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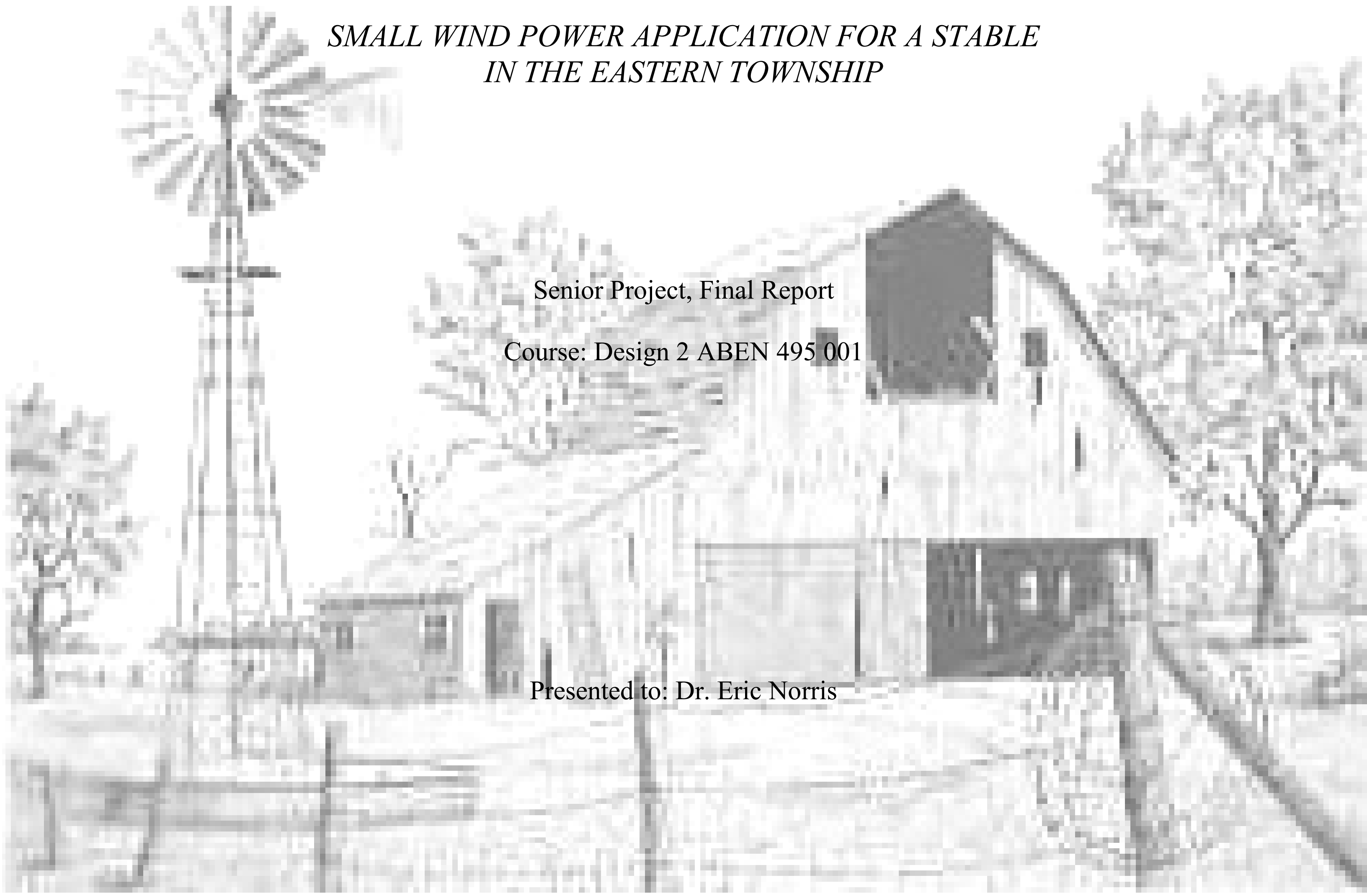
SMALL WIND POWER APPLICATION FOR A STABLE IN THE EASTERN TOWNSHIP



Senior Project, Final Report

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Introduction

Renewable energy and energy savings are hot topics these days, as economies in all industrialized nations are pushed to expand despite limited and increasingly sought-after "traditional" energy supplies, especially petroleum. Multinational oil and gas companies like Shell and BP believe renewable energy will provide 5% to 10% of the world's energy supply by 2020 and 50% by 2050 (CanWEA). Wind power is one of the major renewable energy sources being considered.

The original objective of this project was to verify the feasibility of making a stable in a wooded area energetically autonomous through the use of a wind turbine. Mr. Adrien Pilon and Ms. Louise Potvin are the owners of the site being evaluated in Bolton East, the Eastern Townships, Québec, Canada. During our analysis of the design, our focus of the project shifted. First, we realized how important it was to seek energy economy. Our current design has largely emphasized the necessity to minimize the total energy demand of the stable. Secondly, we discovered that our design approach was backwards; the design method changed from solution-to-problem (i.e. making wind power fit) to the reverse (i.e. based on local reality, find the most reasonable energy solution). Inevitably, this change inspired us to consider the possibility of applying more creative solutions to produce and save energy than might otherwise have been the case. Nonetheless, the original idea of wind power supply was thoroughly assessed and figures prominently in this report.

This report presents the results of the feasibility study for energy autonomy, starting with the energy savings, followed by the wind potential analysis and concluding with the cost analysis and possible improvements.

Objectives

The overarching goal of this project is to leave the smallest possible energy "footprint" for the new stable, or in other words, to minimize the net external energy demand in accordance with the principles of sustainable development, having complete energy autonomy as the ideal scenario.

This general goal is bounded by the following three specific engineering objectives. First, the stable design should minimize total energy requirements and maximize the utility of the available energy supplies (i.e. optimize efficiency). Within this objective, design practicality is addressed, especially with respect to the ease and frequency of system maintenance. Second, the energy supplies recommended should be appropriate for the site and stable. For instance, if the tower for a wind turbine leaves no room for a grazing yard it would be ruled inappropriate for this project. While wind power will be a major focus, other energy sources such as solar and Québec's hydro-electric grid will also be considered. Finally, the design will take into account the financial soundness of any proposed options, relative to the status quo of a Hydro-Québec energy bill.

Minimum Design Load

The first considerations in the design of a power supply system are the expected energy loads to be satisfied. Each load has been analyzed and, where practical, reduced. The design loads assessed where the lighting system, water supply system, the heating and ventilation systems and the electrical supply system. Only the results of the analysis are presented in the main body of the report. All detailed calculations can be found in the relevant appendices.

Lighting System

Based on the Canadian Farm Building Code (CFBC) recommendations, the following lighting scheme is proposed:

Table 1: Stable Lighting Scheme

	Area	Lights per Unit		Units	Lights Required
Interior					
	Feed alley	0.2	per m	12.2	3
	Boxes	1	per box	8	8
	Stalls	1	per stall	1	1
	Hayloft	3	per area	1	3
	Woodchip storage	1	per area	1	1
	Equipment Area	1	per area	1	1
	Subtotal				17
Exterior					
	Spotlights	1	per gable	2	2
	Subtotal				2
Total					**19**

In order to reduce the energy demand for lighting, we have chosen compact fluorescent lights (CFLs) for the stable. The interior lights we chose are rated at 15W and produce the lumen equivalent of 60W incandescent bulbs; the exterior lights consume 28W, equivalent to 100W incandescent bulbs (Nebraska Public Power District). The expected lifetime for CFLs is approximately 10 000 hours.

Assuming that the lights in the hayloft, wood chip storage, and equipment area will be used at most ¼ as often as the other lights and that the lights are operated 9h/day in the winter and 6h/day the rest of the year, the lighting system requires the following annual energy load:

Table 2: Lighting System Design Summary

Annual usage		3555 h
Maximum power demand		0.311 kW
Average power demand		0.255 kWh
Annual energy consumption		**627 kWh**
	Winter	*206.3 kWh*
	Non-winter	*420.3 kWh*
Energy savings relative to incandescent bulbs		74.4 %

The daily operation times are over-estimates, not taking into account day-time light availability. Less conservative operation times would reduce the energy load further.

Water Supply System

Since the water in the stable does not need to be heated, this system consumes a very small proportion of the total energy demand. Therefore, no additional energy reducing measures have been taken than those of a standard water supply system from a well. Such basic measures do, however, include minimization of losses from piping length, number of fittings, leaks, and waste of water in general. A horse-waterer design that minimizes waste, especially from splashing, will be chosen for the stable.

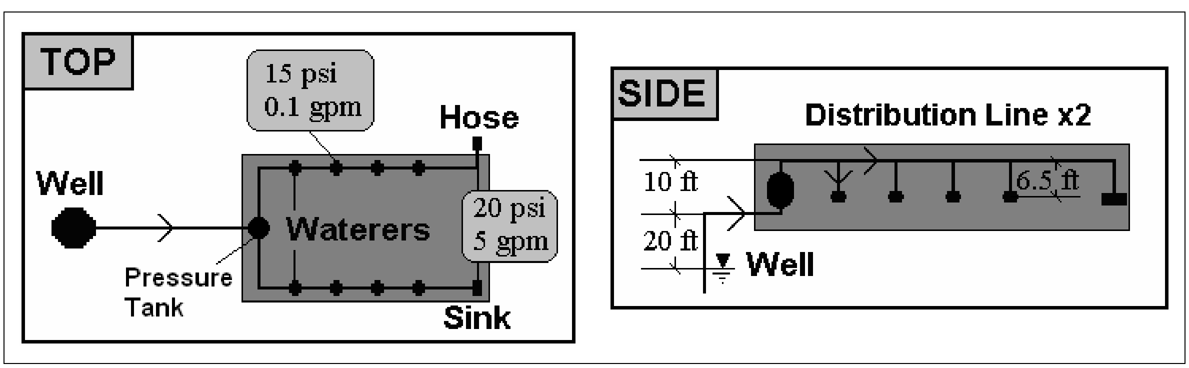


Figure 1: Schematic of the Water Supply System showing the relative positions of the water supply and water consumers.

Table 3: Water Supply System Design Summary

Supply line diameter	1.25 inches
Distribution line diameter	1 inch
Minimum pressure tank setting	50 psi
Maximum pressure tank setting	60 psi
Pump flow rate	12.0 gpm
Pump head	88 ft
Pump brake horsepower	0.560 kW
	0.75 hp
Overall safety factor	2.3
Daily water consumption	146 gal
Annual energy consumption	**40.4 kWh**

The relatively high safety factor (2.3) is due to individual safety factors of 1.5 for both the pressure tank and the pump. Lower safety margins would decrease system reliability, increasing the frequency of human time and energy inputs, as well as frequency of water shortages to the horses.

Heating and Ventilation Systems

Heating and ventilation are closely related systems based on the heat loss of the building and the loss of air (along with the heat and moisture it carries) through leaks. Based on the stable's design, heat loss values were used to determine the degree of ventilation and, subsequently, degree of heating required.

In order to be able to determine the heating requirements, 4 steps have been followed:

1. Find the total heat losses throughout the building
2. Design the ventilation system
3. From steps 1 & 2, determine the heating requirements for the barn.
4. Improve the stable design to achieve heat requirement reductions.

Heat Losses

Walls & Ceiling

In order to minimize heat losses, the thermal resistance (R-value) has been designed as high as possible for both the ceiling and the walls. In addition, an analysis of the condensation inside the walls has been achieved to insure a proper and durable design.

The results for both are the following:

Table 4:Insulation Layers in the Walls

Layer	Thickness (mm)	R (cm^2/W)
Out		
Air film	5	0.04
Wood (hemlock fir)	22	0.187
Air and wood stripping	40	0.22
Wood frame, mineral wool	150	3.2
Aluminium foil	5	0.57
Air and wood stripping	40	0.553
Interior finish (metal siding)	4	0.11
Air film	5	0.12
In		
	Total	**5**

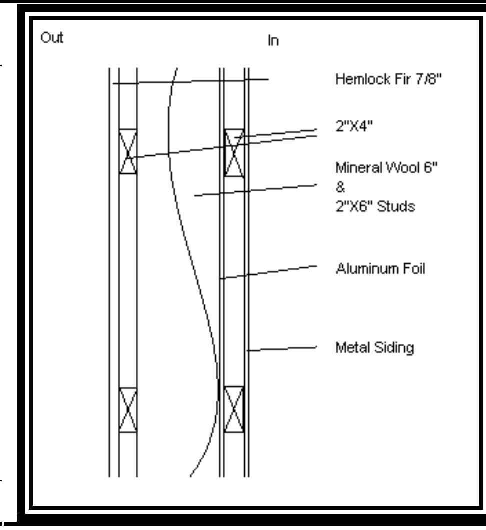
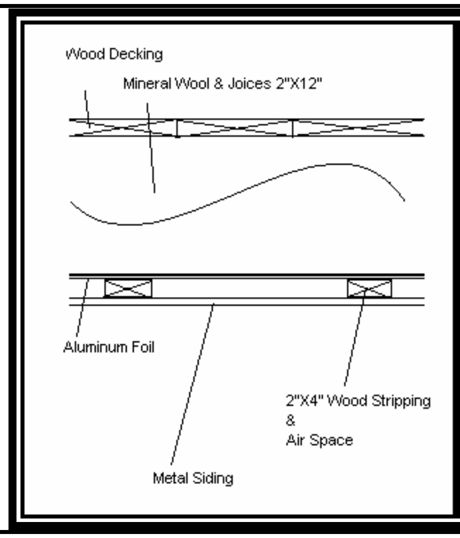


Table 5: Insulation Layers in the Ceiling

Layer	Thickness (mm)	R (cm^2/W)
Attic		
Air film	5	0.04
Wood decking	22	0.187
Wood joists, mineral wool	292	5.36
Aluminum foil	5	0.57
Air and wood stripping	40	0.553
Interior finish (metal siding)	4	0.11
Air film	5	0.12
Inside		
	Total	**6.94**



The space between studs and joists has been fully filled with mineral wool. Aluminum foil was used instead of a polyethylene vapor barrier because it cuts moisture migration more effectively and helps increase the R-value, affecting the air space. Thicker air spaces than commonly used have been favored.

Floor Slab

To improve slab insulation, an aluminum foil sheet has been added beneath the concrete slab. The aluminum foil is sandwiched between two plastic air bubble sheets and can be found in Canada under the brand name Thermofoil. It has the advantage of cutting moisture and reducing heat loss by 50% (Reflective Insulation Manufacturers' Association). This solution is superior to using extruded polystyrene, which gives the same results in terms of R-value (Reflective Insulation Manufacturers' Association) but does not cut moisture migration.

Doors
To minimize losses through the doors, we simply measured heat losses considering steel doors filled with a 1 ½ inch (38mm) thickness of polyurethane. The resulting R-value is 1.32 ($^{o}Cm^{2}$ /W).

Windows
Comparing heat transfer coefficients from greenhouse glazing material, we recognized how significant the difference was between a single and a double pane window; double-paned windows have been selected. To further improve this insulation, the use of a thermal blanket has been suggested for 12 hours a day during the winter. It decreases by half the heat transfer coefficient of the double pane window, giving an R-value of 0.44 ($^{o}Cm^{2}$ /W).

Total Heat Losses
Heat loss rates are the quotient of surface area to thermal resistance (R-value) for each feature. The overall heat loss is the sum of the losses from each feature.

Table 6: Heat Loss Rates from Each Feature of the Building

Feature	**Heat loss rate (Q) per degree of temperature gradient**	
Doors	7.9	W/^{o}C
Windows	15.7	W/^{o}C
Walls	28.2	W/^{o}C
Ceiling	20.1	W/^{o}C
Floor Slab	37.7	W/^{o}C
Total	**0.110**	**kW/^{o}C**

*Value from ASHREA Fundamentals, p.28.13 table 16

Ventilation system
Principle
The ventilation system is an important design aspect upon which the heating requirements are based. The relationship between heating requirements and the

ventilation design comes from the fact that the ventilation rates are calculated to manage heat coming from the animals. First, ventilation removes latent heat, the heat contained in the vapor from the horses' sweat and breath. The second heat managed is sensible heat or the heat that can be felt and that is released from the skin of the animal. Ventilation itself is the largest heat leak in the stable and drastically increases heating requirements.

Three season stages

The ventilation design for fall, winter and spring is done in order to maintain a minimum temperature of 10°C and a maximum relative humidity of 70% inside the barn, desired by the owners and recommended by the *Canadian farm Building Handbook*. The different flow rates for each season are based on the curve drawn from Q_{latent} and $Q_{sensible}$, where the maximum of the two flows is taken as the design air flow rate. The seasonal air flow rates are determined by dividing the curves from graph F.1 (*Appendix F*) into three equal temperature intervals and taking the maximum flow rate in each interval. These three stages have been compared with flow rates recommended from the *Canadian Farm Building Handbook*, taking an average.

Summer stage

The summer ventilation stage is different from the other three seasons. Instead of using only the temperature of the air in its calculation, it also considers the temperature of the walls and ceiling. This is justified by the high temperatures these surfaces can reach when the sun is radiating directly upon them. Such temperatures are often much higher than the outside air temperature. The high temperature levels on the outside surfaces of the building heat up the inside of the barn by heat transfer. Therefore to properly design the summer ventilation rate, an analysis of the effect of radiation on the barn's surfaces has been done (*Appendix G*). It is worth noting that heat from solar radiation is present year-round but that ventilation is only affected when the barn requires net heat removal (in fall, winter and spring, net heat input is the general rule).

Table 7: Seasonal Ventilation Requirements

Season	**Ventilation Rate** (L/s)	**Fan Energy** (hp)	**Energy Consumption** (kWh)
Winter*	120	0,03	57
Transition	300	0,07	115
Spring/Fall*	480	0,07	115
Summer**	800	0,14	230
Annual			**520**

* Value given by Dr. Suzelle Barrington and compared with graph F.1 in appendix F
** Value from the *Canadian Farm Building Handbook*

Heating System

When the ventilation rate required to control humidity is higher than the one to control temperature, the heat that the animals supply is insufficient to maintain the desired building temperature (e.g. below an outside temperature of -5°C on graphs 6.1 & 6.2, *Appendix F*). The heating requirements for the stable are calculated directly from the difference in the two flow rates and the energy required to heat the excess (from a heat perspective) outside air being ventilated through the barn to the design inside temperature, 10°C.

Table 8: Improved Design Results for Latent and Sensible Ventilation Flow Rates and Heating Requirements

T_{out} (°C)	Q_L (L/s)	Q_s (L/s)	q_{heater} (kW)	
-35	191	30	9,0	MAX
-30	195	55	6,4	
-25	196	64	5,8	
-20	200	89	4,2	
-15	207	124	2,6	
-10	218	177	1,0	MIN
-5	234	266	-0,6	–
0	263	443	-2,2	–
3	291	670	-3,3	–

The heating requirement at -35°C corresponds to the maximum power the heater must consume from the power supply. On the other hand, this is not the power at which it will run all the time. To be able to estimate whether a wind turbine can supply enough energy within the heating period, we must quantify the heating requirement in terms of energy in kWh necessary over the entire heating period. This is achieved by taking the average temperature over the three coldest months (*ASHRAE 2001 Fundamentals* and RET Screen). For December, January and February the average temperature is -10°C.

From the previous table, we observe that -10°C corresponds to 1 kW of power for the heater. The energy consumption during these 3 months is then the product of this power and the number of hours operating (see *Appendix F*). Based on this calculation, 2210 kWh are required to maintain the correct barn temperature. This value represents the total heating requirement, because the average temperatures for the adjacent months (November and March @ $T_{average}$ = -4.0 & -0.1°C respectively) are in a range that does not need a heater. Its is obvious that during these months, a heater will be needed for small period of the time, but its neglectable considering that we are going to have power from the wind during these months too; we took into consideration the peak consumption period.

In summary, **the heating system requires 2210kWh of energy from December through February with a peak power demand of 10kW**. These calculations are based on a worst case scenario for this barn. One aspect that is important to notice is that at very low temperatures (below -20°C), the more animals you have the more heating you need, contrary to what might be expected. At such temperatures, the heating requirement is mainly governed by the marked increase in ventilation required by the quantity of humidity produced by a larger number of animals.

Electrical System

This system is designed to handle all of the electrical loads within the stable. These are the heating, ventilation, and lighting systems and electrical outlet items like power tools. The building has been designed with a standard 120/240 VAC potential. The individual circuits are not used beyond 80% of their rating; in fact, the lighting circuit could carry the circuitry required for the water pump without overloading (although a 15 or 20A breaker would be recommended). The water pump, however, is expected to be supplied directly from the wind power storage. The system is summarized below.

Table 9: Electrical System Design Summary

Entrance capacity		
3 Circuits		
Voltage	240	VAC
Amperage	120	A
Wattage	21600	W
Lighting circuit		
Voltage	120	VAC
Breaker size	10	A
Peak wattage demand	311	W
Amperage demand	4	A
Convenience outlets circuit		
Circular Saw		
Voltage	120	VAC
Breaker size	30	A
Peak wattage demand	1800	W
Amperage demand	22	A
Heating and Ventilation circuit		
Voltage	240	VAC
Breaker size	80	A
Peak wattage demand	10000	W
Amperage demand	59	A

The convenience outlet system allows for a variety of power tools and appliances to be used inside the barn. Since there is already a workshop on the site, we do not expect for the stable's system to have to withstand high-use loads. For the purposes of the design, we've taken the peak use to be equivalent to operating a circular saw continuously for a ½-hour every day of the year.

Table 10 : Convenience Outlet Design Load

Circular Saw		
Energy demand	1800	W
Frequency of use	0.5	h/day
Annual energy consumption	**328.5**	**kWh**

Total Design Load

All of the inputs together yield the overall seasonal and annual design loads tabulated below:

Table 11 : Design Load

Load	**Energy Requirement** (kWh)
Winter (Dec-Jan-Feb)	
Heating and Ventilation	2267
Water pumping	10
Lighting (9 h/day)	207
General Purpose	81
	2565
Remainder of the year	
Ventilation	459
Water pumping	30
Lighting (6 h/day)	421
General Purpose	248
	1158
Total Annual Energy	**3723** (kWh)
Peak Power Demand	**12.7** (kW)

Wind Power Supply

Given the design load, the locally available wind power was assessed in order to determine whether or not this renewable resource was appropriate as the sole provider of power for the stable. Based on the data available, we have found that wind power will not be able to satisfy the entire energy demand of the barn. Even if more accurate on-site wind data is more favorable than the data available for Sherbrooke, the technical feasibility of the project will remain a very remote possibility. As an exercise, the average annual wind speed necessary to provide the barn's annual energy demand was calculated. However, even at this wind speed, 4.2 m/s, the peak power demand of the barn would never be met.

The economic feasibility of the project, as discussed later in the report, also shows that wind is not the appropriate solution at this scale.

The following technical assessment of the wind power supply involved analysis of meteorological data, conversion of wind data into available power and energy figures, and conversion of available energy into usable energy.

Wind Resource at Sherbrooke

Based on the results of a recent provincial wind assessment, our region of interest in the Eastern Townships is likely to have average annual wind speeds up to 6.1 m/s at an elevation of 30 m and corresponding wind power densities of up to 240 W/m^2. Based on the assumption that these conditions are applicable to our site, it was reasonable to look into wind power as a potential energy resource.

We were able to get free, long-term average wind speed data for the Sherbrooke A station – the national weather station nearest our site in Eastman, Quebec. The data came as part of the RETScreen® International 2000 project assessment shareware from the government of Canada's Department of Natural Resources. The wind profile at 10.1m for the Sherbrooke station is shown in Figure 2 below.

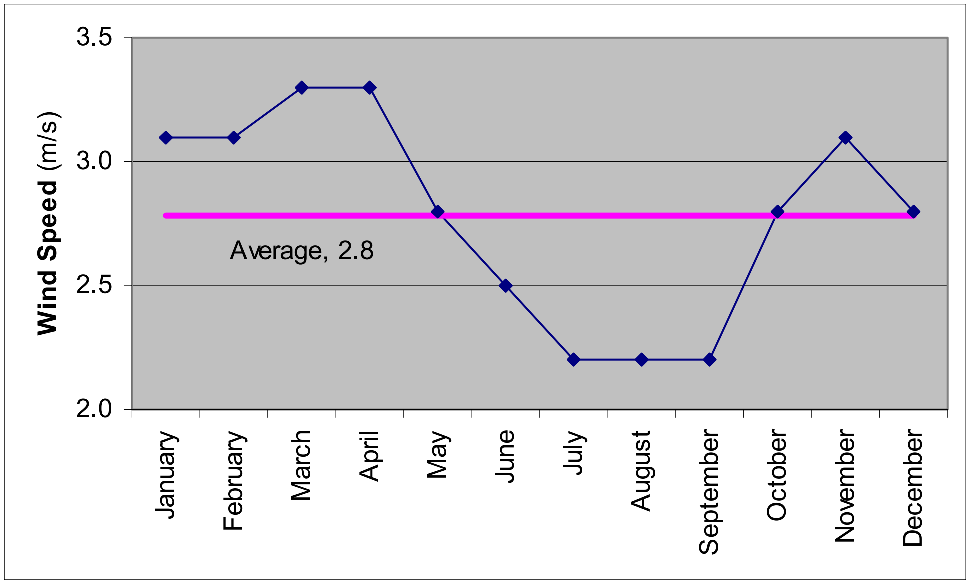


Figure 2: Wind profile and annual average wind speed for the Sherbrooke A weather station at an elevation of 10.1m

In order to be at least 10m above the tree line, the hub of our wind turbine must be placed at a height of at least 30m. For engineering purposes, under the assumption that the terrain is relatively uniform around the site of interest, the power law can be applied to make the transformation. Paul Gipe (1999) suggests a power law coefficient of 0.25 for trees, which is near the top of the applicable 0.1-0.32 range. With this coefficient, **the average wind speed at 30m is 3.7 m/s**

In order to calculate the potential wind energy available at the site, the average wind speed is not sufficient since the distribution of wind speeds has a high impact on the overall energy in the system. Lower wind speeds than the average tend to occur most frequently, while higher wind speeds are less common. However, high wind speeds have a huge impact on the power available. The statistical distribution most commonly used in wind power assessments is the Weibull distribution or its special form, the Rayleigh distribution. The Rayleigh distribution has only one variable parameter, the average long-term wind speed, making it very simple to use. Better estimates may be made if distributional data for the site in question is available. Not having this information, we've used the Rayleigh estimate for our analyses.

Wind Power and Energy

As far as a realistic calculation of power is concerned, several practical limits must be considered. The first is that a wind turbine requires a minimum wind speed to start the rotor, the value of which depends on the design of the rotor and generator. The other is that the generator of a wind turbine has a maximum capacity, at which point further increases in speed no longer improve power output. Depending on rotor design, power may actually drop past a certain design speed.

Keeping in mind that the calculated peak power demand for our system was 12.7kW, power was calculated for speeds from 0.5 to 15.5 m/s, limiting the maximum power to 12.7kW (i.e. assuming this value as the wind turbine rating).

Also, using the popular Bergey Excel (10kW) model as a representative design, all power below 3 m/s windspeeds (the cut-in speed) and above 14 m/s (the furling speed) was taken as zero. Since power is a cubic function of speed, the selection of these design speeds can critically affect the overall effectiveness and efficiency of the wind turbine to collect the available wind energy. They are also, along with the wind distribution itself, the sole determinants of the quantity of battery storage that the system needs. In general, the lower the cut-in speed, the less storage one needs. In our case, the design cut-in speed is relatively close to our annual average, requiring significant storage capacity (see Table 12 in the *Wind Turbine Design Summary* section).

Power is also a square function of rotor radius. The radius considered for our analyses was 3.5m, corresponding to the Bergey Excel's dimensions.

Further factors affecting the available power and energy are the rotor power coefficient and system efficiency factors. The power coefficient has a theoretical maximum close to 0.6, a practical maximum below 0.5 and an expected value of about 0.2 for most small wind turbines. We have used 0.2 in our design. System efficiencies relate to losses after the rotor, including the battery, inverter, wiring, and dumped energy. Losses from the choice of design speed are also included. Finally a 10% safety factor is added. Combined system efficiencies result in 61.8% of the rotor energy being usable.

The energy available over a given period of time is the sum of the products of the Rayleigh distribution and the power at each speed. **At an annual average speed of 3.7 m/s, only 71% of the total energy demand is supplied by the wind turbine**. In order to determine the average annual wind speed at which our demand would be met, we assumed that the monthly wind profile would conserve its shape and scaled-up all the average speeds linearly. By trial and error, the annual wind speed necessary is 4.2 m/s. Annual energy generation and average wind power density are shown in Figure 4.

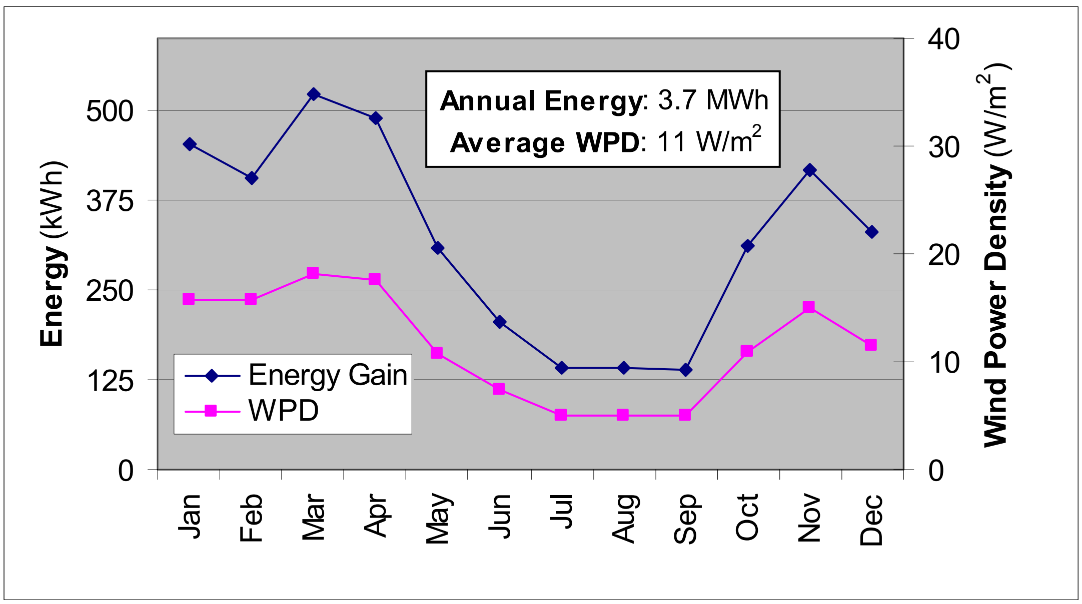


Figure 4: Annual energy production and average wind power density for an annual average wind speed of 4.2m/s based on the Sherbrooke A profile.

Wind Turbine Design Summary

Table 12 lists the main design parameters for our design turbine, including data for both the available Sherbrooke wind profile and the adjusted profile necessary to satisfy our annual energy demand.

The characteristic power and energy curves for the derived necessary wind speed at our site are given in Figure 5.

Table 12: Wind Turbine Design Summary

Design Parameters				
Hub height	30	m		
Rotor diameter	3.5	m		
Turbine capacity	12.7	kW		
Cut-in speed	3.0	m/s		
Furling speed	14.0	m/s		
Rotor power coefficient	0.2			
System efficiency	0.628			
System Load				
Annual energy demand	3711	kWh		
Winter	*2562*	*kWh*		
Rest	*1149*	*kWh*		
Maximum power demand	12.5	kW		
Sherbrooke data				
Average wind speed at hub	3.7	m/s		
Annual energy generation	2625	kWh	71	% of load
Winter	*809*	*kWh*	*32*	*% of load*
Rest	*1816*	*kWh*	*158*	*% of load*
Average wind power density	7.5	W/m^2		
Required battery capacity	306	kWh		
Adjusted data				
Average wind speed at hub	4.2	m/s		
Annual energy generation	3864	kWh	106	% of load
Winter	*1189*	*kWh*	*47*	*% of load*
Rest	*2675*	*kWh*	*237*	*% of load*
Average wind power density	11.0	W/m^2		
Required battery capacity	248	kWh		

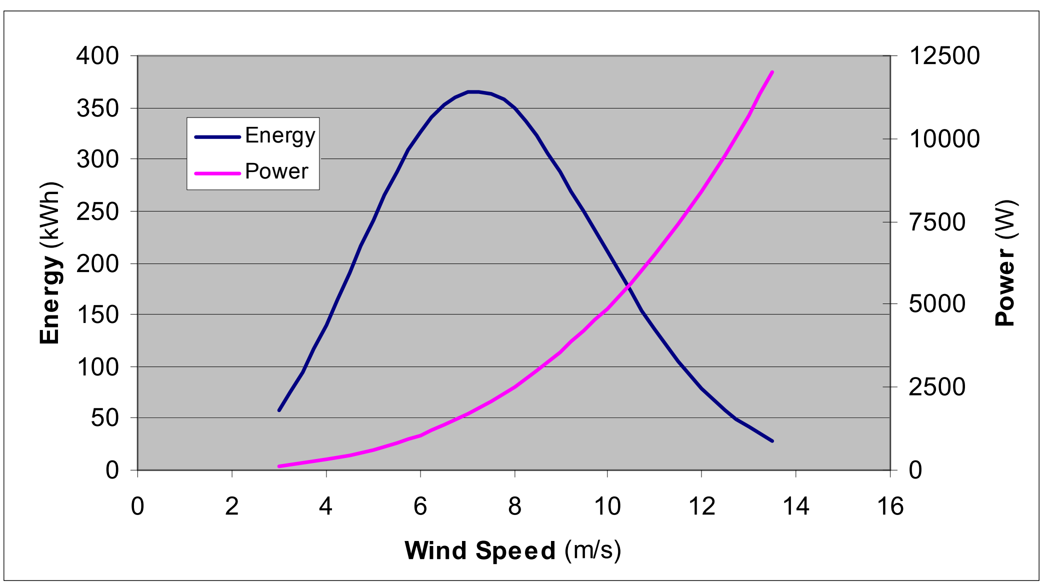


Figure 5: Characteristic curves for the adjusted Sherbrooke wind profile for which the average annual wind speed is 4.2 m/s

Wind Turbine Cost Analysis

Taking Bergey WindPower's battery-charging version of its 10kW turbine as an example, the following are the hardware costs of the autonomous wind power system. Miscellaneous items include installation.

Table 13: Wind power equipment costs

Item	Cost ($CDN)
Bergey BWC Excel-R/240 7.1m diameter, 7.5 kW battery-charging model	24,160
Guyed Lattice Tower, XLG30	10,280
Batteries and DC Power Center	6,685
Inverter (240V)	7,875
Miscellaneous	8,000
GRAND TOTAL	**57,000**

The system cost needs to be expressed in dollars-per-kilowatt for comparison as shown below.

Table 14: Cost comparison of the 7.5kW Bergey power system to a standard Hydro-Québec electrical bill

Total initial costs	57000	$CDN
Expected lifetime	20	yrs
Energy consumption	3711	kWh/year
Cost per kWh	0.7680	$CDN/kWh
Operation & Maintenance	0.1	(relative cost)
Total cost	0.8448	$CDN/kWh
Cost of Hydro (March, 2004)	0.0488	$CDN/kWh
Wind is more expensive by:	**17.3**	**times**

Thus, even when using the cost of a system that is undersized by 5kW for economic analysis, an autonomous wind power system does not make sense in view of current options.

Alternatives

Being concerned with finding an appropriate energy supply design and keeping in mind that ready-made solutions need not be force-fitted to the problem (especially when several other ones are available), we were inspired to push for further ideas and to rethink the way we used the energy available on the site.

Passive Solar Heating Duct

First, while we were designing the ventilation system, we were wondering how could we use the higher temperatures that the walls and roofs reach. From greenhouse energy savings studies, we found that some greenhouse designs use solar walls (see figure 6). Instead of using this design directly (and redesigning the entire barn in the process), we decided to take advantage of the temperature the south-facing roof reaches. Referring to graph G.2 in appendix G, we noticed that the roof inclined to 63.4° can reach a temperature of 30°C around noon for when the outside air temperature is -10°C and the sky is clear. From there, we verified how much heat we could get from that roof by simply letting air flow up along the underside of the surface to be redirected inside the building. Instead of the conventional winter ventilation inlet design (figure H.1, appendix H), we have designed the passive solar heating duct (figure H.2, appendix H), which would have the advantage of being fairly simply retrofit to any other barn.

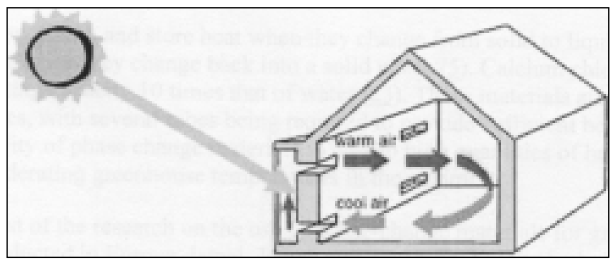


Figure 6: Solar wall collecting radiation from the sun.

The results are quite interesting, providing a heat savings of 10% per year for a low investment (see the cost analysis below). Table 15 shows some results for air entering the barn using after having passed through the duct. Note that above -10°C outside, no additional heating would be required when the sun is shining.

Table 15: Theoretical ventilation inlet temperatures using the passive solar heating duct

T_{out} [°C]	$T_{air\ entering\ barn}$ [°C]
-35	-11.74
-30	-6.74
-25	-1.74
-20	3.26
-15	8.26
-10	13.26
-5	18.26
0	23.26

Water Storage

Another possibility that we considered was to store the energy that we have in surplus during summer season. With the required wind speed of 4.2 m/s, there is a seasonal energy surplus of 2675 kWh. This energy could be used to pump water into storage in the attic and that we could eventually use as potential energy for simple water distribution or to drive a small water turbine. None of the calculations have been performed, but it is an idea for future design!

Smaller wind turbine

Finally, considering the exorbitant cost of an autonomous wind power design, a possible venue would be to stay with Hydro-Quebec for the main energy source, but use a smaller wind turbine size, like 1 or 2.5 kW, as a supplemental supply. Smaller wind turbines are designed for a lower cut-in speed and maximum energy capture for wind speeds in the region. Having Hydro-Québec as the main energy supplier would reduce (or eliminate) expensive battery capacity since wind power would only be used when available. Perhaps just having a wind turbine, even if it does not give the entire power needed, provides a first step in inspiring people to think about different ways of producing power.

Alternatives Cost Analysis

Table 16: Comparison of Cost of Material to Money Saved in Heating (Based on hydroelectricity cost in March 2004)

Hydro-Québec cost per kWh:	**0,0488**	**$/kWh**
Material:	Mineral wool	
Cost mineral wool 12":	0,89	$/ft2
Cost mineral wool 6":	0,45	$/ft2
Cost mineral wool 4":	0,30	$/ft2
Cost Thermofoil:	0,38	$/ft2
Cost 2" Extruded Polystyrene:	0,94	$/ft2
Cost of plywood 1/2":	0,89	$/ft2
Area walls:	1600	ft2
Area ceiling:	1500	ft2
Area slab:	1500	ft2
Area per duct for insulation purpose:	22,3	ft2
# of ducts:	12	Ducts
Cost of material for ducts:	490,09	$
Cost of labor for ducts:	400,00	$
Total cost for ducts:	890,09	$
Standard design, cost of material:	1143,71	$
Improved design, cost of material:	4664,85	$
Difference in cost for material:	3521,14	$
Standard design, kWh of heating per year:	7142,40	kWh
Improved design, kWh of heating per year:	2010,00	kWh
Difference in kWh per year:	5132,40	kWh
Heating energy consumption of improved vs standard:	28%	
Difference in cost for heating:	250,46	$
Payback period:	**14**	**Years**

Conclusion

After spending a lot of time and effort on wind power analysis, it can seem disappointing to generate such negative results. On the other hand, studying many aspects of energy savings, studying how wind power works, it all adds together to give us new knowledge, and a better understanding of the reality of energy saving and renewable energy sources. This sector is the future and we have to redefine the way we consume energy. Energy consumption at contemporary, current and projected rates in both industrialized and industrializing nations has a massive impact on the global ecosystem's finite energy sources, and this environmental cost is hardly ever considered when plans for expansion and "development" are made. This design project has opened us to this field, an area in which we both want to become professionally and personally involved. People need to be conscientious about their consumption and the effects that it has. For us, this project was another solid step towards this green and living direction.

Through this design, we realized how much we can save by focusing on energy savings from insulation and the effect that it has on the heating requirements. Referring to appendix F, graph F.3, we end up with a heater consuming about 3 time more than the one with the more efficient design (The comparison between our improved design and a standard one has been done in appendices E and F, table E.3 & graphs F.2 and F.3).

Some of the designs and ideas mentioned in this report can be easily applied, as with the insulation improvement and the solar duct design. Concerning the use of a wind turbine, however, the wind speeds around Sherbrooke are not high enough to produce 10kW on a single turbine. Considering the heavy cost of wind power systems, it would not be reasonable to construct a small wind farm either. Thus the only possibility for wind would be as a supplementary supply from a micro turbine. Other issues of appropriateness (such as land use and aesthetic value) that were not discussed in the original wind power analysis (because technical and economic reasons ruled it out as a solution) should be taken into consideration for any power supply deemed feasible.

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Thanks

- Adrien Pilon and Louise Potvin for the opportunity that they offer to us.

- Jérome Breton, Technician from the CNRC who introduced us to windmills. "Strive for negative wattage wherever possible [relative to standard designs]," told us Mr. Jerome Breton technician at National Research Council Canada.

- Mr. Tomas Lawand for his philosophy. According to Mr. Lawand, our project had a big lack in its start; we were creating a problem from a solution, which went against his philosophy. He believes that being stuck to the fact that we want a wind turbine to supply energy makes your mind closed to many other solutions which could have been considered before hand.

Appendices

Appendix A: Lighting System Calculations

Given the following system parameters:

17 15W interior lights (12 general and 5 in utility areas)
2 28W exterior lights
Expected use of the utility area lights only ¼ of the time the others are on
9h of use per day during the winter (December through February)
6h of use per day during the rest of the year

The maximum power demand of the lighting system, calculated for simultaneous use of all lights, is the sum of the power consumed by each light:

$$\begin{aligned} P_{\max} &= 17 \times 15W + 2 \times 28W \\ &= 311W = \underline{0.311kW} \end{aligned}$$

The average power demand, used in annual and seasonal energy consumption calculations, takes into account the relative use of each light:

$$\begin{aligned} P_{average} &= \sum Qty \times Power \times Use\ Factor \\ &= (12 \times 15W \times 1) + (5 \times 15W \times 0.25) + (2 \times 28W \times 1) \\ &= 255W = \underline{0.255kW} \end{aligned}$$

Energy consumption was calculated for both the winter and non-winter periods:

$$\begin{aligned} E_{annual} &= E_{winter} + E_{non-winter} \\ &= (9\, h/day \times 90days + 6\, h/day \times 275days) \times 0.255kW = \underline{627kWh} \end{aligned}$$

Savings relative to incandescent bulbs is the ratio between power requirements for the two technologies. Since equivalent incandescent bulbs to those used have power ratings of 60 and 100W, their average power consumption would be 995W, yielding:

$$\begin{aligned} E_{savings} &= 1 - \text{Relative energy consumption} \\ &= 1 - \frac{0.255kW}{0.995kW} = 1 - 0.256 \\ &= 0.744 \text{ (or } \underline{74.4\%}\text{)} \end{aligned}$$

Appendix B: Water Supply System Calculations

Given the following system parameters:

8	horses
12	gallons per horse per day
15	psi at each waterer
2	hoses or sinks
5	gpm per hose/sink
50	gallons per day from the hose and sink
20	psi at each hose/sink

Stable dimensions of 30' wide by 50' long
Layout according to Figure 1 in the *Water Supply System* section

100'	Maximum distance from the well to the stable
0.75	Assumed long-term pump efficiency

Assuming that the peak rate of water consumption is the horses' daily water requirement consumed in 2 hours, the maximum flow rate can be established as:

$$q_{horses} = 12\frac{gal}{horse \cdot day} \times 8 horses \times \frac{day}{2h} \times \frac{h}{60\min} = 0.8 gpm$$

$$q_{general} = 5\frac{gpm}{outlet} \times 2 outlets = 10 gpm$$

$$q = q_{horses} + q_{general} = \underline{10.8 gpm}$$

From this value, we chose 12gpm as the design flow rate since it is a standard value. From the flow rate, pipe and fitting losses can be calculated. The parameters that were used are tabulated below:

Table 2.1: Pipe and Fitting Loss Data

Pipe Section	**Pipe material**	**Nominal Diameter** (inches)	**Diameter** (inches)	**Length** (feet)	**Design Flow** (gpm)
Pump to pressure tank	plastic	1.25	1.380	110	12.0
Distribution from tank	plastic	1	1.049	71.5	6.0

Pipe Section	**Friction Coefficient** C	**Fittings**	**Equivalent Length** (ft)	**Friction Loss** (ft/100ft)	**Friction Loss** (ft)
Pump to pressure tank	140	1 - 90^{o} elbow 1 - tee, turning	19	2.3	3.0
Distribution from tank	140	2 - 90^{o} elbows 4 - tees, straight	28	2.4	2.4

The length of pipe from the pump to pressure tank includes a 10ft rise in elevation. The length of distribution pipe from the tank combines half the width of the building, its full length and a 6.5ft drop. Flow is split in two from the tank in order to service each side of the building. Friction losses in the fittings are expressed as equivalent lengths of similar diameter pipe. Friction loss in the pipe was calculated with the below empirical formula:

$$h = \frac{1043.8(gpm/C)^{1.85}}{d^{4.87}}\left[ft/100\,ft\right]$$
$$= \frac{1043.8(12/140)^{1.85}}{1.380^{4.87}} = 2.3\,ft/100\,ft$$

Total friction loss was calculated by multiplying *h* by the effective length of pipe (i.e. the length including the fittings loss equivalents).

The required pressure head includes:

$$H = Lift + Elevation + Friction + Outlet$$
$$= 20\,ft + (10 - 6.5)\,ft + (3.0 + 2.4)\,ft + (20\,psi \times 2.307\ ft/psi)$$
$$= 75\,ft\ H_2O = \underline{33\,psi}$$
$$(H \times 1.5 \cong 115\,ft \cong 50\,psi)$$

Applying a safety factor of 1.5 for accidental leaks, the resulting head of 50psi represents the minimum setting for the pressure tank. By rule of thumb, adding 20ft of pressure yields the maximum pressure setting for the tank. That value is 125ft or 60psi.

The maximum tank setting is the minimum pump head requirement. The brake horsepower is then calculated by:

$$P = \frac{gpm \times 8.33\,lb/gal \times ft}{33000\ ft \cdot lb/hp} \div efficiency$$
$$= \frac{12 \times 8.33 \times 125}{33000} \div 0.75$$
$$= 0.50hp = \underline{0.375kW}$$

From this calculation we selected a 0.560 kW (3/4 hp) pump, yielding a pump safety factor of 1.5 and an overall safety factor of 2.3 (1.5 x 1.5). The energy consumed by this pump is found by:

$$W = P \times t$$
$$t = \frac{\text{daily use}}{\text{supply rate}} \times \text{number of days}\left[h/year\right]$$
$$= \frac{(8 \times 12 + 50)\,gal/day}{12gpm} \times \frac{hour}{60\,\text{min}} \times 365\,days/year$$
$$= 72.2\,h/year$$
$$W = 0.560kW \times t = \underline{40.4\,kWh/year}$$

Appendix C: Heat Losses though Walls and Ceilings

Heat Loss vs. Condensation

These calculations show the insulation design with its condensation behavior inside the walls and ceiling at each layer. It uses the worst case scenario, the maximum relative humidity inside allowed of 70% with the coldest temperature outside (-30°C for Sherbrooke, increased to -35°C as safety factor, ASHREA table 2A p 27.24). The maximum relative humidity for summer is not considered, because the air temperature is sufficient to dry out any condensation that may occur within the insulating layers.

The parameters encountered in these calculations are the R-value for insulation, the D-value for water vapor (humidity resistance), the P_s value for air water vapor pressure at saturation, and finally, the P_w value or air water vapor pressure.

Insulation and R-value

All R-values used and combined together to establish a total equivalent R-value come from notes of Dr. Barrington in the course of Ventilation of Agricultural Structures which refers to ASHREA Fundamentals Handbooks 2001, table 4 p. 25.5-25.8. R-values for materials are based on material properties and R-values for air space (or 1/f in tables) are based on tables which relate the emissivity of both surfaces surrounding the air space affecting natural convection and giving a value for R.

When more than one substance (air or material) is found in a single layer, the following formula is used to calculate the equivalent thermal resistance:

$$R_{eq} = \frac{R_1 \times R_2}{R_1 \times \dfrac{A_2}{A_T} + R_2 \times \dfrac{A_1}{A_T}}$$

For example, R_{eq} for air and wood stripping @ 610mm c/c, 38mm thick by 89mm wide is:

$$R_{eq} = \frac{R_{wood} \times R_{air}}{R_{wood} \times \dfrac{A_{air}}{A_{air+wood}} + R_{air} \times \dfrac{A_{wood}}{A_{air+wood}}}$$

$$R_{eq} = \frac{\left(8.5\, km^2/W \cdot m \times 0.038m\right) \times 0.59\, km^2/W}{\left(8.5\, km^2/W \cdot m \times 0.038m\right) \times \dfrac{(610-89)mm}{610mm} + 0.59\, km^2/W \times \dfrac{89mm}{610mm}}$$

$$R_{eq} = 0.53\, km^2/W$$

Finally, to determine an effective average temperature for any particular insulation layer, useful in humidity transport problems, the R-value is applied as follows:

e.g.: Layers: Air film and wood
R-value: 0.04 and .187 [cm^2/W], respectively
R-value total: 4.44 [cm^2/W]
T_{in} of 10°C and a T_{out} of -35°C

$$\begin{aligned} T_{Air/Wood} &= T_{Out} + (T_{In} - T_{Out}) \times \frac{\sum R_{Air}(\text{Out to In})}{R_{Total}} \\ &= -35^oC + \left[10^oC - \left(-35^oC\right)\right] \times \frac{\sum (0.04 + 0.187)°C \cdot m^2/W}{4.44\,°C \cdot m^2/W} \\ &= -32.7°C \end{aligned}$$

Water vapor resistance and D-value:

All D-values are taken from ASHRAE 2001 fundamentals handbook, table 9 p 25.16. For D_{eq} and passage from a layer to another for P_w, it uses the same principle as the R-value uses (as shown above).

P_s is found from psychrometric relation using many constant and parameter (please refer to Dr Barrington note of ventilation of Agricultural Structure for these relationship). It is calculated relatively to the temperature at each layer (T in °K).

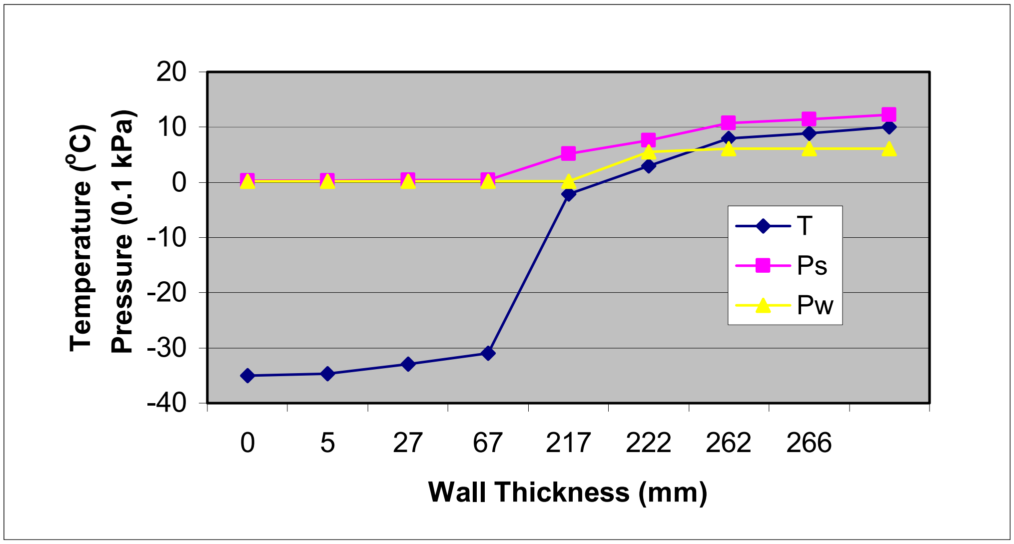
P_w is calculated based on P_s for the inside and outside temperatures:

$P_w = P_s \times \phi / 100$ ϕ which is the relative humidity at the respective surface.

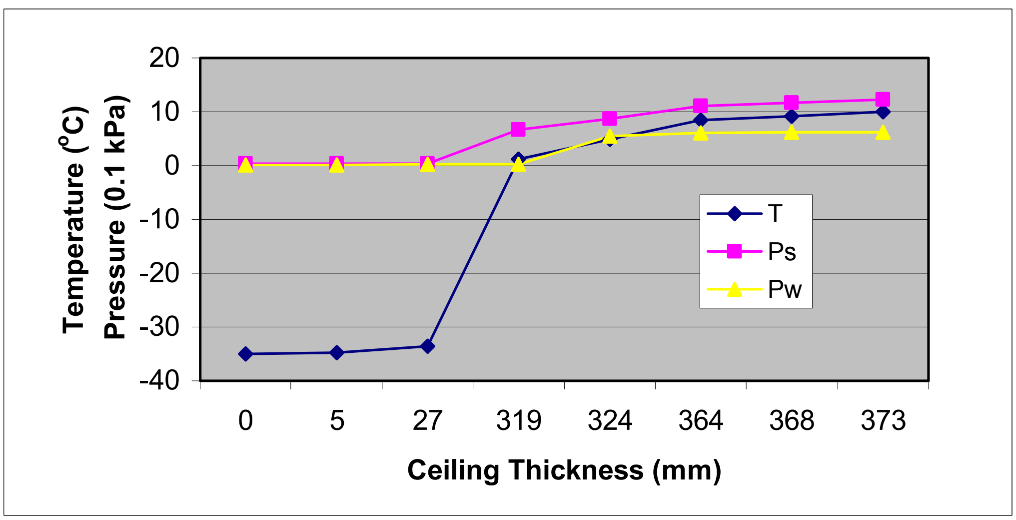
P_w is also calculated for the intermediate layer based on the D-value accumulated through the wall with the following formula:

$P_{w_{1/2}} = P_{w_{out}} + \Delta P_w \times \dfrac{\sum D_1}{D_{total}}$ Used in the same principle as the R-value does.

The principle of the following graphs relies on P_s and P_w which dictates whether we are going to have condensation in the walls and ceiling or not. Ps being the air water vapor pressure at saturation and P_w being the actual air water vapor pressure, both calculated at same temperature. From there, if P_w is greater than P_s that means we have condensation at that point inside the wall, because we exceed saturation point.



Graph C.1: Humidity migration through the walls for condensation verification given outside and inside temperatures of -35°C and 10°C, respectively, at standard relative humidity.



Graph C.2: Humidity migration through the ceiling for condensation verification given outside and inside temperatures of -35°C and 10°C, respectively, at standard relative humidity.

Appendix D: Heat Losses though Doors, Floor and Windows

Heat loss through doors

$$\frac{Q}{\Delta T} = \left(\frac{A}{R}\right) \times \# \, doors$$
$$= \left(\frac{3.5' \times 8'}{34.7[Km/W] \times 0.038[m]}\right) \times \left(\frac{0.3048m}{ft}\right)^2 \times 4$$
$$= \underline{7.9[W/°C]}$$

*The 34.7 Km/W is thermal resistance of extruded polystyrene from ASHREA fundamental p25.6 table 4

Heat loss through floor slab

$$\frac{Q}{\Delta T} = Perimeter\_of\_building[m] \times U[W/K \cdot m\_of\_perimeter] \div coefficient\_for\_temperature$$
$$= (2*(30+50)[m]) \times \left(\frac{0.3048m}{ft}\right) \times 2{,}32[W/Km] \times 50\% \div 1.5$$
$$= \underline{37.7[W/°C]}$$

*The 1.5 factor comes from the fact that earth is an isolator and has an influence on the ΔT. With earth considered: ΔT is (T_{in}-($T_{mean\ in\ January} - T_{amp.}$); T_{amp} and $T_{mean\ in\ January}$, respectively, were taken from ASHREA fig. 6 p 28.and table 2A p27. T_{in} as in other calculations, uses the worst case event: 10 °C. Without earth consideration it is simply $T_{in} - T_{out}$ with the correction factor of 1.5
**U value and formula used are taken from ASHREA fundamental p. 28.13, table 16
***The 50% comes from the Reflective Insulation Manufacturers Association which approximated the heat loss reduction using an aluminum foil at 50%

Heat loss through Windows

Considering: -10 windows
-Double glass sealed using a thermal blanket for night 12 hours per day

$$\frac{Q}{\Delta T} = (A \times U) \times \# \, windows$$
$$\frac{Q}{\Delta T} = \left(2.5' \times 3' \times 2.25[°Km^2/W]\right) \times \left(\frac{0.3048m}{ft}\right)^2 \times 10$$
$$\frac{Q}{\Delta T} = \underline{15.7[W/°C]}$$

*The 2.25 [W/m^{2}K] heat transfer coefficient comes from the average of 3 & 1.5 [W/m^{2}K], which are the values for double glazing windows and double glazing windows with a thermal curtain respectively. From notes of Dr. Barrington in the course of Agricultural Structure, Greenhouse glazing section, table 11.4.

Appendix E: Total Heat Losses

Calculation

$$\frac{Qtotal}{\Delta T} = \frac{(Qdoors + Qwindows + Qwalls + Qceiling + Qslab)}{\Delta T}$$
$$= \frac{(7.9W + 15.7W + 28.2W + 20.1W + 37.7W)}{\Delta T}$$
$$= \underline{0.110 kW / \Delta T}$$

Comparison between Improved and Standard Designs

Table E.1: Results for Heat Losses in each feature of the building for the improved design

Feature	Heat Loss
Doors	7,9 W/°C
Windows	15,7 W/°C
Walls	28,2 W/°C
Ceiling	20,1 W/°C
Slab*	37,7 W/°C
Total	**0,110 kW/°C**

* U value from ASHREA Fundamentals, p.28.13 table 16

Table E.2: Results for Heat Losses in each feature of the building for the standard design

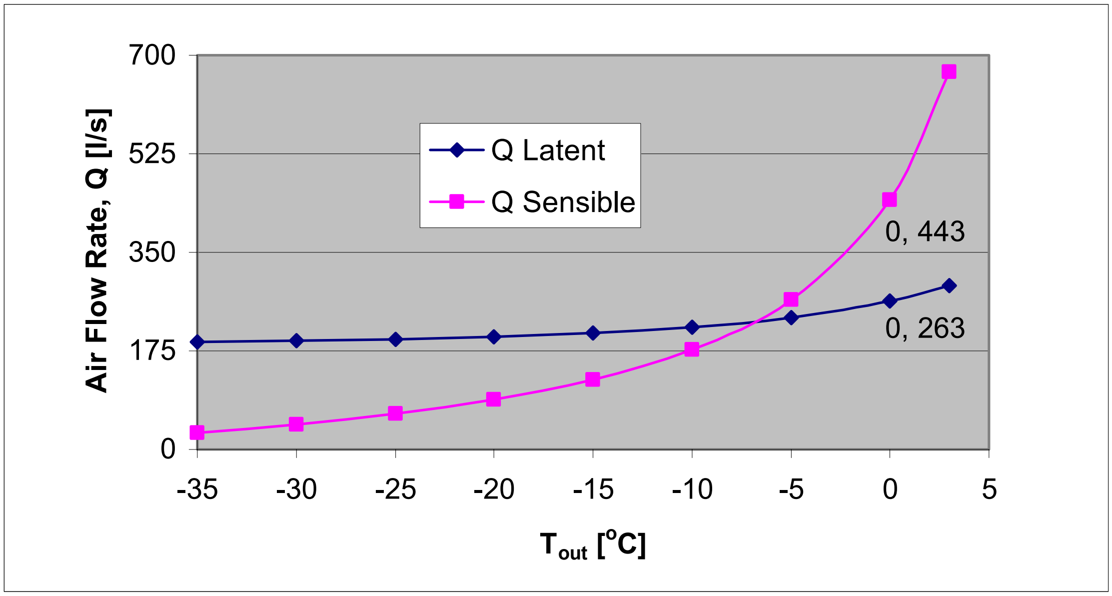
Feature	Heat Loss
Doors	15,8 W/°C
Windows	43,9 W/°C
Walls	51,4 W/°C
Ceiling	36,8 W/°C
Slab*	75,4 W/°C
Total	**0,223 kW/ °C**

* U value from ASHREA Fundamentals, p.28.13 table 16

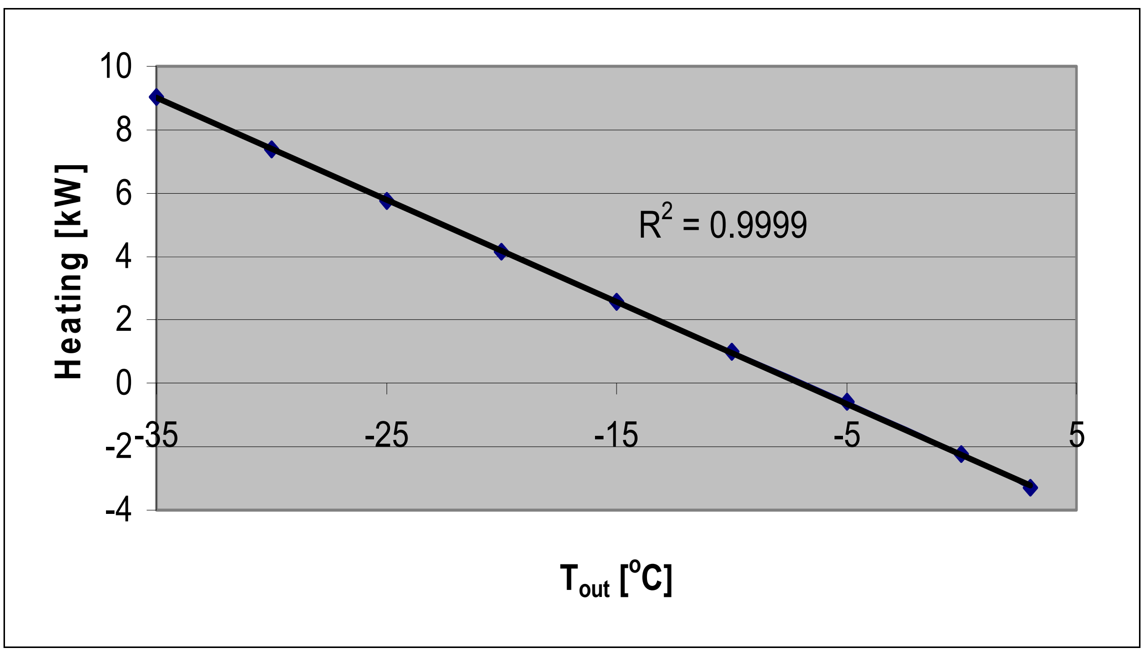
Table E.3: Comparison between the two designs in terms of heat loss reduction

Feature	Heat Losses (W/°C) Improved	Heat Losses (W/°C) Standard	Reduction
Doors	8	16	2,0 x
Windows	16	44	2,8 x
Walls	28	51	1,8 x
Ceiling	20	37	1,8 x
Slab	42	75	1,8 x
Total	**114**	**223**	**2,0 x**

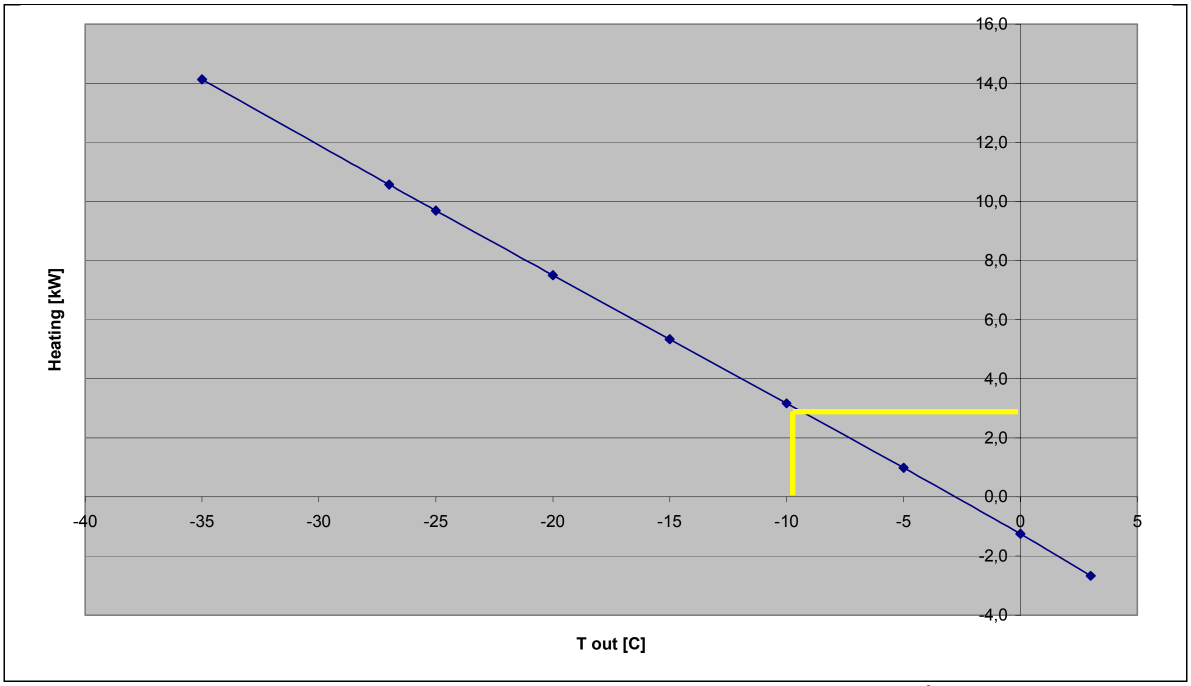
Appendix F: Ventilation and Heating Requirements



Graph F.1: Ventilation curves based on the improved design to maintain an 8 horse stable at 10°C and 70% R.H.



Graph F.2: Heating requirements based on the improved design to maintain an 8 horse stable at 10°C and 70% R.H.



Graph F.3: Heating requirements for a standard horse stable design maintained at 10°C and 70% R.H. for 8 horses weighing 550kg each. For example, at -10°C outside roughly 3kW of heating power is necessary (note that this is triple the value achieved for the efficient design).

Design Parameters

- 8 horses: Haflinger breaded with Standard Bread, average mass of 550kg each.
- T_{in}: 10°C , T_{out}: -35 to 3°C
- ϕ_{in}: 70%, ϕ_{out}: 70% — Relative humidity approximated and recommended for the building by Dr. Barrington. (maximum relative humidity for inside would be 75%)
- $q_{sensible}$ and q_{latent} are taken from Ventilation of Agricultural Structure p.134, table 7.1, based on the value for beef cattle (recommended by Dr. Barrington)
- W_{in}: Taken from a psychometric chart based on T_{in} and ϕ_{in}
- W_{out}: Taken from a psychometric chart based on each T_{out} and for ϕ_{out}
- hfg_{in} =2502,5-2,386* T_{in}
- q loss: Total heat loss throughout the building

Here are the equations used to calculate the sensible and latent heat air flow rates (For the example, the calculation used temperature and properties @ T_{out}: -35°C):

Latent Heat

$$Q_l = \frac{q_l[w/kg] \times \#animals \times mass.per.animal[kg] \times 1[kW]/1000[W] \times 1000[l/m^3]}{(W_{in} - W_{out})[g/kg\_dry.air \times 1[kg]/1000[g] \times \rho[kg/m^3] \times hfg_{in}[kJ/kg]}$$

$$Q_l = \frac{0.7[w/kg]\times 8horses\times 550[kg]\times 1[kW]/1000[W]\times 1000[l/m^3]}{(5.3-0.137)[g/kg\_dry.air]\times 1[kg]/1000[g]\times 1.24[kg/m^3]\times 2478.64[kJ/kg]}$$
$$Q_l = 194[l/s]$$

Sensible Heat

$$Q_s = \frac{\{(q_s[w/kg]\times \#_{animal}\times mass[kg]\times 1[kW]/1000[W]) - (Q_{loss}[kW/°C])\times(T_{in}-T_{out})[°C]\}\times 1000[l/m^3]}{1.006[kJ/kg°C]\times(T_{in}-T_{out})[°C]\times \rho_{in}[kg/m^3]}$$
$$Q_s = \frac{\{(1.5[w/kg]\times 8\times 550[kg]\times 1[kW]/1000[W]) - (0.110[kW/°C])\times(10--35)[°C]\}\times 1000[l/m^3]}{1.006[kJ/kg°C]\times(10--35)[°C]\times 1.24[kg/m^3]}$$
$$Q_s = 29[l/s]$$

Heating Requirements

$$q_{heater} = (Q_l - Q_s)[l/s]\times(1.006[kJ/kg°C](T_{in}-T_{out})[°C])\times \rho_{in}[kg/m^3]/1000[m^3/l]$$
$$q_{heater} = (194--30)[l/s]\times(1.006[kJ/kg°C](10--35)[°C])\times 1.24[kg/m^3]/1000[m^3/l]$$
$$q_{heater} = 9.2[kW]$$

Energy Requirement for Heating

$$Heating\_Energy = 1[kW]\times 24[h/day]\times 31[days/month]\times 3[month]$$
$$Heating\_Energy = 2210[kWh]$$

Ventilation Requirements

E.g.: <u>Winter flow rate</u>

From graph F.1, the ventilation flow rates calculated ends up to about 200 l/s, but the data from book given by Dr. Barrington give a value of 12 l/s per animal (96 l/s). So an approximation to 120 l/s has been done. This approximation represents well the steps followed to calculate the air flow rate.

The conversion from hp to kWh is simply

$$Energy\_Consumption = Fan\_Energy[hp]\times 0.7457[kW/hp]\times 24[h/day]\times 31[day/mth]\times 3[mth]$$
$$Energy\_Consumption = 57[kWh]$$

Appendix G: Incoming Radiation

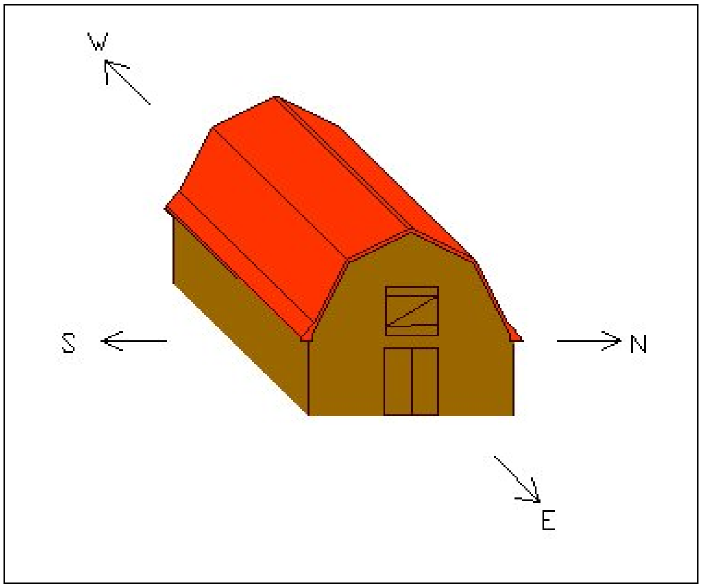
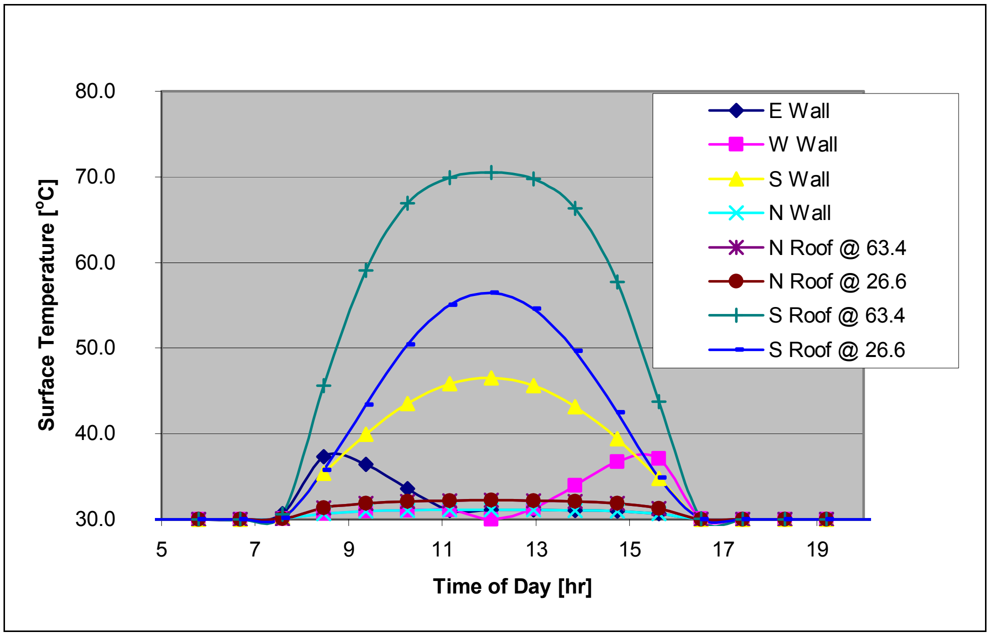
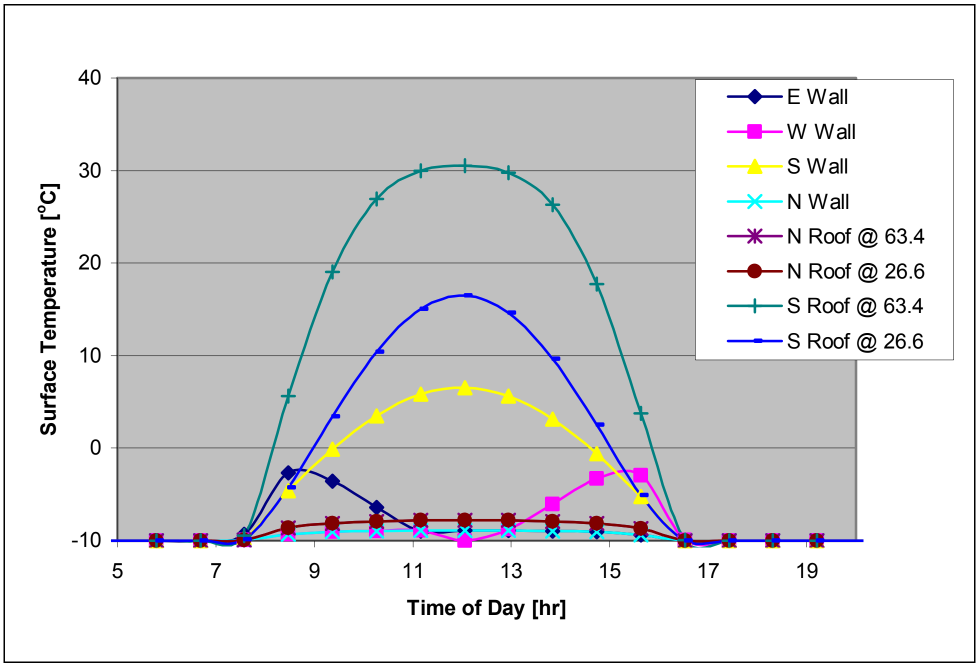


Figure G.1: Schematic representation of the horse barn and its orientation.



Graph G.1: Temperature of the barn's surfaces given an outside temperature of 30^{o}C and the building orientation shown in figure G.1



Graph G.2: Temperature of the barn's surfaces given an outside temperature of -10°C and the building orientation shown in Figure G.1

Appendix H: Design of a Passive Solar Heating Duct

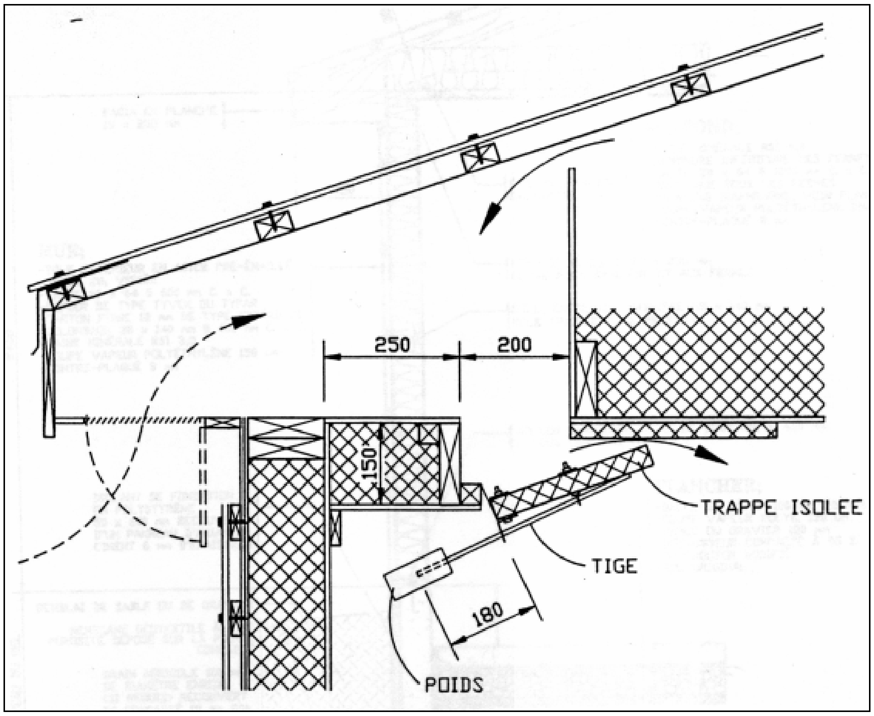


Figure H.1: Ventilation air inlet by the eave with counter-weight opening, for barn use.

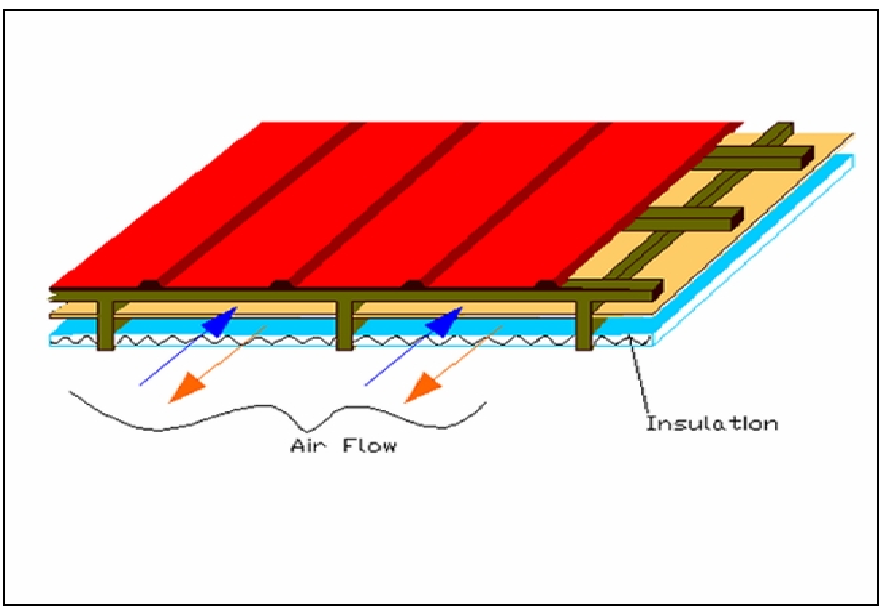


Figure H.2: Schematic representation of the passive solar heating duct system

This section describes a system to preheat the air coming in, simply by letting this latter passing right underneath the south roof at 63.4° of the gambrel roof barn. This roof section has a length of 3.4 meters. By simple heat transfer from the hot surface to the air in a rectangular duct underneath the surface, the air is heated as it passes all along the duct (see figure H.2, appendix H).

Calculations are based on winter ventilation flow rate calculated before.

Determination of the opening

From the equation: $Q = AC_d \sqrt{\dfrac{2 * P[Pa]}{\rho[kg/m^3]}}$

Where: Q is the air flow rate in the duct

A is the cross-sectional area of the duct

C_d is a coefficient to consider the loss in speed at the entrance

P is the pressure desired to run the ventilation system properly (usually 30 Pa)

ρ is the air density at outside temperature.

So:

$$A = \frac{Q}{C_d \sqrt{\dfrac{2 * P[Pa]}{\rho[kg/m^3]}}}$$

Where $C_d \sqrt{\dfrac{2 * P[Pa]}{\rho[kg/m^3]}}$ represents the air speed

$$A = \frac{120[l/s]}{0.5\sqrt{\dfrac{2 * 30[N/m^2]}{1.14[kg/m^3]}}} \times \frac{1[m^3]}{1000[l]}$$

$$A = 0.034m^2 \qquad \& \; Vair = 3.5[m/s]$$

For the number of openings, it is simply the latter total area relative to the air flow rate divided by the area of the duct which appeared to be reasonable at 2cmX57cm. This cross-sectional area fits between two roof trusses perfectly and has a height which yields to a good heat transfer from hot surface to cold air as demonstrated in those following calculations.

$$\#Openings \cong \frac{Atotal}{Aper\_inlet}$$

$$\#Openings \cong \frac{0.034m^2}{(0.02 \times 0.57)m^2}$$

$$\#Openings \cong 3inlets$$

Heat Transfer Calculations:

That is how we proceed to calculate heat transfer from the hot surface to the cold air:

1st: Determine the average temperature on the surface per day: Taking the average radiation on the surface at 63.4° for each month and by linear interpolation, on the excel spreadsheet for incoming radiation at -10°C, determine the surface temperature over the entire day.
2nd: Calculate the exit air for that average surface temperature, as demonstrated on next page.
3rd: Find the time duration over which the radiation occurs for each month. It represents at the same time the percentage of light per day. (In addition the sun is at this radiance, 26% of the time relative to the hours of light in accordance to www.sunwize.com for Sherbrooke in winter)
4th: Calculate the percentage of time we do not need a heater during heating period.

e.g. for January:

Average radiation $_{\text{surf. @ 63.4°}}$ = 632.3 [W/m2]

$T_{\text{over surface @ 63.4°}}$ =13.2 [°C]

Air properties evaluated @ $T_{b\,mean} = \frac{1}{2}(T_{bi} + T_{bo}) \approx -5$°C and μ_w @ T_w
Where : T_{bi} is the air temperature coming in the duct
T_{bo} the air temperature exiting the duct
T_w the wall surface of the duct
(Kreith & Bohn, 2001)

c_p: 1010	J/kgK
ρ: 1.352	kg/m^3
k: 0.0223	W/mK
μ: 1.66E-5	Ns/m^2
Pr: 0.71	

Hydraulic diameter

$$D_H = \frac{4 \times flow.cross - section.area}{Wetted.perimeter}$$

$$D_H = \frac{4 \times 0.02m \times 0.57m}{2 \times (0.02m + 0.57m)}$$

$$D_H = 0.039m$$

Mass Flow rate

$$\overset{o}{m} = Q \times \rho$$

$$\overset{o}{m} = \frac{120[l/s]/1000[m^3/s]}{3inlets} \times 1.352[kg/m^3]$$

$\overset{o}{m} = 0.054[kg/s]$

Reynold's number is

$$R_{D_H} = \frac{\bar{U} \times \rho \times D_H}{\mu}$$

$$R_{D_H} = \frac{3.5[m/s] \times 1.352[kg/m^3] \times 0.039m}{1.66E-5[Ns/m^2]}$$

$R_{D_H} = 11{,}117 > 10{,}000$ so turbulent flow which will have a greater heat transfer.

Is the flow fully developed?

$$\left(\frac{x_{fd,h}}{D}\right) \cong 4.4(\mathrm{Re}_D)^{1/6}$$

$$\left(\frac{x_{fd,h}}{D}\right) \cong 4.4(11{,}117)^{1/6}$$

$\left(\frac{x_{fd,h}}{D}\right) \cong 20.79 > 10$ so we can assume it is fully developed.

Using Seider & Take correlation we are able to calculate the heat of air at the exit of its passage into the duct system.

from (Kreith & Bohn, 2001)

$$Nu_D = \frac{hD}{k_f}$$

$$Nu_D = 0.027\,\mathrm{Re}_D^{4/5}\,\mathrm{Pr}^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

$$Nu_D = 0.027 \times 11117^{4/5} \times 0.71^{1/3} \times \left(\frac{1.66E-5}{1.75E-5}\right)^{0.14}$$

$$Nu_D = 41.24$$

$$h = \frac{Nu_D k_f}{D}$$

$$h = \frac{41.24 \times 0.0223[W/mK]}{0.039[m]}$$

$$h = 23.58[W/m^2K]$$

$$q_{conv.total} = \overset{o}{m} c_p \left(T_{b_o} - T_{b_i}\right) = hA_s \Delta T_{LMTD}$$

Equation 1: $$\overset{o}{m}\, c_p\left(T_{b_o} - T_{b_i}\right) = hA_s\left(\frac{\left(Tw - T_{b_o}\right) - \left(Tw - T_{b_i}\right)}{\ln\left(\frac{\left(Tw - T_b\right)_o}{\left(Tw - T_b\right)_i}\right)}\right)$$

Using Maple Software, we are then able to evaluate T_{bo} for different T_w and T_{bi}
At -10°C, the exit temperature from the heating duct is 2°C, which back to Appendix 6, table 6.1.2, at that temperature do not need any heater. (In reality, at that temperature, an adjustment to the Q_s should be made, because the formula:

$$Q_s = \frac{\{(q_s[w/kg] \times \#_{animal} \times mass[kg] \times 1[kW]/1000[W]) - (Q_{loss}[kW/°C]) \times (T_{in} - T_{out})[°C]\} \times 1000[l/m^3]}{1.006[kJ/kg°C] \times (T_{in} - T_{out})[°C] \times \rho_{in}[kg/m^3]}$$

the T_{out} in the denominator must be changed to the exit temperature. Considering this aspect, it still gives us the same answer, we do not need a heater at the exit air temperature from the duct.)

Energy savings over the entire year in terms of heating requirements

(Coldest month temperatuer at Sherbrooke taken from ASHREA Fundamentals 2001)

Because when the sun is shinning, we do not need a heater, we just based the calculation on the percentage of time the sun is present. It has been calculated as follow:
Light time = 8.9h => 37% of day time (x 26%) ≈ 10%

$$\%light\_time = \frac{8.9h}{24h} = 37\%$$ 26% comes from: www.sunwize.com (in winter)

Conclusion
For January, which we take as the approximation for December and February also (remember, in adjacent month, we do not need a heater): 10% of the time we save energy without any heat requirement. Based on a average temperature per month at that time of -10°C (ASHREA, Fundamental, 2001) the barn need heating requirement generally at an average of 2210kWh, subtracting the energy saving. Down to a total of 1989 kWh, but for the calculations of the appropriate wind turbine, we have not consider this energy savings.

Appendix J: Electrical System Calculations

The capacity required for the convenience outlet circuit is based upon the assumption that a single circular saw, operated for a cumulative of 30 minutes per day every day. The standard energy consumption formula applies:

$$\begin{aligned} W &= P \times t \\ &= 1.8kW \times \left(0.5\, h/day \times 365 days\right) \\ &= \underline{329kWh} \end{aligned}$$

Voltages for the building were chosen arbitrarily and correspond with North American norms for a three-phase entrance. Voltage requirements per circuit were determined by the design values of the equipment to be used. Amperage use on each circuit is simply the quotient of the peak power demand and the circuit RMS voltage, for example:

$$A_{outlet} = \frac{W}{V} = \frac{1800W}{120VAC/\sqrt{2}} = \underline{22A}$$

The breaker selected for each circuit exceeded the demand by at least 20% (i.e. with a safety factor of at least 1.25).

$$\frac{A_{outlet}}{A_{breaker}} = \frac{22A}{30A} = \underline{0.733 < 0.8}$$

The electrical entrance to the building is then determined by the largest voltage requirement, 240VAC in this case, and the sum of the breaker amperages:

$$\begin{aligned} A_{entrance} &= A_{light} + A_{outlet} + A_{HV} \\ &= 10A + 30A + 80A = \underline{120A} \end{aligned}$$

Appendix K: Total Design Load Calculations

In order to get a better idea of the annual distribution of energy consumption in the system, the year was split into a winter section, comprising December through February (90 days), and the rest (275 days). The general and water supply loads are simply split on the basis of the above fractions of 365 days. The lighting load also has a differential seasonal use component factored in (9 h/day in the winter, 6 h/day during the rest). Heating is only used during the winter, and ventilation rates change with the seasons. Their energy requirements are summed up seasonally and annually.

The peak power demand is calculated based on everything operating simultaneously. From table 4 in the *Electrical System* section, the calculation is:

$$\begin{aligned} P_{\max} &= \sum P_{\max,i} = P_{light} + P_{outlet} + P_{HV} + P_{pump} \\ &= (0.311 + 1.8 + 10 + 0.560)kW = \underline{12.7kW} \end{aligned}$$

Appendix L: Wind Resource Calculations

Table L.1: Monthly atmospheric and wind data for the Sherbrooke A weather station from RETScreen® International 2000

Month	Monthly average temperature (°C)	Monthly average relative humidity (%)	Monthly average wind speed at 10m (m/s)
January	-11.6	72.0	3.1
February	-10.4	68.0	3.1
March	-4.0	67.0	3.3
April	3.9	64.5	3.3
May	10.8	64.5	2.8
June	15.5	71.0	2.5
July	18.0	73.5	2.2
August	16.6	77.0	2.2
September	12.0	77.0	2.2
October	6.4	79.0	2.8
November	-0.1	78.0	3.1
December	-8.3	76.5	2.8
Average	**4.1**	**72.3**	**2.8**

Since the above data is for an elevation of 10m, it must be adjusted for a more accurate estimate of the true wind profile at 30m. This is most commonly done with the power law for wind shear:

$$u = u_o \times \left(\frac{z}{z_o} \right)^{\alpha}$$

Where u is the wind speed at the desired elevation, z,
u_o and z_o are the reference wind speed and elevation,
and α is the power law coefficient.

For most terrains, the value of α falls between 0.1 and 0.32. Being surrounded by tree tops, the recommended value for α is 0.25. For the Sherbrooke data, the formula reduces to:

$$u_{30m} = 2.8\,m/s \times \left(\frac{30m}{10m} \right)^{0.25} = \underline{3.7\,m/s}$$

One of the most important determinants for energy generation is the distribution of wind speeds with time. This is done by the means of an adapted Weibull distribution where the shape factor, k, is set at 2. The name for this special distribution is the Rayleigh distribution, and it has the following form:

$$f = \frac{\pi}{2}\left(\frac{u}{\bar{u}^2}\right) \times \exp\left(-\frac{\pi}{4}\left(\frac{u^2}{\bar{u}^2}\right)\right)$$

Where f is the fractional frequency of occurrence of wind speed u [m/s] given long-term average wind speed $\bar{u}$ [m/s].

The only parameter in the distribution is the average wind speed, making it extremely easy to use. The distribution for the Sherbrooke A station is shown in Graph L.2.

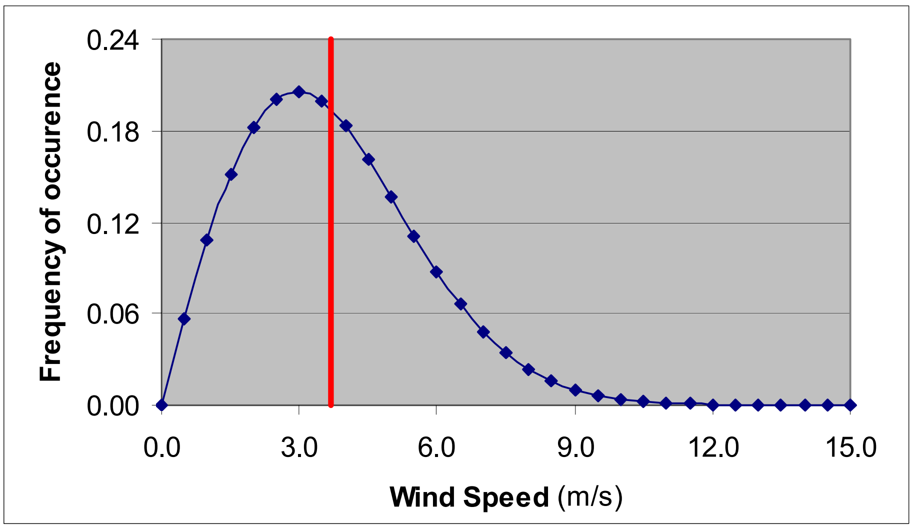
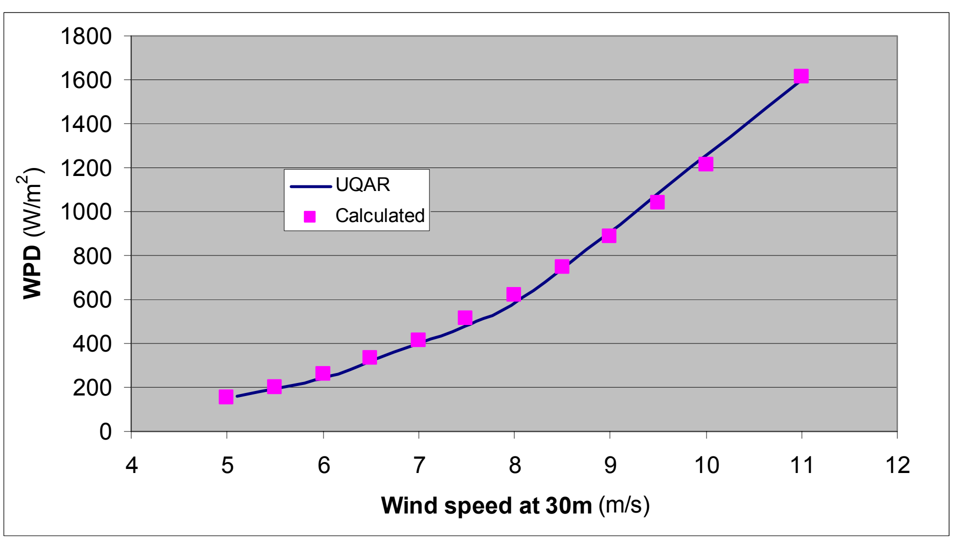


Figure L.2: Rayleigh Distribution for an average wind speed of 3.7 m/s

As far as accuracy is concerned with respect to our assessment of the wind potential on the site and the resulting power and energy figures, we closely match the data of the Université de Québec à Rimouski's study – the most extensive study of the province's wind potential to date, as illustrated in Graph L.3.



Graph L.3: Comparison of our calculated wind power densities at 30m with those found in the UQAR wind potential assessment (Ilinca et al., 2003)

Appendix M: Wind Power Calculations

Power

Wind power is used by transferring momentum from the moving air to a rotor connected to some sort of load. Dealing with momentum transfers involves mainly mass and speed. The power transfer equation for wind turbines is:

$$P_{WT} = c_p \times \frac{\rho}{2} \pi R^2 u^3$$

where P_{WT} is the rotor power generated [W],
c_p is the rotor power coefficient, ρ is the air density [kg/m^3],
R is the radius [m] of the swept area, and
u is the wind speed [m/s].

If the power coefficient is removed, the equation reduces to the power inherent in the wind. The power coefficient is therefore a direct measure of the rotors ability to capture wind power. The maximum theoretical value for c_p, called the *Betz factor*, is 0.593. In practice, the power coefficient is currently limited to a maximum value near 0.45 (Valtchev et al., 2000), and most small rotors achieve roughly $c_p = 0.2$ (Gipe, 1999).

Air density was extrapolated from a psychrometric chart based on the relative humidity and temperature values of the Sherbrooke site.

Radius was selected on the basis of the Bergey Excel 10kW wind turbine design. It was set at 3.5m for the entire analysis.

Wind speed variation has the largest impact on rotor power, but can only be controlled by proper site selection.

Energy and power density

Having calculated rotor power and the distribution of wind speeds with time, it is possible to calculate the energy generated by the wind turbine over any given period. The general formula for a given wind speed has the form:

$$W_i = f_i T P_i (\Delta u) \times e$$

Where W_i is energy generated at a given wind speed [kWh],
f_i is the fractional frequency of occurrence of that wind speed [per m/s],
T is the duration of the time period [h], P_i is rotor power at that wind speed [kW],
Δu is the step size between velocities analyzed [m/s], and
e is the fractional system efficiency.

Since each *f*-value represents a range of velocities, the size of the range must be included in order to provide the correct sum of probabilities. The total energy generated for a given period of time is the sum of all non-zero W_i-values. The following table gives an example of the calculation performed in a spreadsheet:

Table M.1: Spreadsheet calculation of annual energy generation for an average wind speed of 4.2m/s

Wind Speed (m/s)	**Step Size**	**Frequency** (0.0 - 1.0)	**Turbine Power** (W)	**Usable Energy** (kWh)
0.5	0.5	0.044	0.0	0.0
1.0	0.5	0.085	0.0	0.0
1.5	0.5	0.121	0.0	0.0
2.0	0.5	0.149	0.0	0.0
2.5	0.5	0.169	0.0	0.0
3.0	0.5	0.179	131.6	63.5
3.5	0.5	0.181	208.9	101.8
4.0	0.5	0.175	311.8	147.0
4.5	0.5	0.163	444.0	194.9
…	…	…	…	…
11.5	0.5	0.003	7410.5	56.8
12.0	0.5	0.002	8419.8	39.9
12.5	0.5	0.001	9516.7	27.2
13.0	0.5	0.001	10705.0	18.0
13.5	0.5	0.000	11988.3	11.6
14.0	0.5	0.000	0.0	0.0
14.5	0.5	0.000	0.0	0.0
15.0	0.5	0.000	0.0	0.0
			Total (kWh)	**3646.6**

The *Usable Energy* column hides the calculation of the time period and inclusion of the efficiency term. The time period is simply:

$$T_{year} = 24\,h/day \times 365\,days/year = \underline{8760\,h/year}$$

The system efficiency used was determined by the following table, where the total efficiency is the product of the others:

Table M.2: System efficiencies

Component	**Efficiency**
Battery	0.850
Inverter	0.900
Regulators	0.980
Wiring	0.980
Dumped Energy	0.950
Safety Margin	0.900
Total	**0.628**

These efficiencies are standard values from Bergey WindPower's "Predicting Performance and Designing Systems" document.

Table M.1 has an interesting feature; it includes consideration for the two design speeds. The cut-in speed is the wind speed at which the rotor will begin turning. The furling speed is the minimum speed at which the rotor will feather itself out of the wind to avoid overspeed damages. Beyond both of these limits, the rotor does not generate power or capture energy (as seen in table M.1).

Aside from reducing overall energy capture, these design speeds, and especially the cut-in speed, affect the amount of battery storage that needs to be supplied in order to satisfy the energy demand. Since most of the time the wind speeds are small, increasing the cut-in speed means that the wind turbine will be operating less often and that, correspondingly, the system will have to rely on stored power more often. Also, a lower design wind speed with respect to the cut-in speed (i.e. the cut-in speed is relatively close to the average wind speed) also increases the need for battery storage. This effect was seen in the *Wind Turbine Design Summary* section.

Battery storage was calculated based on the month with the longest outage time and an assumption that the risk of the entire outage being continuous was 50%. Thus the greatest monthly outage power demand was divided by two to give the battery storage. The results are in Table M.3:

Table M.3: Battery storage calculation

Max. monthly winter outage	418.4	h
Average winter consumption	1.186	kWh/h
Energy used during outage	496.3	kWh
Max. monthly non-winter outage	615.6	h
Average consumption	0.174	kWh/h
Energy used during outage	107.2	kWh
Risk of a single outage	0.5	
Worst case	Winter	
Battery storage needed	**248**	**kWh**