilation. In the past, the volume of air flow created by mechanical ventilation systems has typically been assumed to add to the ventilation provided by natural ventilation. Palmiter and Bond (1991a) developed a simple model, based on extensive empirical (tracer gas) evidence, which takes into account the interaction between these two types of air movement.

The model examines unbalanced fan flow, such as that exhibited by standard exhaust fans. The flow is unbalanced because a supply fan of the same size as the exhaust fan is generally not present. The model reliably estimates the amount of ventilation added to building by operation of the fan, taking into account the building is already exchanging some of its air with the outside environment because of natural forces (stack effect and wind).

The additional ventilation provided by the exhaust fan is one-half of the flow measured through the fan is the exhaust fan flow is less than twice the natural infiltration rate. Otherwise, the added ventilation is the difference between the fan flow and the natural infiltration rate.

In other words, if the flow produced by the exhaust fan is more than twice the natural infiltration rate, then the total ventilation rate will simply be the flow through the fan. If the exhaust fan flow is less than twice the natural infiltration rate, then the total ventilation rate will be half of the fan flow plus the natural infiltration rate.

The reason for this effect is that the fan must overcome the stack pressure distribution within the building. The exhaust fan flow must be strong enough to reverse the flow through the highest leaks in the building before it will completely dominate the ventilation of the building. The model assumes a uniform distribution of leakage area, and a neutral pressure level half-way up the outside walls.

Shown below is a table which pairs fans of different sizes with different homes in order to determine average and effective ventilation rates under various fan duty cycles. The natural/mechanical ventilation interactive effect is calculated in each case. The last entry under each home size (shown in *italics*) is the continuous ventilation option. In each case, the fan size meets the performance specification for continuous ventilation. These performance specifications are shown in the second table. Note that in nearly every case, the fan as specified would meet MAP or HUD requirements for exhausting air at a rate of 0.35 ACH (as shown in the last column of the table); however, the effective ventilation rate is very different depending on how many hours per day the fan runs.

Table C-1
Fan Run Times and Effective Ventilation Rates
(assumes natural infiltration rate of 0.20 ACH)
(continuous ventilation scenario in italics)

House Size (ft^2)	Approx. House Volume (ft^3)	Fan Size (certified flow at 0.10" W.C.)	Fan Run Time (hrs/day)	ACH Effective	ACH Heat Loss	ACH when fan running
1000	8000	90 cfm	8	0.26	0.36	0.68
		70 cfm	12	0.29	0.37	0.53
		50 cfm	18	0.32	0.34	0.39
		50 cfm	*24*	*0.39*	*0.39*	*0.39*
1500	12000	90 cfm	12	0.28	0.33	0.45
		90 cfm	14	0.30	0.35	0.45
		110 cfm	12	0.29	0.38	0.55
		70 cfm	*24*	*0.38*	*0.38*	*0.38*
2500	20000	90 cfm	16	0.28	0.29	0.34
		90 cfm	20	0.30	0.32	0.34
		110 cfm	16	0.29	0.32	0.37
		90 cfm	*24*	*0.34*	*0.34*	*0.34*

In most cases, the half-flow rule is used to add exhaust fan flow to the specified natural infiltration rate of 0.20 ACH.

APPENDIX D: EFFICIENCY MODEL

The magnitude of energy loss due to forced-air distribution systems depends on complex interactions between conductive losses and duct air leakage. The interaction of duct-induced pressure imbalances and natural infiltration can also result in increased energy usage during the heating season. This subject has received some study in the past several heating seasons, and measurements from these studies have informed attempts to mathematically model the interactions and calculate an overall heating system efficiency..

For several years, Larry Palmiter of Ecotope has directed the development of mathematical model which can calculate heat delivery efficiency based on physical details of a heating system (duct area, insulation, air temperatures in and around the duct during operation) and measurement of whole-house and duct air leakage. The model is intended to calculate steady-state heat delivery efficiency (that is, it does not consider off-cycle losses).

The model is based on first principles of mass flow, heat balance, and fluid dynamics, and was initially developed as part of work conducted with Lawrence Berkeley Laboratories (LBL) for the Electric Power Research Institute (EPRI) in the late 1980s. The early version of the model is summarized in Palmiter and Bond (1991). The form currently in use (Palmiter and Francisco 1996) was presented at the 1995 ASHRAE spring meeting. During the last few years, the model has been calibrated repeatedly with detailed data from short-term coheat tests. The full model is currently in publication and is summarized in very condensed form here. The interested reader should obtain a copy of the Palmiter and Francisco report later in 1996 to follow the general development of the model and examine test cases.

The model includes factors for duct conduction losses, duct air leakage, and a factor which accounts for the interaction of the air handler fan with the home's natural infiltration rate. The model's full form is shown below. Although there is no ducted return system, the middle term is included for completeness:

$$\eta = \alpha_s\beta_s - \alpha_s\beta_s(1-\alpha_r\beta_r)\frac{\Delta T_r}{\Delta T_e} - \alpha_s(1-\beta_s)\frac{\Delta T_s}{\Delta T_e} - \varepsilon_{in} \qquad (1)$$

Where

η	=	overall steady-state heat delivery efficiency of the heating system
α	=	air leakage efficiency of the supply ducts, equal to (1 - (supply leakage fraction))
β_S	=	conductive efficiency of the supply ducts, found from physical measurements of the duct system and the flow rate of air in the duct. This value is calculated, register-by-register, for each piece of duct and then is weighted by the measured register flows. The crossover duct is included in the overall conductive efficiency.
ΔT_s	=	the temperature difference (in °F) between the home's interior and the belly space during the steady-state heat delivery efficiency measurement
ΔT_e	=	the temperature rise across the furnace during the steady-state heat delivery efficiency measurement

Removing the term for the return system (since there is no ducted return in a manufactured home), the resulting equation is

$$\eta = \alpha_s\beta_s - \alpha_s(1-\beta_s)\frac{\Delta T_s}{\Delta T_e} - \varepsilon_{in} \tag{2}$$

The supply leakage fraction ($1 - \alpha_S$) is based on the Duct Blaster™ measurement of exterior duct leakage (last column of Table 5.1). The model assumes all air leaks occur at the end of duct runs instead of at the furnace. If we assume all leaks occur at the air handler, the predicted overall efficiency typically decreases by 1-3%.

The extra infiltration induced by air handler operation (ε_{in}) is estimated based on the blower door tests and a mathematical correction from measurement conditions (where the house is subjected to a pressure gradient of around 25 or 50 Pa) and normal winter conditions (where pressure differentials across the envelope are usually much less than 5 Pa). In most cases, the ACH_{50} air leakage is divided by 27 to estimate ACH_{nat}. The divisor of 27 (rather than 20) is used based on a comparison of blower door and long-term PFT results from a study of Pacific Northwest manufactured homes built to near-MAP thermal and air-sealing standards in the late 1980s (Palmiter et al. 1992). Manufactured homes have limited low and high leaks and are one-story structures; therefore, stack-only air leakage in these homes is generally over-predicted by simply dividing the ACH_{50} by 20.

The house infiltration rate estimated with this method is compared to the estimated duct leakage at normal furnace operating conditions and a combined natural/mechanical infiltration rate is calculated. Duct leaks cause depressurization of the home because air which leaks from the ducts into the belly area and crawlspace must be replaced by "return" air (which, in a manufactured home, comes from inside the home and from intentional air leaks into the home (such as the make-up air duct) and unintentional leaks (collectively known as "infiltration").

The combined rate depends on the relative sizes of the natural and mechanical infiltration. Detailed measurements by Ecotope (Palmiter and Bond 1991) have determined that simply adding the measured duct leakage to the natural infiltration rate does not produce a correct combined infiltration rate for determining heat loss. Instead, the measurements have found that the combined rate is equal to the natural infiltration rate plus half of the measured duct leakage until he duct leakage is twice the natural infiltration rate. At this point, the combined infiltration is simply equal to the duct leakage.

In the measurement protocol undertaken in this report, all temperatures could not be measured. This is some cause for concern; however, sensitivity analysis on this issue has shown the overall efficiency predicted by the model is much less sensitive to the ratio $\Delta T_s/\Delta T_e$ than to changes in the estimated B_s or α_s, which are much better determined by this measurement protocol.