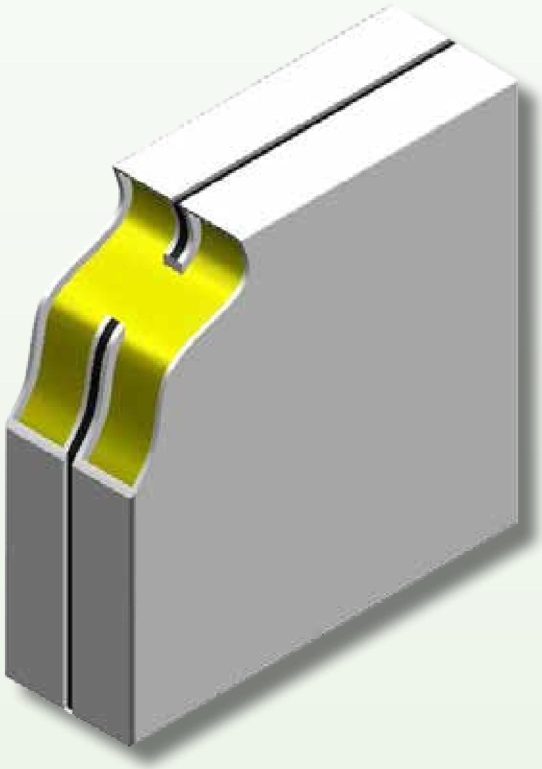


Energy $avings
with AAON
Rigid Polyurethane Foam Cabinet Construction

Annual Savings from AAON Rigid Polyurethane Foam Cabinet

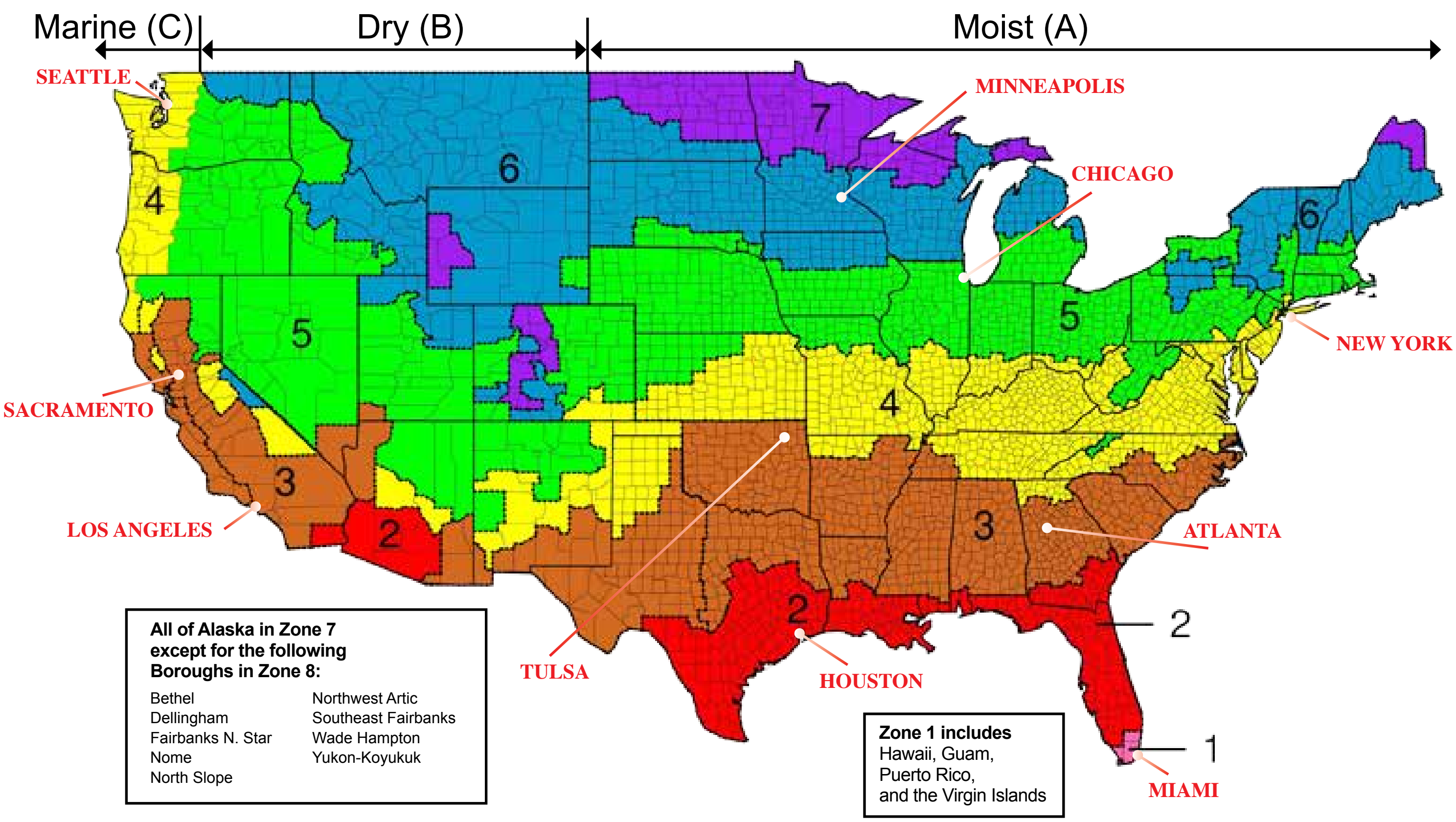


Figure 1: ASHRAE Climate Zones

	Nominal Tons								Annual Savings
	5	10	15	30	70	120	170	210	
Atlanta	$254	$473	$691	$1,308	$2,970	$5,122	$7,210	$8,900	19%
Chicago	$243	$441	$638	$1,179	$2,645	$4,640	$6,513	$8,038	19%
Houston	$286	$549	$812	$1,575	$3,621	$6,261	$8,838	$10,914	17%
Los Angeles	$47	$91	$135	$265	$611	$1,053	$1,488	$1,837	5%
Miami	$405	$776	$1,147	$2,222	$5,108	$8,835	$12,472	$15,401	22%
Minneapolis	$261	$472	$681	$1,257	$2,817	$4,945	$6,939	$8,563	19%
New York	$225	$409	$593	$1,102	$2,477	$4,340	$6,095	$7,522	19%
Sacramento	$175	$313	$452	$824	$1,836	$3,233	$4,531	$5,590	13%
Seattle	$172	$307	$442	$802	$1,783	$3,145	$4,405	$5,435	20%
Tulsa	$286	$528	$770	$1,449	$3,283	$5,727	$8,056	$9,944	16%

Table 1: Estimated Savings From AAON Rigid Polyurethane Foam Cabinet

EXECUTIVE SUMMARY

Lost energy is lost money. As utility costs increase, poor insulation and poor air seals will result in more wasted heating and cooling dollars. Architects, engineers, and owners know that utilizing quality materials and construction techniques will minimize losses in the building. For instance, the ASHRAE standard 90.1-2010 requires all above deck insulation in climate zones 2 through 8 to have a minimum thermal insulating value of R-20 and climate zone 1 to have a minimum R-13. Likewise, ASHRAE 90.1-2010 requires sealing, caulking, gasketing, or weather-stripping of various areas of the building envelope. Like the building, the Heating Ventilating and Air Conditioning (HVAC) cabinet can lose energy through poor insulation and poor air seals. Like the building, the HVAC cabinet should have requirements for minimum insulation and air seals. **An HVAC cabinet can become a pathway for heating and cooling energy to enter or escape from a building.**

Conventional HVAC cabinets incorporate narrow one-half to one inch fiberglass insulation attached to a single wall of galvanized sheet metal. Thermal resistance values for fiberglass panels are typically less than R-2. **The R value of a fiberglass panel is reduced even more without a thermal break between the HVAC tunnel air temperature and the outside ambient air temperature.** Due to the lack of published standards on the HVAC cabinet air leakage, air infiltration percentages are not available. Although air infiltration rates may vary between manufacturers, 4% of total air flow at 1 inch of internal negative static water pressure can be used as a representative leakage rate.

AAON uses a different approach. **AAON manufactures a double wall Rigid Polyurethane Foam panel with thermal breaks for the HVAC cabinet's air tunnel.** By implementing a composite structure, the foam and sheet metal are combined into a cabinet with improved insulation and air seals. The AAON two inch wall panel has a thermal resistance of R-13. This is equivalent to using panels with thermal breaks and four inch thick, 0.5 pound per square foot fiberglass. This would commonly be found on more costly custom equipment. All AAON panels incorporate thermal breaks to separate the HVAC tunnel air temperature with the outside ambient air temperature. Cabinet testing has shown the air infiltration rate is below 1.5% of total air flow at 1 inch of internal negative static water pressure.

The increased insulation of an AAON cabinet will reduce the undesired heating and cooling of the cabinet airstream due to the outside ambient temperature. By improving air tightness, the AAON cabinet reduces any load created from the outside air. **One benefit of an AAON cabinet is a reduction in owner's operating cost up to 22% each year.**

The utilization of the AAON foam panel design has improved:

- Operating Costs
- Thermal Resistance and Breaks
- Air Seals
- Rigidity
- Impact Resistance
- Maintainability
- Indoor Air Quality
- Equipment Lifetime

The conventional fiberglass insulated cabinet creates a thermal bridge for heating and cooling energy between a building's interior and the outside environment. The AAON Rigid Polyurethane Foam cabinet saves cooling and heating energy through improved insulation and air seals. This reduces the energy lost to the environment and reduces the building owner's operating cost. *Saved energy is saved money.*

Polyurethane Foam Core Panel with Thermal Break

RIGID POLYURETHANE FOAM PANEL DESIGN

The manufacture of AAON Rigid Polyurethane Foam implements cutting edge technology. Low viscosity polyurethane liquid is injected into a galvanized steel casing with integrated thermal breaks. As the liquid foam fills the casing, trapped air escapes through designed openings. After the casing is evacuated of all air, the polyurethane liquid is cured through a pressurized baking process. The resulting closed cell foam core has a density of two pounds per cubic foot. This composite design results in improved operating costs, thermal resistance and breaks, air seals, rigidity, impact resistance, maintainability, indoor air quality, and equipment lifetime.

THERMAL RESISTANCE

The one-half to one inch thick fiberglass panels used in conventional HVAC cabinets will have a thermal resistance of R-0.5 to R-3.5. AAON Rigid Foam Panels utilize two inch thick walls and a two and one half inch thick roof. The foam panels have a thermal resistance of R-13 and R-16 respectively. This is equivalent to four inches of fiberglass commonly found in more costly custom HVAC cabinets.

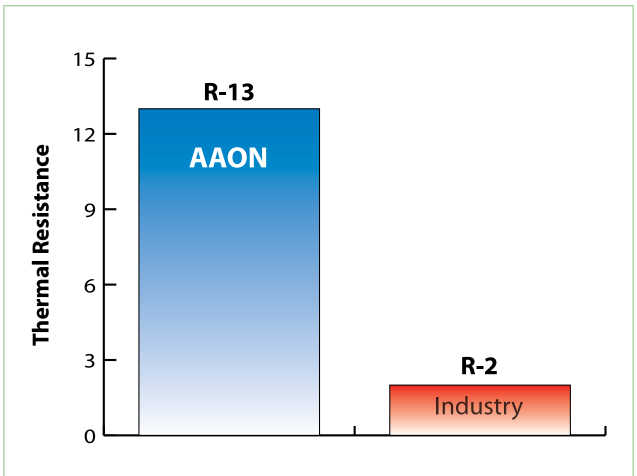


Figure 2: Two Inch Double Wall Rigid Polyurethane Foam Insulation Compared to One Inch Fiberglass Insulation

THERMAL BREAK

Conventional fiberglass cabinets have sheet metal panels that have no thermal break, creating a direct pathway from the cabinet's interior to the atmosphere. The published R-value of fiberglass insulated cabinets is much less because of this thermal bridge.

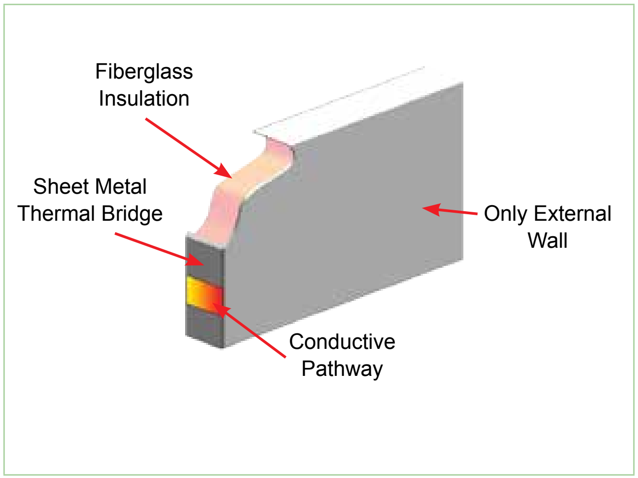


Figure 3: Conventional Fiberglass Panels

AAON incorporates a thermal break in all its foam panels. This break disconnects the conductive path between the cabinet interior and the atmosphere, meaning R-13 is the actual thermal resistance of the two inch panels.

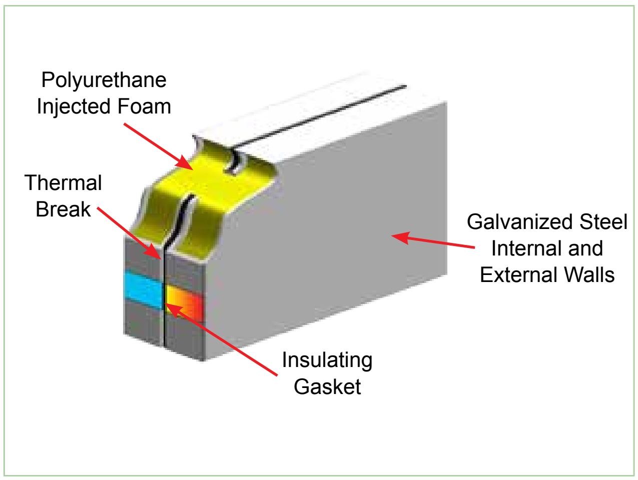


Figure 4: AAON Rigid Polyurethane Foam Panels

AIR SEALS

Fiberglass cabinets have no seal, or the seal is cut in the corners. Without a seal, the tolerance difference between the opening and the door will lead to increased infiltration. Likewise, seals joined at the corners allow ambient air into the airstream through this gap. One reason for the low infiltration rate in the AAON cabinet is the continuous door sealing. On doorways and openings into the air stream, AAON uses a continuous seal to prevent leakage at corner joints. **The continuous seals reduce the infiltration rate on AAON cabinets.**

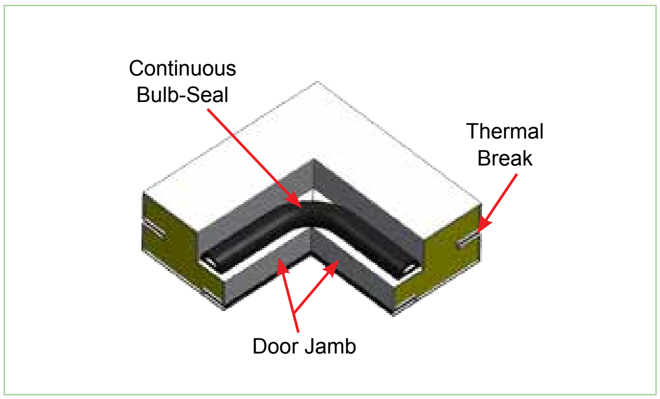


Figure 5: AAON Rigid Polyurethane Foam Panels

RIGIDITY

The laminate construction of the polyurethane foam and galvanized steel increases the panel rigidity. This construction yields a light weight panel, while still providing an exceptional L/240 deflection ratio at eight inches of static pressure. For example, a panel five feet long subjected to eight inches of static pressure would have a maximum deflection of one-quarter of an inch. AAON standard design exceeds custom HVAC cabinets that have panels designed to L/200 deflection. **The increased rigidity is another reason for the low infiltration rate on AAON cabinets.**

Other HVAC cabinets may claim heavier gauge sheet metal makes a panel rigid. This statement is misleading; the rigidity of an object is not just based on material thickness. Rigidity is also affected by the geometry of the panel and the method of cabinet assembly.

IMPACT RESISTANCE

An AAON Rigid Polyurethane Foam cabinet is remarkably impact resistant. The addition of the foam increases the impact resistance of the panels. In a recent product test, the AAON cabinet was able to withstand two direct hits from an 8 foot long 2” x 4” traveling at over 35 miles per hour. Figure 6 shows no reduction of cabinet integrity after two strikes in the same place.

Figure 6: AAON Rigid Polyurethane Foam Panel Impact Resistance

MAINTAINABILITY

One other advantage to the AAON Rigid Polyurethane Foam panel is the internal protective barrier. The conventional fiberglass panel is exposed to condensation and particulates in the air stream. **By using a galvanized internal barrier with a thermal break, the AAON panel is washable and resists moisture formation.** In climates where air streams may contain salt, the ability to clean the salty condensate will prevent corrosive damage.

INDOOR AIR QUALITY

Two components in the Rigid Polyurethane Foam panel inhibit microbial growth. The thermal break in the panel will inhibit moisture formation on the internal barrier surface, while the closed cell foam will resist moisture absorption and microbial growth. **Compared to conventional fiberglass cabinets, the AAON cabinet promotes a higher indoor air quality.**

EQUIPMENT LIFETIME

Another important factor in the HVAC cabinet design is the heating and cooling savings over the entire equipment lifetime. As a fiberglass cabinet ages, the fiberglass insulation peels away and sags off of the sheet metal structure. This will decrease the fiberglass cabinet thermal resistance. The AAON HVAC cabinet is designed to maintain a consistent thermal resistance. **Over the equipment lifetime, the AAON cabinet would increase its savings compared to the fiberglass cabinet**.

OPERATING COST

The following examples quantify the savings a building owner can experience with an AAON Rigid Polyurethane Foam cabinet. A simplified 1 foot cube will be used to compare the additional cooling load between a fiberglass insulation design and an AAON design. After the simplified cube is explained, the same calculations are performed on typical HVAC cabinet sizes. Then additional cooling infiltration loads between the fiberglass insulated cabinet and the AAON cabinet will be compared. Similarly, the additional heating load through insulation and infiltration will be compared. **At the end, various cities with different climates will be presented with the cooling, heating, and total savings.**

CALCULATION CONSIDERATIONS

In a laboratory setting, a test was performed comparing a double wall, 1 inch thick fiberglass cabinet to a Rigid Polyurethane Foam AAON cabinet. When internal negative static pressure was equal to 1 inch water gauge, the fiberglass design allowed 4% ambient air flow to enter the cabinet. The fiberglass unit incorporated seals for all access panels, grommets for copper tubing (i.e. condenser coil or compressor compartment interface with air tunnel), and gaskets in the electrical access points. If a cabinet did not incorporate these design features, the expected infiltration percentage could easily be 6% or more. **The AAON Rigid Polyurethane Foam cabinet allowed only 1.5% air flow to enter the cabinet at the same 1 inch internal negative static water pressure.**

Testing of the two designs indicates the percentage infiltration between the designs increases disproportionately as internal static pressure increases. Figure 7 shows a 63% difference between the fiberglass design and the AAON design at 1 inch internal static pressure, while the difference at 3 inch internal static is 70%. This study shows that heating and cooling load calculations should use infiltration rates of 4% or more for the fiberglass design and 1.5% for the AAON Rigid Polyurethane Foam design.

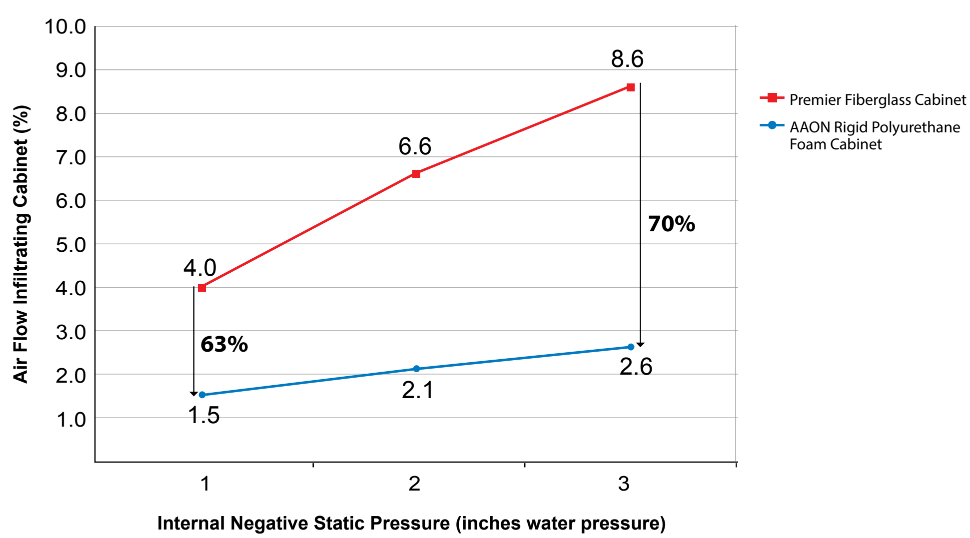


Figure 7: Cabinet Infiltration Testing - Fiberglass Cabinet Versus AAON Rigid Polyurethane Foam Cabinet

INCREASED COOLING LOAD (INSULATION)

If the outside ambient is warmer than the HVAC tunnel air stream, conductive heat transfer will occur through the HVAC cabinet and increase the cooling load. The conductive energy entering the HVAC cabinet is proportional to the thermal resistance of the panels.

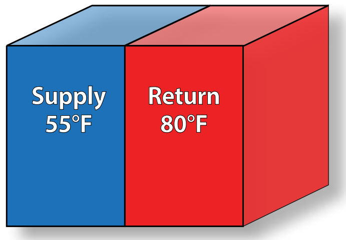


Figure 8: HVAC Cooling Model

$$Q = \frac{1}{R} * A * \Delta T$$

$$Q = Conductive\ Heat\ Transfer\ [Btu/hr]$$

$$R = Thermal\ Resistance\ (R_{value}) \left[\frac{ft^2 * {}^{o}F * hr}{Btu}\right]$$

$$A = Surface\ Area\ [ft^2]$$

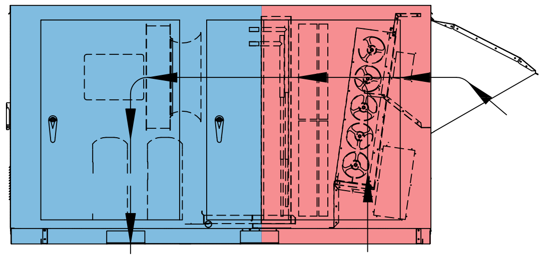


Figure 9: Example of HVAC Cabinet Cooling Model

$$\Delta T = Outside\ Dry\ Bulb\ Temperature\ [{}^{o}F] - Air\ Stream\ Temperature\ [{}^{o}F]$$

In this example, the R value for a one-half to one inch thick, 0.5 pound per square foot fiberglass cabinet is 2. For the initial calculations, a one foot cube will have a 55ºF supply air and an 80ºF return air equally distributed in the air tunnel. The supply side roof and return side roof surface areas' are each 0.5 ft^2.

$$Half\ Roof\ Surface\ Area\ [ft^2] = \frac{Roof\ Length\ [ft]}{2} * Roof\ Width\ [ft]$$

$$Half\ Roof\ Surface\ Area\ [ft^2] = \mathbf{0.5ft^2} = \frac{1\,ft}{2} * 1ft$$

Dry Bulb Temperature (°F)	Time At Condition (hours)
97	9
92	32
87	220
82	758
77	768
72	1,037
67	1,162

Table 2: Atlanta Cooling Bin Data

The initial outside ambient temperatures will be for Atlanta, Georgia and will come from the ASHRAE "Bin and Degree Hour Weather Data for Simplified Energy Calculations." Table 2 shows the dry bulb temperatures from 67ºF to 97ºF. These temperatures and the hours at each condition will be used to total the year's increased cooling load.

Solar radiation can also increase the surface temperature of the HVAC cabinet. The solar radiation calculations have been excluded from these conductive heat transfer calculations since daylight, solar altitude, cloud index, shading, etc. require detailed site information for proper modeling.

By inputting the supply side temperature difference, roof surface area, and roof R-value into the conductive heat transfer equation, the increased cooling load imposed through the supply side roof is found.

$$Supply\ Roof\ Increased\ Cooling\ Load\ (Insulation)[Btu/hr] = \frac{(Outside\ Dry\ Bulb\ Temperature\ [°F] - Supply\ Air\ Temperature[°F]) * Supply\ Roof\ Surface\ Area\ [ft^2]}{R_{value}[(ft^2 * °F * hr)/Btu]}$$

$$Supply\ Roof\ Increased\ Cooling\ Load\ (Insulation)\ [Btu/hr] = \mathbf{10.50\ Btu/hr} = \frac{(97°F - 55°F) * 0.5ft^2}{2ft^2 * °F * hr/Btu}$$

Similarly, the extra cooling load through the return side roof is found by changing the air stream temperature.

$$Return\ Roof\ Increased\ Cooling\ Load\ (Insulation)\ [Btu/hr] = \frac{(Outside\ Dry\ Bulb\ Temperature\ [°F] - Return\ Air\ Temperature[°F]) * Return\ Roof\ Surface\ Area\ [ft^2]}{R_{value}[(ft^2 * °F * hr)/Btu]}$$

$$Return\ Roof\ Increased\ Cooling\ Load\ (Insulation)\ [Btu/hr] = \mathbf{4.25\ Btu/hr} = \frac{(97°F - 80°F) * 0.5ft^2}{2ft^2 * °F * hr/Btu}$$

The fiberglass panels will allow heating of the air stream 10.50 Btu/hr through the supply roof and 4.25 Btu/hr through the return roof. By repeating the process for the entire cube, 73.75 Btu/hr of increased cooling is required when the outside dry bulb temperature is 97ºF. Similarly, the process can be repeated for a cube using AAON Rigid Polyurethane Foam design of R-13 walls and R-16 roofs. The entire AAON cube would only require 10.92 Btu/hr of additional cooling. Table 3 gives the detailed values for the entire cube. By multiplying the whole cube's conductive heat transfer by 9 hours, the year's increased cooling load at the 97ºF condition is 664 Btu for the fiberglass insulated cube. The AAON cube only requires an additional 98 Btu. **The AAON design has decreased the extra cooling load by 85%.**

	Fiberglass		AAON		
	R Value	Additional Cooling Load (Btu/hr)	R Value	Additional Cooling Load (Btu/hr)	
Supply Roof	2	10.50	16	1.31	
Supply Walls	2	42.00	13	6.46	
Return Roof	2	4.25	16	0.53	
Return Walls	2	17.00	13	2.62	
1 ft Cube Insulation Increased Cooling ***(Btu/hr)***	73.75		10.92		**AAON SAVINGS**
9 Hours at 97°F (Atlanta Bin Data)	**664**		**98**		**85%**

Table 3: Atlanta 1 Foot Cube 97°F Condition Increased Cooling Load

Considering all cooling conditions between 67ºF and 97ºF, the increased cooling load required for the fiberglass and the AAON design is shown in Table 4. At 67ºF the additional cooling required from the heat transfer into the supply section is less than the heat transfer out of the return section to the outside ambient. At this condition, the net thermal transfer is negative.

Dry Bulb Temperature	Time At Condition	Fiberglass Increased Cooling		AAON Increased Cooling	
°F	*hrs*	*Btu/hr*	*Btu*	*Btu/hr*	*Btu*
97	9	73.75	664	10.92	98
92	32	61.25	1,960	9.07	290
87	220	48.75	10,725	7.22	1,588
82	758	36.25	27,478	5.37	4,069
77	768	23.75	18,240	3.52	2,701
72	1,037	11.25	11,666	1.67	1,728
67	1,162	-1.25	-1,453	-0.19	-215
100% Occupied Increased Cooling Load (Btu)			**69,280**		**10,259**
50% Occupied Increased Cooling Load (Btu)			**34,640**		**5,129**

Table 4: Atlanta 1 Foot Cube Total Annual Increased Cooling Load

As Table 4 shows, the difference between the fiberglass design and the AAON Rigid Polyurethane Foam design is 59,021 Btu at 100% occupancy and 29,511 Btu at 50% occupancy. **Compared to the fiberglass insulation design, the AAON Rigid Polyurethane Foam design decreases superfluous cooling loads by 85%.**

Figure 10 reinforces the results from Table 4. The increased cooling load through the fiberglass insulation design is considerably higher than the AAON Rigid Polyurethane Foam design.

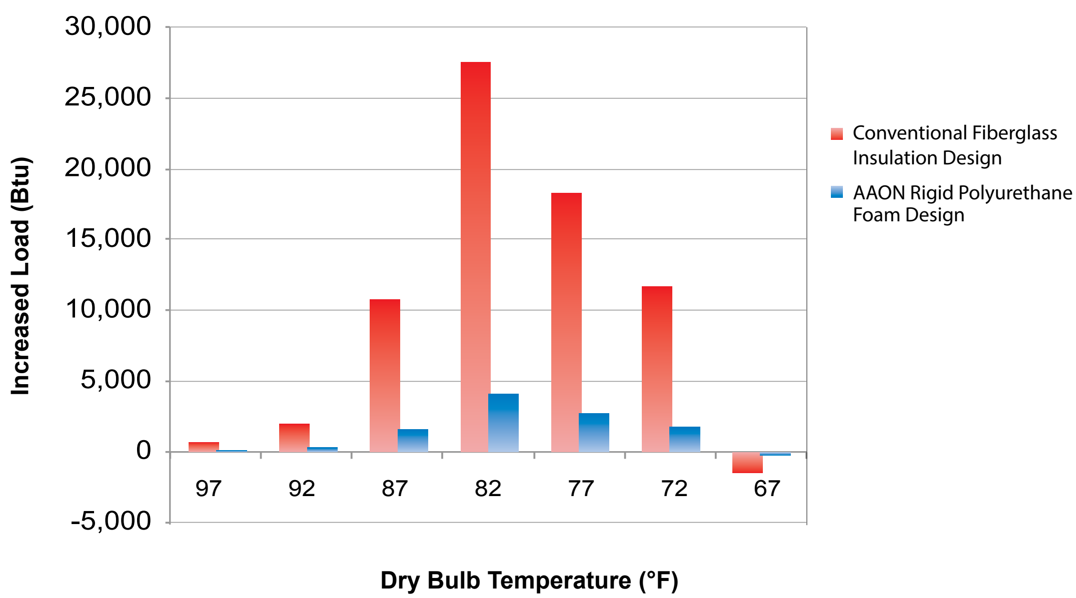


Figure 10: Atlanta 1 Foot Cube Increased Cooling Load Through Insulation Based on Bin Hours of Occurrence

Now that the calculations with a model cube are complete, the same calculations are performed with the approximate HVAC cabinet sizes and tons as indicated in Table 5.

		Approximate Tons							
		5	10	15	30	70	120	170	210
Cabinet Dimensions	Approximate Flowrate (ft^3/min)	2,000	4,000	6,000	12,000	28,000	48,000	68,000	84,000
	Length (in)	82	82	88	110	155	294	506	533
	Width (in)	44	79	96	101	100	142	100	142
	Height (in)	43	44	50	59	97	102	102	102
	Approximate Surface Area (ft^2)	100	143	186	250	451	908	1,210	1,482

Table 5: Approximate HVAC Cabinet Dimensions

Figure 11 gives a good indication of the increased cooling load based on the various cabinet sizes. **At every cabinet size, the AAON design saves cooling energy.**

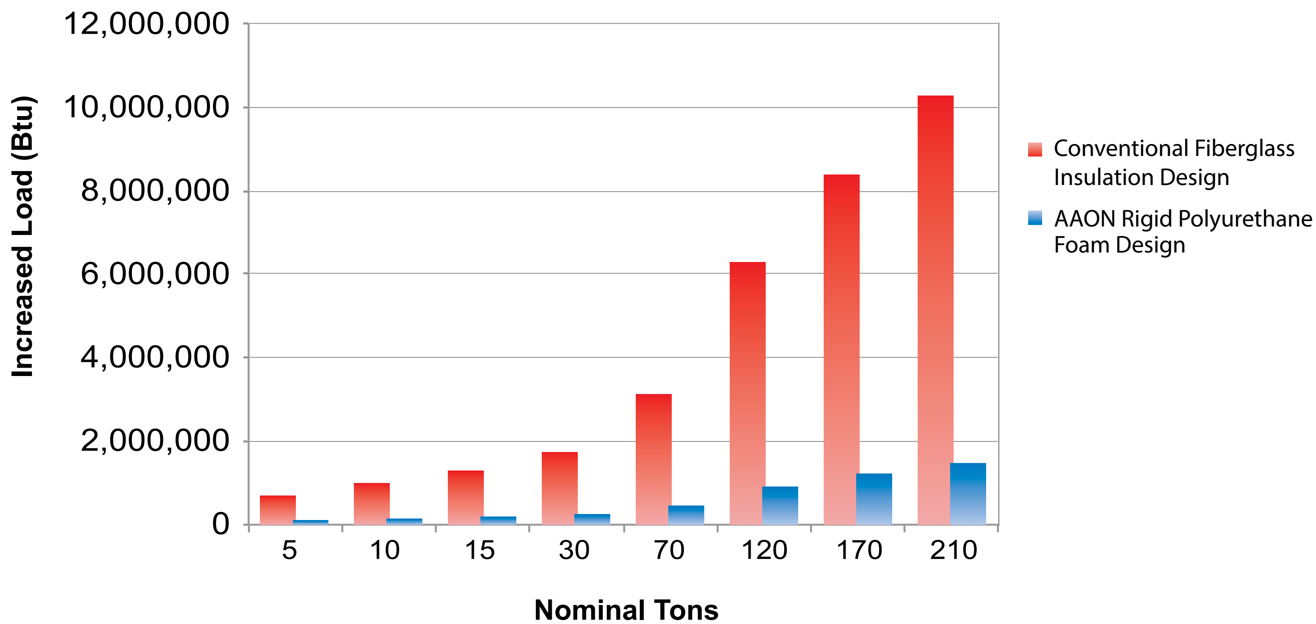


Figure 11: Annual Atlanta Cabinet Increased Cooling Load Through Insulation

INCREASED COOLING LOAD (INFILTRATION)

At 1 inch negative internal static water pressure, 4% infiltration enters the premier fiberglass cabinets. This means that 4% of the air flow can enter the HVAC cabinet and create an additional load. The additional 4% air flow is significant because of the sensible and latent load added to the system. For instance, a 10 ton unit with an estimated 4,000 cubic feet per minute (cfm) would have 160 cfm of infiltration. Using Atlanta, Georgia bin data, 97ºF (Dry Bulb) and 75.67ºF (Wet Bulb), the air density is 0.0696 lb/ft³ with an enthalpy of 39.77 Btu/lb. The supply air enthalpy will be 23.22 Btu/lb at 55ºF.

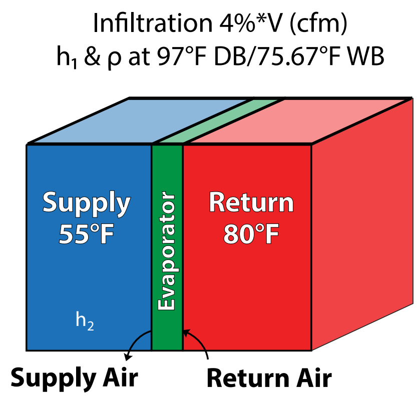


Figure 12: Cooling infiltration Model

$$Increased\ Cooling\ Load\ (Infiltration)[Btu/hr] = \dot{m}\,[lb/hr] * \Delta h[Btu/lb]$$
$$= V[ft^3/min] * 60[min/hr] * \rho[lb/ft^3] * (h_2[Btu/lb] - h_1[Btu/lb])$$

$$Increased\ Cooling\ Load\ (Infiltration)\ [Btu/hr] = \mathbf{11,058\ Btu/hr}$$
$$= 160\,ft^3/min * 60\,min/hr * 0.0696\,lb/ft^3 * (39.77\,Btu/lb - 23.22\,Btu/lb)$$

As the calculation shows, the additional cooling required due to infiltration is 11,058 Btu/hr. At the 97ºF conditions, the 10 ton unit can have an additional 0.92 tons of cooling required due to infiltration. The Atlanta bin data shows 9 hours of this condition each year. During these 9 hours, the infiltration causes 8.3 ton-hr of lost energy. These 9 hours would require an additional 99,522 Btu of cooling. Table 6 shows the losses for dry bulb temperatures 67 ºF and higher.

Dry Bulb Temperature	Mean Coincident Wet Bulb Temperature	Air Density	Increased Cooling At Condition	Time At Condition	10 Ton Fiberglass Annual Increased Cooling At Condition	
ºF	ºF	lb/ft³	tons	hours	ton•hr	Btu
97	75.67	0.0696	0.92	9	8.3	99,522
92	74.69	0.0702	0.88	32	28.2	337,930
87	72.84	0.0708	0.79	220	173.6	2,082,947
82	69.93	0.0716	0.65	758	492.8	5,913,564
77	67.88	0.0723	0.56	768	430.4	5,165,288
72	66.30	0.0730	0.49	1,037	511.7	6,140,865
67	62.50	0.0739	0.33	1,162	381.3	4,575,250
10 Ton Total Annual Increased Cooling Load					**2,026.3**	**24,315,367**

Table 6: 10 Ton Fiberglass Cabinet Annual Increased Cooling Load From Infiltration

Each year 24,315,367 Btu of additional cooling is required because of the fiberglass cabinet's infiltration rate. The fiberglass cabinet requires an additional 2,026 ton-hr of cooling each year. By conservatively estimating that the building is only occupied for 50% of those temperatures gives an additional cooling requirement of 12,157,683 Btu, or 1,013 ton•hr, each year.

Now, consider the additional cooling required when a cabinet only has 1.5% of air flow infiltrating. At 50% occupancy, the AAON cabinet would only require an additional 4,559,131 Btu, or 380 ton-hr, each year. **The low infiltration rate from an AAON Rigid Polyurethane Foam cabinet reduces this superfluous cooling load by 63%.**

TOTAL INCREASED COOLING LOAD

The insulation losses and infiltration losses for temperatures at or above 67°F can be summed to yield each year's increased cooling load. To compare the increased cooling loads for each cabinet size, see Figure 13.

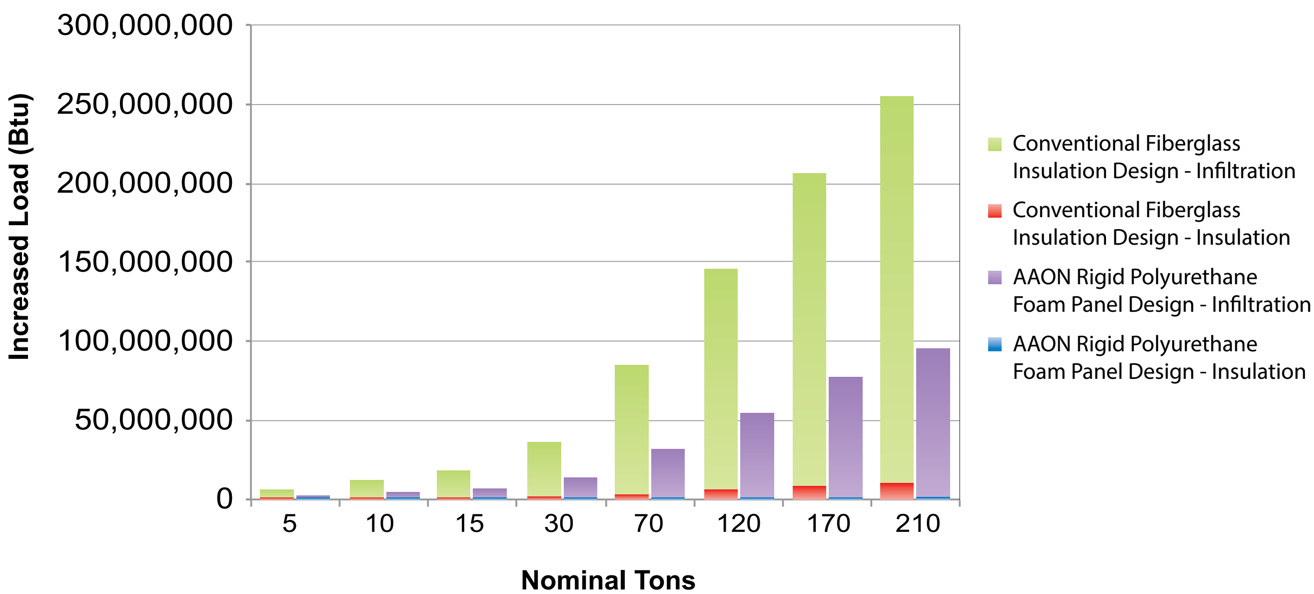


Figure 13: Annual Atlanta Cabinet Increased Cooling Load Through Insulation and Infiltration

The lower resistance insulation combined with the high infiltration rate can dramatically increase the cooling load on an HVAC unit. For instance, operating at 10 ton fiberglass HVAC cabinet at 50% occupancy could require an additional 13,150,985 Btu of cooling each year in Atlanta, Georgia. Table 7 provides a sample calculation for annual increased cooling load of a building with twenty 10 ton units.

	Annual Increased Cooling Fiberglass		Annual Increased Cooling AAON	
	ton•hr	*Btu*	*ton•hr*	*Btu*
50% Occupancy Insulation Losses	83	993,302	12	143,825
50% Occupancy Infiltration Losses	1,013	12,157,683	380	4,559,131
Total Increased Cooling Load	1,096	13,150,985	392	4,702,956
Twenty 10 Ton Units' Increased Load	**21,920**	**263,019,700**	**7,840**	**94,059,120**
Increased Annual Usage (kWh)	**77,084**		**27,566**	
Increased Annual Cooling Cost ($0.08/kWh)	**$6,167**		**$2,205**	
Increased Annual Cooling Cost ($0.10/kWh)	**$7,708**		**$2,757**	
Increased Annual Cooling Cost ($0.12/kWh)	**$9,250**		**$3,308**	

Table 7: Increased Cooling Cost Of An Example Office Building With Twenty 10 Ton Units

A twenty unit office building using 10 ton fiberglass cabinets could have 263,019,700 Btu of increased cooling due to the fiberglass insulated cabinet's lower resistance insulation and higher infiltration rate. At $0.08 a kWh, a building owner with these fiberglass HVAC cabinets is wasting about $6,166 each year. An equal size AAON cabinet would require about $2,205 each year for increased cooling loads. If utility costs are $0.12 a kWh, the fiberglass cabinet uses an additional $9,250 for cooling, while the AAON cabinet only uses an additional $3,308. **A building owner with the AAON designed HVAC cabinet could be saving $5,942 each year in cooling costs.**

INCREASED HEATING LOAD (INSULATION)

If the outside ambient is cooler than the HVAC tunnel air stream, conductive heat transfer will occur through the HVAC cabinet and increase the heating load. The conductive energy leaving the HVAC cabinet is proportional to the thermal resistance of the panels.

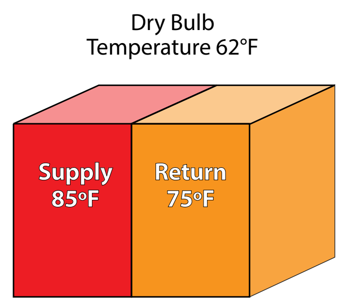


Figure 14: HVAC Heating Model

$$Q = \frac{1}{R} * A * \Delta T$$

$$Q = Conductive\ Heat\ Transfer\ [Btu/hr]$$

$$R = Thermal\ Resistance\ (R_{value}) \left[\frac{ft^2 * {}^{\circ}F * hr}{Btu}\right]$$

$$A = Surface\ Area\ [ft^2]$$

$$\Delta T = Air\ Stream\ Temperature\ [{}^{\circ}F] - Outside\ Dry\ Bulb\ Temperature\ [{}^{\circ}F]$$

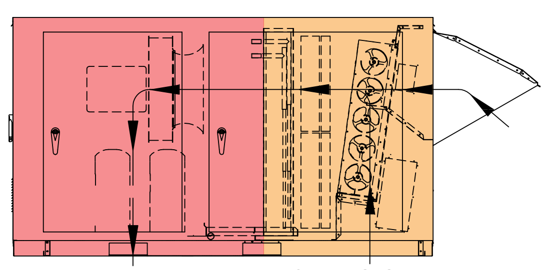


Figure 15: Example of HVAC Cabinet Heating Model

Dry Bulb Temperature (°F)	Time At Condition (hours)
62	1,027
57	790
52	673
47	494
42	583
37	560
32	323
27	156
22	97
17	64
12	7

Table 8: Atlanta Heating Bin Data

With one-half to one inch thick, 0.5 pound per square foot insulation, the fiberglass cube's roof has an R value of 2. The one foot cube will have 85ºF supply air and 75ºF return air equally distributed in the air tunnel. The supply side roof and return side roof surface areas' are 0.5 ft^2.

The outside ambient temperatures for Atlanta, Georgia will be taken from the ASHRAE 'Bin and Degree Hour Weather Data for Simplified Energy Calculations'. Table 8 shows the dry bulb temperatures from 62ºF to 12ºF. The temperature and hours at each condition will be used to determine additional heating loads for the year.

$$Supply\ Roof\ Increased\ Heating\ Load\ (Insulation)\ [Btu/hr] = \frac{(Supply\ Air\ Temperature\ [°F] - Outside\ Dry\ Bulb\ Temperature[°F]) * Supply\ Roof\ Surface\ Area\ [ft^2]}{R_{value}[(ft^2 * °F * hr)/Btu]}$$

$$Supply\ Roof\ Increased\ Heating\ Load\ (Insulation)\ [Btu/hr] = \mathbf{5.75\ Btu/hr} = \frac{(85°F - 62°F) * 0.5ft^2}{2ft^2 * °F * hr/Btu}$$

Similarly, the additional heating load through the return side roof is found by changing the air stream temperature.

$$Return\ Roof\ Increased\ Heating\ Load\ (Insulation)\ [Btu/hr] = \frac{(Return\ Air\ Temperature\ [°F] - Outside\ Dry\ Bulb\ Temperature[°F]) * Return\ Roof\ Surface\ Area\ [ft^2]}{R_{value}[(ft^2 * °F * hr)/Btu]}$$

$$Return\ Roof\ Increased\ Heating\ Load\ (Insulation)\ [Btu/hr] = \mathbf{3.25\ Btu/hr} = \frac{(75°F - 62°F) * 0.5ft^2}{2ft^2 * °F * hr/Btu}$$

The fiberglass panels will allow 5.75 Btu/hr of heat through the supply side roof to the outside ambient. It also allows 3.25 Btu/hr of heat through the return side roof. The entire fiberglass cube would allow 45.00 Btu/hr of heat to transfer out of the cube. With the AAON Rigid Polyurethane Foam design, the entire cube would only allow 6.67 Btu/hr of heat transfer to the ambient air.

Table 9 has detailed values for the fiberglass and AAON cube. For the 1,027 hours, the fiberglass cube is at 62ºF and requires an extra 46,215 Btu of heating. The AAON design only requires an additional 6,843 Btu. **The AAON design has decreased the superfluous heating load by 85%.**

	Fiberglass		AAON		
	R Value	Additional Heating Load (Btu/hr)	R Value	Additional Heating Load (Btu/hr)	
Supply Roof	2	5.75	16	0.72	
Supply Walls	2	23.00	13	3.54	
Return Roof	2	3.25	16	0.41	
Return Walls	2	13.00	13	2.00	
1 ft Cube Insulation Increased Heating *(Btu/hr)*	45.00		6.67		**AAON SAVINGS**
1,027 Hours at 62ºF (Atlanta Bin Data)	**46,215**		**6,843**		**85%**

Table 9: Atlanta 1 Foot Cube 62ºF Condition Increased Heating Load

For all heating conditions between 62ºF and 12ºF, the increased heating load for both designs are shown in Table 10.

Dry Bulb Temperature	Time At Condition	Fiberglass Increased Cooling		AAON Increased Cooling	
ºF	*hours*	*Btu/hr*	*Btu*	*Btu/hr*	*Btu*
62	1,027	45.00	46,215	6.67	6,843
57	790	57.50	45,425	8.51	6,726
52	673	70.00	47,110	10.37	6,976
47	494	82.50	40,755	12.22	6,035
42	583	95.00	55,385	14.07	8,201
37	560	107.50	60,200	15.92	8,914
32	323	120.00	38,760	17.77	5,739
27	156	132.50	20,670	19.62	3,061
22	97	145.00	14,065	21.47	2,083
17	64	157.50	10,080	23.32	1,493
12	7	170.00	1,190	25.17	176
100% Occupied Increased Cooling Load (Btu)			**379,855**		**56,248**
50% Occupied Increased Cooling Load (Btu)			**189,928**		**28,124**

Table 10: Atlanta 1 Foot Cube Total Annual Increased Heating Load

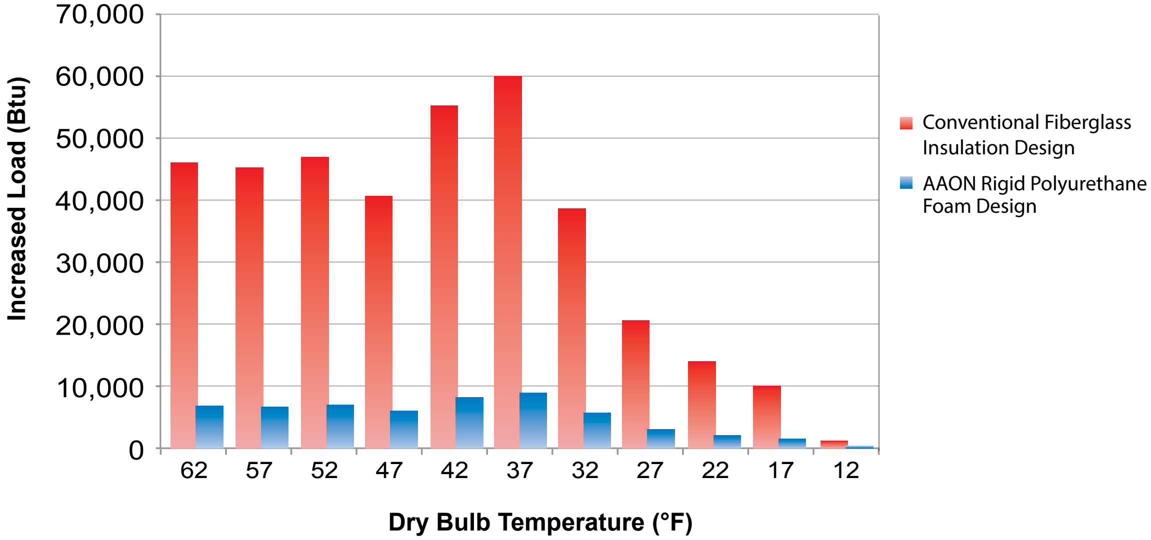


Figure 16: Atlanta 1 Foot Cube Increased Heating Load Through Insulation Based on Bin Hours of Occurrence

The AAON Rigid Polyurethane Foam design would save 323,607 Btu at 100% occupancy and 161,804 Btu at 50% occupancy. **Compared to the fiberglass insulation design, the AAON Rigid Polyurethane Foam design decreases additional heating loads by 85%.** Figure 16 gives a visual representation of the results found in Table 10. The increased heating load through the fiberglass insulation design is considerably higher than the AAON Rigid Polyurethane Foam design.

INCREASED HEATING LOAD (INFILTRATION)

During heating conditions, the infiltration losses occur when cold air enters the HVAC cabinet and absorbs heat from the system. The bin data for Atlanta will be used to find the sensible heat required due to infiltration. A 10 ton fiberglass cabinet, with 4,000 cubic feet per minute air flow, has 4% (160 cfm) of 62ºF outside ambient air infiltrating the cabinet.

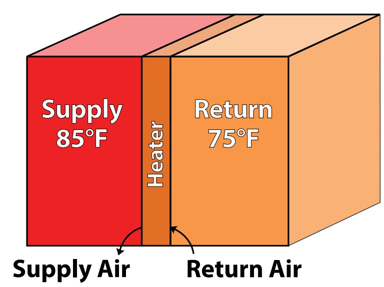


Figure 17: Heating infiltration Model

$$Infiltration\ Heating\ Lost\ [Btu/hr] = \dot{m}\ [lb/hr] * C\rho[Btu/lb * {}^{o}R] * \Delta T[{}^{o}R]$$

$$= V[ft^3/min] * 60[min/hr] * \rho[lb/ft^3] * C\rho[Btu/lb * {}^{o}R]$$

$$* \left((Supply\ Temperature[{}^{o}F] + 460)\,{}^{o}R - (Dry\ Bulb\ Temperature\ [{}^{o}F] + 460)\,{}^{o}R\right)$$

$$= 160\, ft^3/min * 60\, min/hr * 0.0749\, lb/ft^3 * 0.24\, Btu/lb * {}^{o}R * ((85\,{}^{o}F + 460)\,{}^{o}R - (62\,{}^{o}F + 460)\,{}^{o}R)$$

$$= \mathbf{3,969\,\frac{Btu}{hr}}$$

As the calculation shows, infiltrating cold air would require the HVAC system to heat at an additional 3,969 Btu/hr to maintain the 85ºF supply air. The bin data shows that Atlanta will experience 1,027 hours at 62ºF, for at total of 4,076,267 Btu of additional heating. Table 11 shows the losses for dry bulb temperatures 62ºF and below.

If all heating losses at or below 62ºF are summed, the result is 31,604,358 Btu. Estimating a building occupancy of 50% would result in 15,802,179 Btu of extra heating required because of the fiberglass cabinet's 4% infiltration rate.

Dry Bulb Temperature	Mean Coincident Wet Bulb Temperature	Air Density	Time At Condition	10 Ton Fiberglass Annual Increased Heating At Condition
ºF	ºF	lb/ft³	hours	Btu
62	56.39	0.0749	1,027	4,076,267
57	51.66	0.0758	790	3,863,108
52	46.61	0.0767	673	3,924,703
47	42.73	0.0776	494	3,356,253
42	38.66	0.0784	583	4,528,304
37	33.01	0.0794	560	4,917,363
32	29.00	0.0802	323	3,163,263
27	23.79	0.0811	156	1,690,659
22	19.72	0.082	97	1,154,539
17	15.19	0.0829	64	831,239
12	10.29	0.0838	7	98,661
10 Ton Total Annual Increased Heating Load				**31,604,358**

Table 11: 10 Ton Fiberglass Cabinet Annual Increased Heating Load From Infiltration

By comparison, an AAON cabinet with 1.5% infiltration would only require 5,925,817 Btu for a building occupied 50% of the heating year. **An AAON Rigid Polyurethane Foam cabinet has reduced the additional infiltration heating load by 63%.**

TOTAL HEATING LOSSES

Insulation and infiltration losses for temperatures at or below 62ºF can be summed for the total annual increased heating load. Figure 18 indicates the lower resistance value and higher infiltration rates can dramatically increase the heating load on an HVAC system.

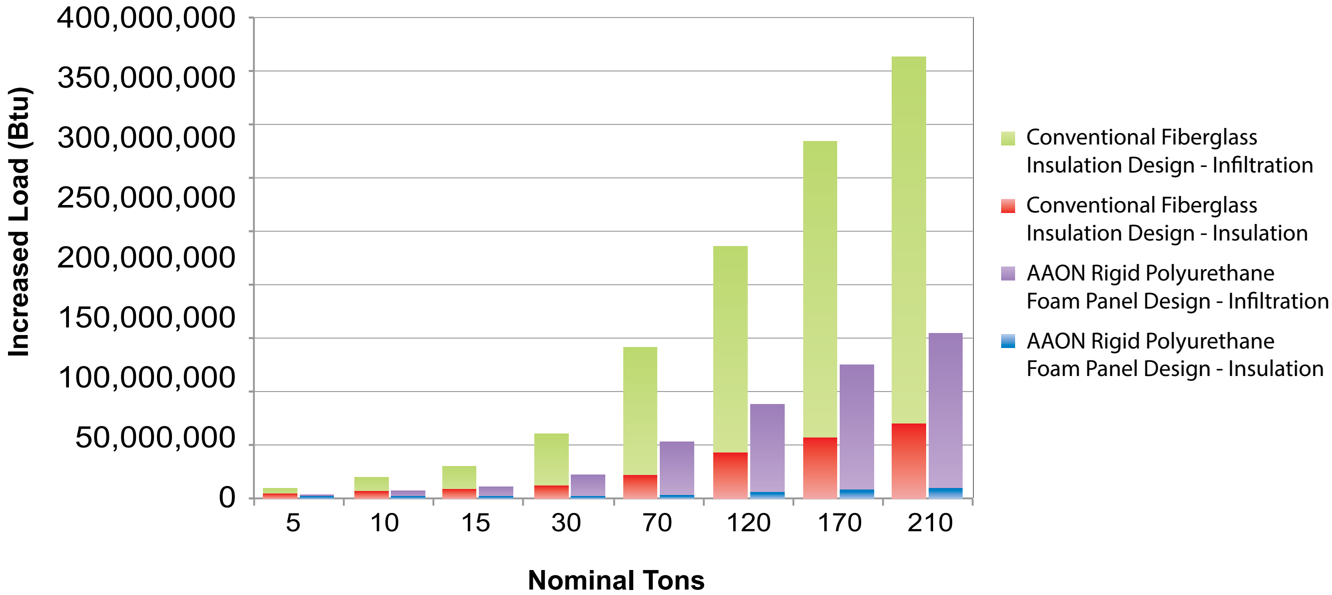


Figure 18: Annual Atlanta Cabinet Increased Heating Load Through Insulation and Infiltration

For instance, at 50% occupancy, a fiberglass 10 ton unit would require an additional 21,248,350 Btu each year. Table 12 provides an example of annual increased heating cost for a building with twenty 10 ton units.

	Annual Increased Heating Fiberglass	Annual Increased Heating AAON
	Btu	*Btu*
50% Occupancy Insulation Losses	5,446,171	788,580
50% Occupancy Infiltration Losses	15,802,179	5,925,817
Total Increased Heating Load	21,248,350	6,714,397
Twenty 10 Ton Units' Increased Load	**424,967,000**	**134,287,940**
Increased Annual Usage (Therm)	4,250	1,343
Increased Annual Heating Cost ($0.80/Therm)	**$3,400**	**$1,074**
Increased Annual Heating Cost ($1.00/Therm)	**$4,250**	**$1,343**
Increased Annual Heating Cost ($1.20/Therm)	**$5,100**	**$1,611**

Table 12: Increased Heating Cost Of An Example Office Building With Twenty 10 Ton Units

A twenty unit office building using 10 ton fiberglass cabinets could have 424,967,000 Btu of heat wasted through the cabinet structure each year. At $0.80 a therm, the fiberglass cabinet uses an additional $3,400 in heating costs each year. The same building, with the AAON Rigid Polyurethane Foam cabinets, would only require an additional 134,287,940 Btu, or $1,074. If utility costs were $1.20 a therm, the fiberglass cabinet uses an additional $5,100 of additional heating, while the AAON cabinet only uses an additional $1,611. **A building owner with the AAON designed HVAC cabinet could save $3,489 each year in heating costs.**

REDUCED OPERATING COST

Now that insulation and infiltration losses for both cooling and heating have been calculated, Figure 19 shows how the AAON Rigid Polyurethane Foam design can save an Atlanta, Georgia building owner between $254 and $8,900 per unit each year with $0.12/kWh and $1.20/therm utility rates.

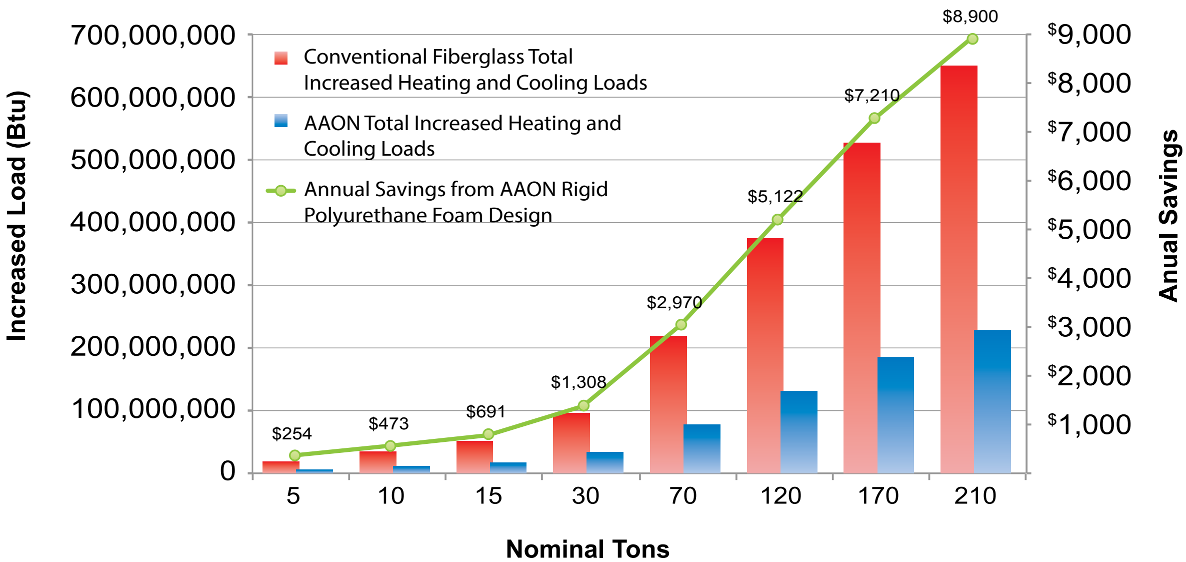


Figure 19: Annual Atlanta Total Cooling and Heating Increased Load With AAON Savings

Savings are calculated for cities in various ASHRAE climate zones shown in Figure 20. When calculating the savings, each city's bin data is based on the ASHRAE "Bin and Degree Hour Weather Data for Simplified Energy Calculations." The utility rates used for each city is $0.12/kWh and $1.20/therm. The cooling and heating savings for Atlanta, Chicago, Houston, Los Angeles, Miami, Minneapolis, New York, Sacramento, Seattle, and Tulsa are shown in Table 13 and Table 14.

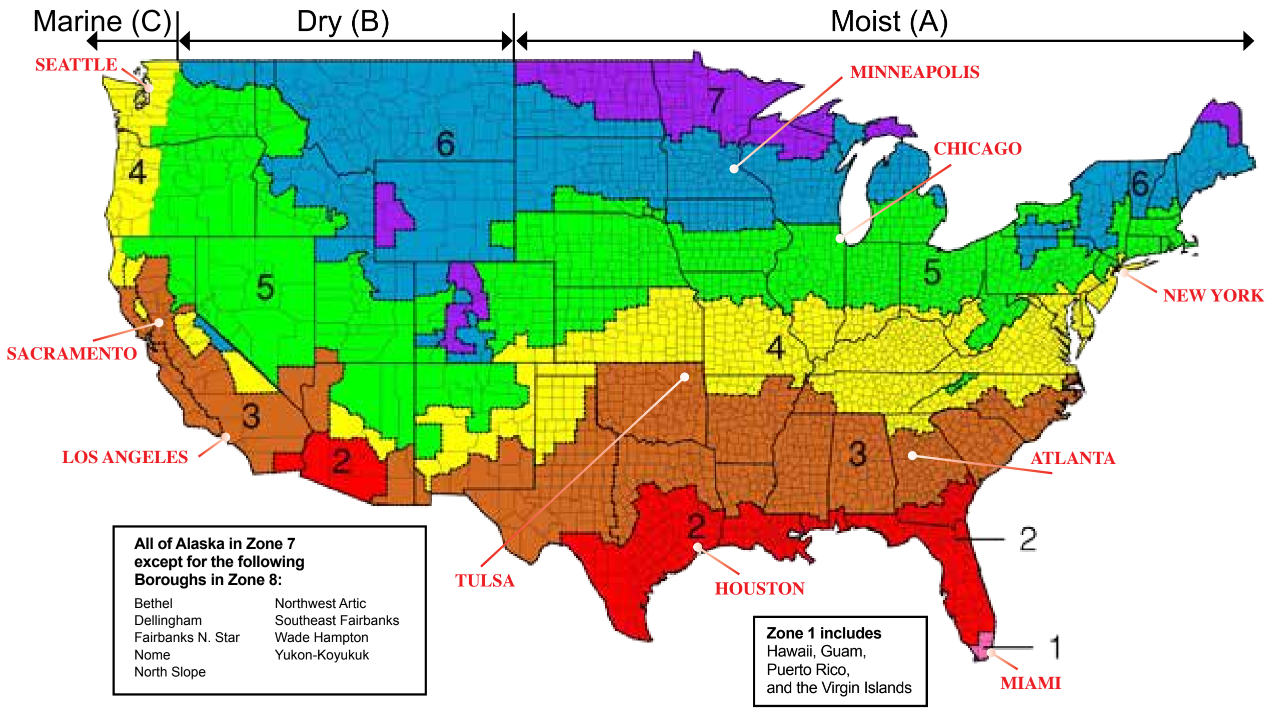


Figure 20: ASHRAE Climate Zones

		Nominal Tons							
		5	10	15	30	70	120	170	210
Cooling Savings	Atlanta	$155	$298	$440	$854	$1,964	$3,396	$4,794	$5,920
	Chicago	$77	$146	$215	$415	$952	$1,648	$2,325	$2,871
	Houston	$286	$549	$812	$1,575	$3621	$6,261	$8,838	$10,914
	Los Angeles	$47	$91	$135	$265	$611	$1,053	$1,488	$1,837
	Miami	$405	$776	$1,147	$2,222	$5,108	$8,835	$12,472	$15,401
	Minneapolis	$69	$132	$194	$376	$863	$1,492	$2,106	$2,600
	New York	$85	$161	$237	$458	$1,050	$1,818	$2,565	$3,167
	Sacramento	$59	$108	$157	$291	$654	$1,145	$1,609	$1,985
	Seattle	$14	$27	$39	$75	$170	$295	$416	$513
	Tulsa	$172	$327	$482	$928	$2,127	$3,685	$5,198	$6,418

Table 13: Estimated Cooling Savings from AAON Rigid Polyurethane Foam Cabinet ($0.12/kWh)

		Nominal Tons							
		5	10	15	30	70	120	170	210
Heating Savings	Atlanta	$99	$175	$251	$454	$1,006	$1,726	$2,416	$2,980
	Chicago	$166	$295	$423	$764	$1,693	$2,992	$4,188	$5,167
	Houston	–	–	–	–	–	–	–	–
	Los Angeles	–	–	–	–	–	–	–	–
	Miami	–	–	–	–	–	–	–	–
	Minneapolis	$192	$340	$487	$881	$1,954	$3,453	$4,833	$5,963
	New York	$140	$248	$356	$644	$1,427	$2,522	$3,530	$4,355
	Sacramento	$116	$205	$295	$533	$1,182	$2,088	$2,922	$3,605
	Seattle	$158	$280	$403	$727	$1,613	$2,850	$3,989	$4,922
	Tulsa	$114	$201	$288	$521	$1,156	$2,042	$2,858	$3,526

Table 14: Estimated Heating Savings from AAON Rigid Polyurethane Foam Cabinet ($1.20/therm)

Conclusion

As utility prices increase, heating and cooling energy lost through poor insulation and poor air seals will result in significant monetary losses to building owners. The purchase of an AAON Rigid Polyurethane Foam cabinet will reduce these monetary losses through improved thermal resistance, thermal breaks, and quality air seals. **The AAON cabinet will reduce the cost of cooling and heating operations.**

Using the AAON Energy and Economic Analysis Program, the estimated total heating and cooling costs for the representative cities are calculated. Then, using the results in Table 13 and Table 14 the annual savings for the example office building with twenty 10 ton units are shown in Table 15.

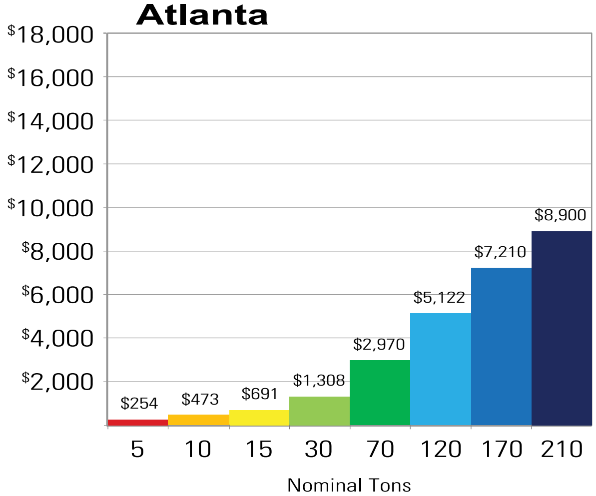
	Twenty 10 Ton AAON Cabinets Savings	Estimated Annual Cooling and Heating Costs	Estimated Percentage Savings
Atlanta	$9,460	$50,250	**19%**
Chicago	$8,820	$46,500	**19%**
Houston	$10,980	$63,750	**17%**
Los Angeles	$1,820	$38,850	**5%**
Miami	$15,520	$72,150	**22%**
Minneapolis	$9,440	$50,550	**19%**
New York	$8,180	$42,000	**19%**
Sacramento	$6,260	$49,050	**13%**
Seattle	$6,140	$30,750	**20%**
Tulsa	$10,560	$64,950	**16%**

Table 15: Percentage Savings Of An Example Office Building With Twenty 10 Ton Units

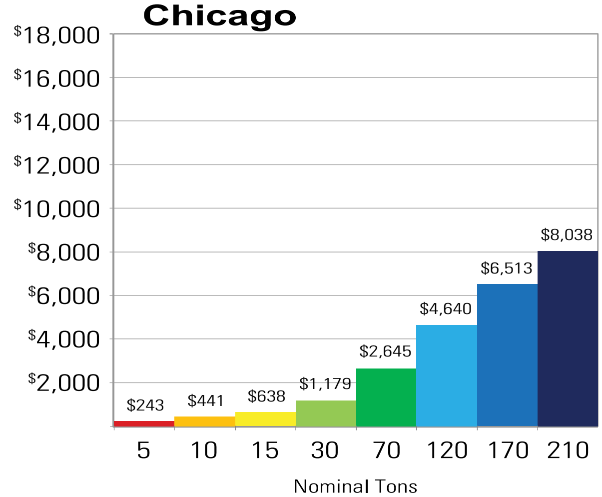
The AAON Rigid Polyurethane Foam Cabinet provides an improvement in operating cost, thermal resistance, air seals, rigidity, impact resistance, maintainability, indoor air quality, and equipment lifetime.

Annual Savings from AAON Rigid Polyurethane Foam Cabinet

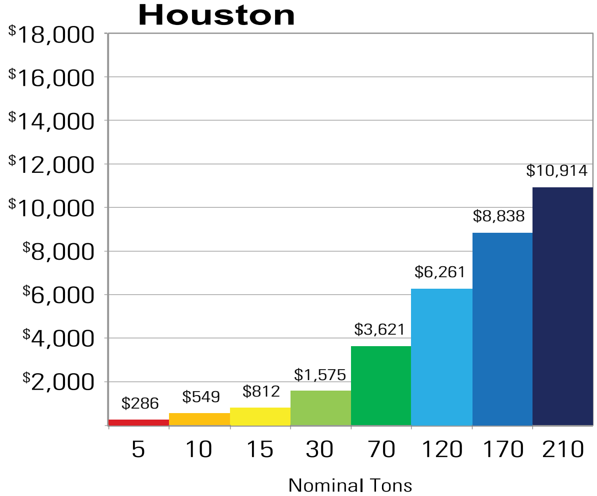
Atlanta



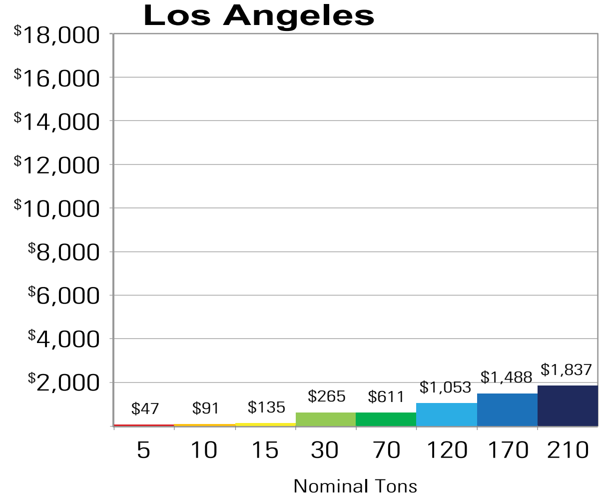
Chicago



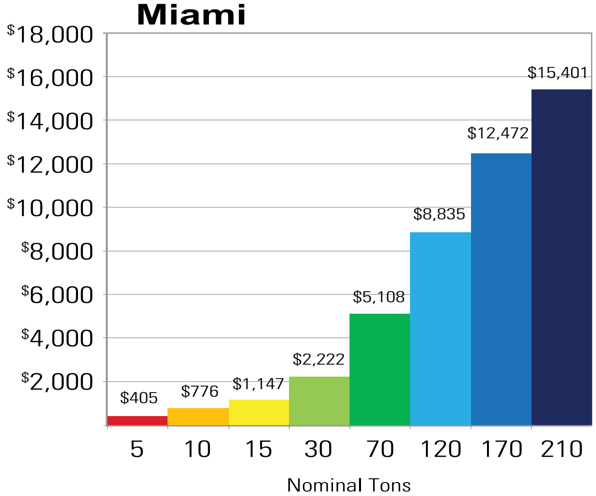
Houston



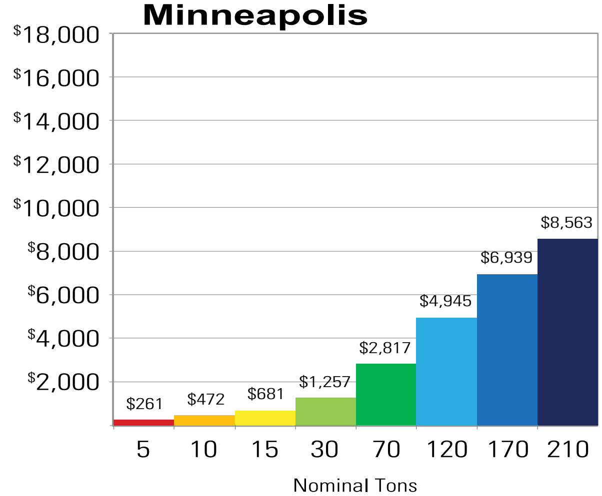
Los Angeles



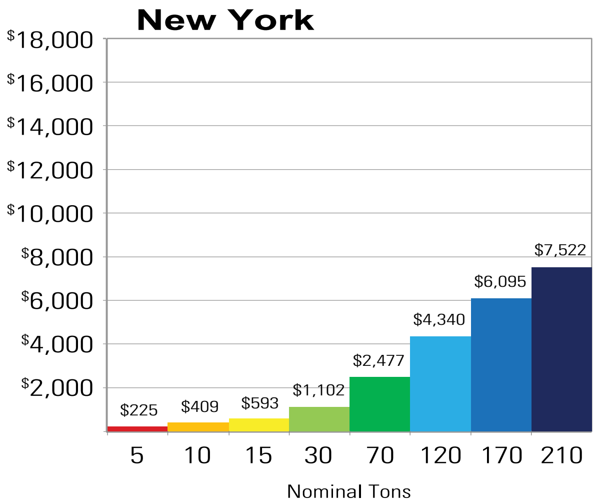
Miami

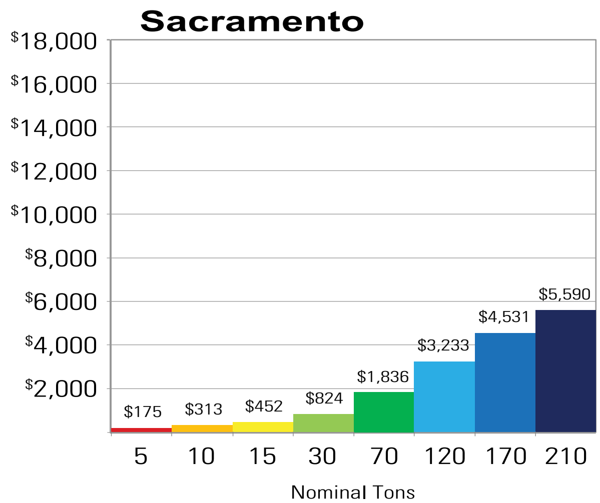


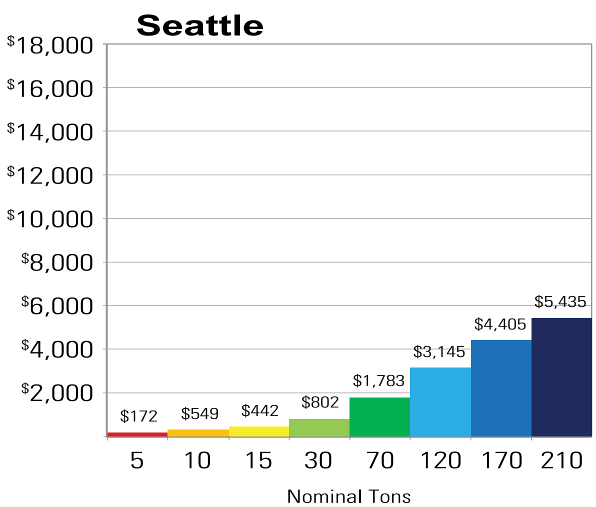
Minneapolis

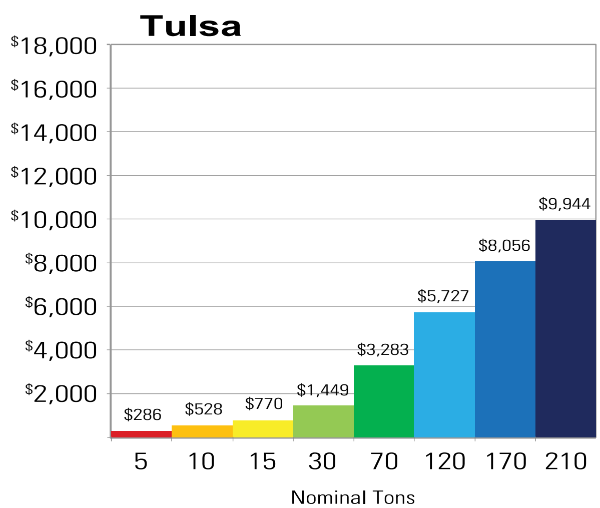


Annual Savings from AAON Rigid Polyurethane Foam Cabinet









For more information about how AAON rigid polyurethane foam panel construction can save you money, contact your local AAON sales representative.

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