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Utilization of wind energy in space heating and cooling with hybrid HVAC systems and heat pumps

İbrahim B. Kilkis *

Geoenergy International Consortium, 3131, West Chestnut Expwy, Springfield, MO 65802, USA

Abstract

Approximately one-third of the primary energy resources are consumed in space heating, cooling, and air-conditioning with a very low exergetic efficiency. The depleting nature of primary energy resources, negative environmental impact of fossil fuels and low exergetic efficiencies obtained in conventional space heating and cooling are the main incentives for developing alternative heating, ventilating, and air-conditioning (HVAC) techniques which can employ low density and interrupted energy sources. In this respect, in spite of difficulties primarily encountered in coupling wind energy with conventional space heating and cooling equipment, wind energy seems to be an exciting alternative provided that synectic combinations are pursued and applied. In this paper, a new wind turbine coupled hybrid HVAC system is presented, which consists of an optimum combination of convective and radiant heating and cooling systems with in-space thermal energy storage. A design case for a single family home is presented. In this study a 6 kW(e) wind turbine drives a ground source heat pump (GSHP) which is coupled to a hybrid HVAC system to satisfy the thermal loads of a 100 m^2 home. In this example, sensible heating and cooling loads are satisfied by the high mass radiant floor which matches the daily peak demand and the available peak wind energy. Latent heating and cooling loads, along with ventilation requirements are satisfied by a forced-air system. Variable radiant and convective split type of control is implemented, and both systems are served by the same GSHP which also satisfies the domestic hot water (DHW) demand.

Keywords: Hybrid panel system; Heat pump; Wind energy

1. Introduction

Direct utilization of primary energy resources in conventional heating, ventilating, and air-conditioning (HVAC) systems without any cogeneration has a very low exergetic efficiency. This phenomenon is displayed for space heating in Fig. 1, where space heating exhibits the biggest difference in energy and exergy efficiencies compared to other energy-consuming sectors [1].

Energy and exergy efficiencies for principal types of processes are defined as follows [1]:

1. Energy efficiency: energy in products/total energy input
2. Exergy efficiency: exergy in products/total exergy input

Exergy is the expression for loss of available energy due to the creation of entropy in irreversible systems or processes. The exergy loss in a system or component is determined by multiplying the absolute temperature of the surroundings by the entropy increase. Entropy is the ratio of the heat absorbed by a substance to the absolute temperature at which it was added. While energy is conserved, exergy is accumulated.

In order to decrease the difference between the energy and exergy efficiencies, the rational use of primary energy resources calls for a district heating and cooling system coupled to cogeneration or trigeneration plants. A trigeneration plant has one more cascade of energy utilization as

Abbreviations: ASHRAE, American Society of Heating, Refrigerating, and Air-conditioning Engineers; AUST, average unheated surface temperature of the room enclosure; COP, coefficient of performance; DHW, domestic hot water; GSHP, ground source heat pump; HVAC, heating, ventilating, and air-conditioning; MOH, merit of hybrid number; PR, ratio of the sensible heating or cooling load which is assigned to a radiant panel; TES, thermal energy storage system

* Tel.: +1-417-8646108; Fax: +1-417-8648161.

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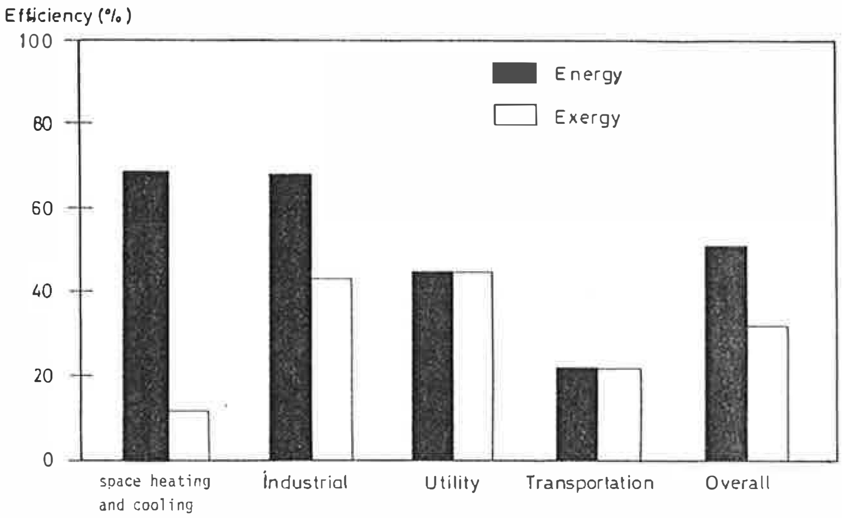

Fig. 1. Energy and exergy efficiencies in typical sectors [1].

compared to cogeneration. However, the cost-intensive nature of district heating and cooling may not be suitable for districts with low demand density (MW(t)/km^2). These arguments also hold true for centralized distribution of natural gas which is apparently cheap, abundant and widely used in space heating and now even in absorption cooling. With these difficulties in mind for low demand density districts or isolated buildings in remote areas, the difference between energy and exergy efficiencies may be decreased by the use of: (i) on-location alternative energy resources with low enthalpy and density, (ii) alternative HVAC equipment in order to establish an efficient and direct coupling with alternative energy resources by also taking into account of the interrupted supply feature of solar and wind energies.

Until now wind energy has not been considered viable for HVAC applications, except some electric heating applications in greenhouses and similar spaces. Unless the electricity generated by a wind turbine is used to drive a heat pump first, the heating coefficient of performance (COP) of the system in such applications will be less than one, and cooling will not be possible. Even when a heat pump is coupled to a wind turbine, the following problems exist for a conventional HVAC system like a convective (forced-air) system.

(i) Temperatures required by the conventional HVAC equipment at peak loads are generally too demanding for a heat pump,

(ii) Thermal storage is essential, because thermal loads can only be satisfied on-line (instantly).

(iii) Since the early eighties, the cost of wind-delivered energy has decreased considerably. Cost of wind energy today is around 6 cents per kW(e) h. Wind turbine prices are in the range of US$800 to US$1000 per installed kW(e) [2]. Although these figures are very promising, the competitiveness of wind energy systems are still marginal. Due to marginal advantages of a wind turbine system, the installation and operating costs must be distributed to 12 months as evenly as possible. This generally requires seasonal thermal storage. Overall savings in system cost are also very desirable.

In view of the above points, the attributes for an alternative system may be summarized as follows:

(i) Decrease thermal building loads, implement peak load shaving,

(ii) Decrease demand on required temperature levels by the HVAC system, thus increase the COP both in cooling and heating modes,

(iii) Shift daily peak thermal loads in order to match with peak wind energy,

(iv) Level summer and winter thermal loads by seasonal energy storage,

(v) Provide cheap (or free) domestic hot water (DHW),

(vi) Maximize the use of the building mass for short-term energy storage,

(vii) Minimize the installation and operating cost, and improve comfort control by decoupling latent and sensible thermal loads.

2. Hybrid panel HVAC systems

Several attributes of radiant panel heating and cooling were reported earlier by Kilkis [3]. Main attributes are the following.

(i) Sensible thermal loads are reduced about 20% in heating and about 10% in cooling. The main reason is that indoor air temperature may be less conservative for human comfort due to the fact that the operative temperature is

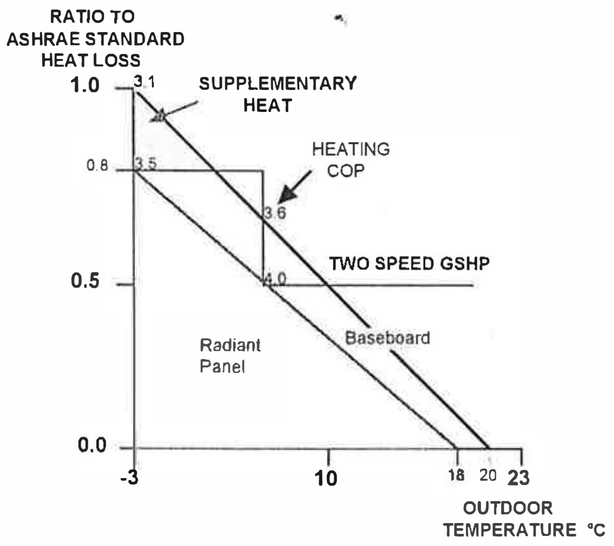

Fig. 2. Comparison of GSHP performance with radiant panel and baseboard heating systems.

improved by the radiant heat transfer component of radiant panels.

(ii) Moderate fluid temperatures in the hydronic system improve the COP of any heat pump connected to the system. In addition, periods of operation cut-off and/or system back-up during peak loads are minimized [3].

(iii) Definite zoning is possible.

(iv) Part of the building mass is utilized for thermal storage (i.e., floors, ceilings, walls). For example, peak load may be shifted in an hourly order of magnitude, depending upon the thermal mass of the panel and the enclosure walls.

Fig. 2 shows the performance enhancement of a heat pump in winter when coupled to a radiant panel system instead of a baseboard system. In this figure, a two-stage ground source heat pump (GSHP) is demonstrated. First of all, compared to standard American Society of Heating, Refrigerating, and Air-conditioning Engineers (ASHRAE) heat-loss calculations, the sensible heating load decreases by about 20%. Secondly, the indoor design temperature is 18°C instead of 20°C for equivalent human thermal comfort. These two factors shift the load–temperature line farther from the baseboard line. As a result, any heat pump sized for 80% of the standard design load will not require any supplementary heat during peak load periods. Also, due to the fact that the hydronic panel system supply temperature is lower than a baseboard system by as much as 35°C, the heat pump cut-off is eliminated during peak loads [3]. As shown in this figure the COP in heating will be 3.5 instead of 3.1 when a radiant panel system replaces a baseboard system. In the same respect, at 50% load, the COP will be 4.0 instead of 3.6. A similar argument holds true for sensible cooling, while a radiant panel can satisfy the sensible cooling loads at a moderate cold water supply temperature.

In spite of these advantages, indoor air quality and latent cooling loads can not be satisfied by radiant panels. These indoor air-conditioning functions may be satisfied by an undersized convective (forced-air) system. Generally such a function decoupling results in better indoor comfort, control and energy savings as will be shown in the design example.

A hybrid panel system can be operated by the same heat pump with a variable output split between air and hydronic delivery systems. DHW may be supplied in the same token.

3. Wind energy vs. building thermal loads

Fig. 3 shows a generalized comparison between the thermal space loads and wind speed for winter and summer months. In winter, unlike solar energy, wind energy is

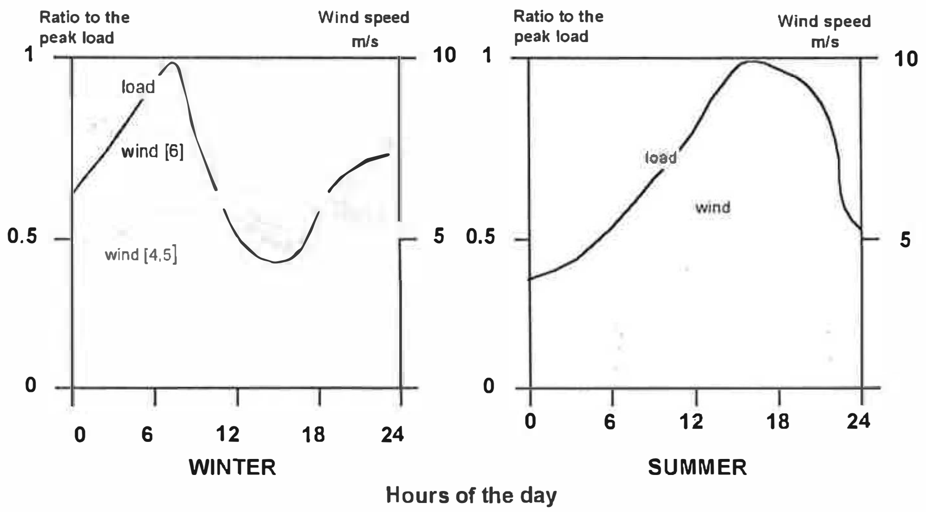

Fig. 3. Comparative daily variations of the thermal loads and typical wind speeds [4–6].

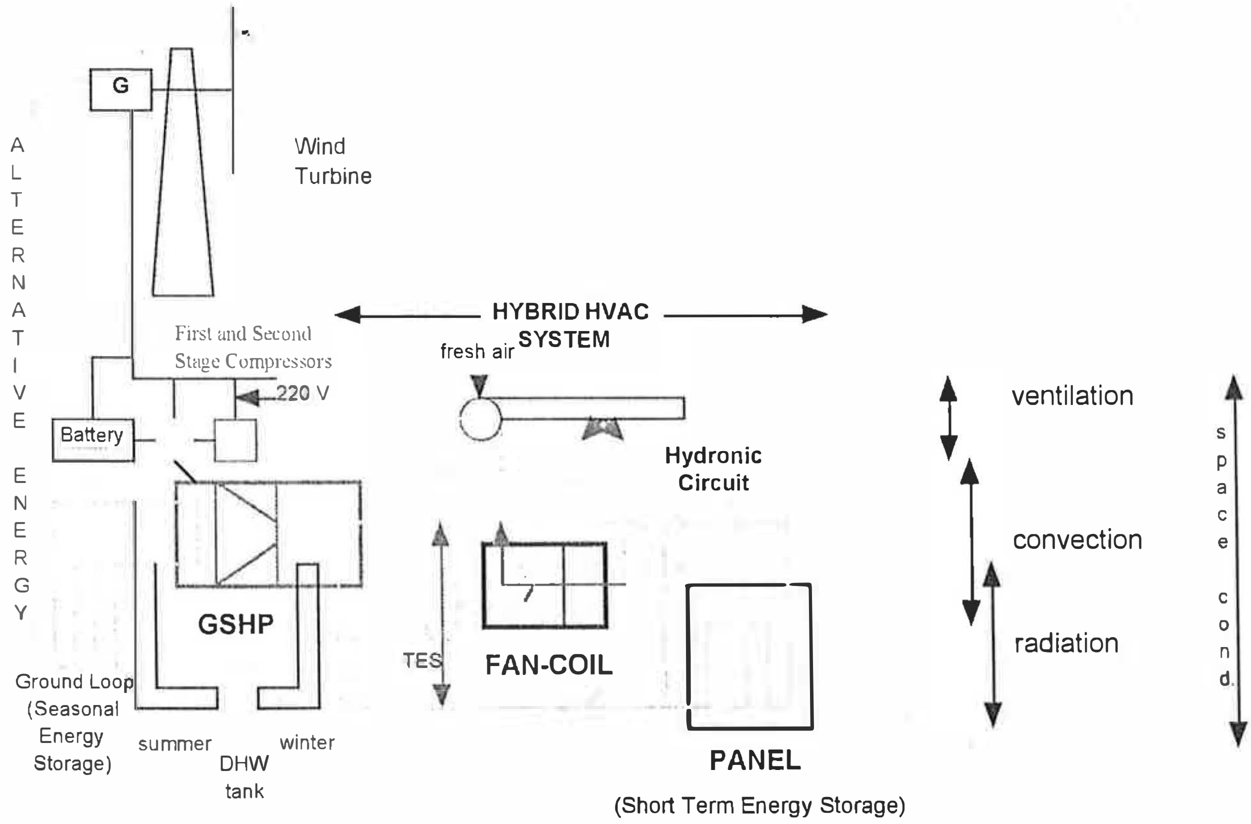

Fig. 4. Wind turbine driven GSHP coupling to a hybrid HVAC system [7].

more sustainable and steady, where sunshine is strongly intermittent in winter months. In many cases there may be almost 6 to 8 h of phase difference between the peaks of wind speed and the space heating load [4,5]. Yet in certain climatic conditions and geographic locations, peak wind speed may follow the space heating load more closely [6]. In any case wind energy seems to be more suitable when compared to solar energy. In summer, however there is a good correlation between the wind speed and sensible cooling loads.

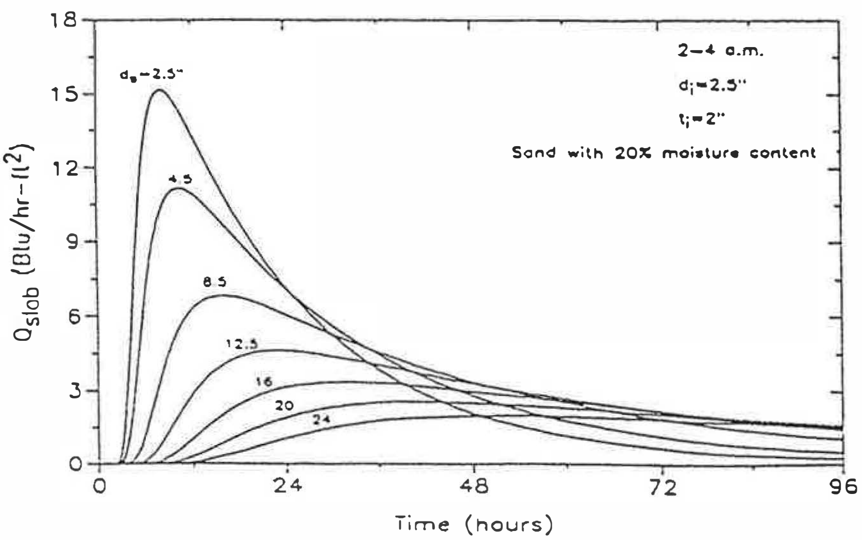

Fig. 5. Radiant panel heat flux as a function of slab thickness [8].

4. Wind energy coupled hybrid panel HVAC system

In order to optimally couple the attributes of hybrid panel HVAC systems and wind energy, the key element appears to be a GSHP which enables seasonal energy storage. Fig. 4 shows a synectic combination of a wind turbine, GSHP, short term thermal and electrical storage system, and a panel hybrid HVAC system [7]. In design, the word synectic means to combine known elements and/or systems in an unusual manner in order to innovate a new system. Wind turbine generates electricity which drives directly or from the electric storage unit (batteries) compressors of GSHP which are in tandem in order to better follow the thermal loads of the building. Wind turbine and the GSHP form the combined alternative energy system. GSHP has two types of HVAC interface: one is a hydronic interface for radiant panels, and the second one is the air interface for the undersized forced-air (convective) system. In winter fan coils which are generally sized for cooling loads in a two-pipe system may operate at reduced water temperatures like radiant panels. Therefore, heating COP is enhanced. Part of the heat is delivered to the system through the ground circuit. DHW is available on demand through the hydronic interface of the GSHP. In summer operation, latent loads are satisfied by the fan coil. Whenever the latent loads are negligible, fan coils are bypassed in order to enhance the cooling COP through the use of radiant panels which operate at higher cold water temperatures. Heat is rejected to the ground circuit in summer. Otherwise, DHW is supplied and stored in the DHW tank.

Preheating or precooling of fresh air for ventilation purposes is also possible. Thus the indoor human thermal comfort is achieved at three stages namely, ventilation, sensible conditioning, and latent conditioning. A water tank may be optional for thermal energy storage system (TES) both in winter and summer. The thermal mass of the radiant panel provides some degree of thermal storage too. Fig. 5 shows the heat flux curve against time for an electrically heated floor slab with 2.5 in. (63 mm) thick sand under layer and a 2 in. (51 mm) thick back insulation [8].

5. Case design

A wind turbine driven hybrid panel HVAC with a GSHP is to be used for a single-family home of 100 m^2 floor area. Floor heating and cooling will be accomplished in a 2.5 in. (63 mm) over pour concrete, with 2.5 in. (63 mm) sand under layer, and 2 in. (51 mm) thick back insulation. Thermoplastic piping will be used to transfer the heat to the slab from the hydronic circuit.

Design conditions are the following:

In winter	
Outdoor design temperature, t_o	− 3°C,
Indoor design temperature, t_a	18°C
Floor panel area, A_p	80 m^2
Floor panel thermal resistance, r_u	0.15 m^2 K/W
Floor concrete over pour	2.5 in. (6.3 cm)
Sensible panel heating load, q_u	80 W/m^2 (80% of standard load)
Peak load shaving factor without TES	0.8
Hydronic pipe spacing in the floor, M	15 cm
Area average temperature of unconditioned surfaces (AUST)	t_u (assumed)
Thermoplastic pipe inside diameter	19 mm
In summer	
Outdoor design temperatures	26°C WB/40°C DB
Indoor comfort conditions	26°C and 50 RH
Sensible panel cooling load, q_u	−40 W/m^2 (90% of standard load)
Peak load shaving factor with TES	0.6
Latent cooling load	1.5 kW
Area average temperature of unconditioned surfaces (AUST)	t_a

Although it seems reasonable that, sensible loads should be assigned to the sensible equipment only in a hybrid system, if there is ventilation, part of the sensible load may still be assigned to a forced-air system such that ventilation flow rate is not exceeded. Also, in order to establish some degree of back up, fan coils may also assume most of the sensible heating, and some part of the sensible cooling loads. This is in fact a matter of optimization for operation and installation costs. These considerations were plotted on a merit of hybrid system (MOH) diagram, as shown in Fig. 6. This is a plot of specific power consumption vs. PR. PR is the ratio of the sensible load assigned to the radiant panel to the total sensible load [9]. This figure reveals that a PR value of 0.6 seems to hold true for an optimum both for cooling and heating. There are several constraints on this diagram, like surface condensation of the panel, floor surface temperature limit, and required thermal response-control sensitivity.

Using this ratio ($q_u = 0.6 \times 80$ W/m$^2 = 48$ W/m^2), the hydronic circuit for radiant panel was designed for winter first: the radiant panel requires a mean water temperature of 31°C (18° + 13°C from Fig. 7). At a supply–re-

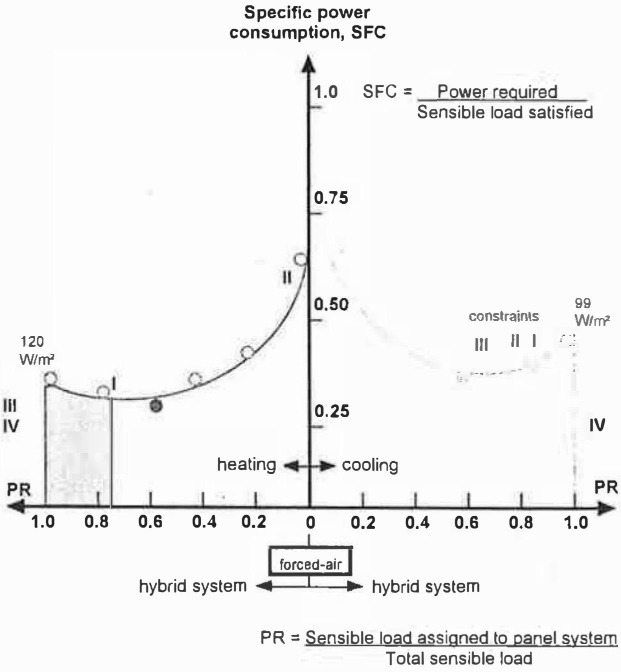

Fig. 6. Merit of hybrid systems (MOH) diagram for case design.

turn water-temperature difference of 10°C, the supply water temperature will be 36°C, which is very suitable for GSHP. However, this temperature is too low for fan coils if operated parallel with radiant panels, unless they are oversized, which is undesirable. Therefore, the supply temperature was increased to 55°C for fan coils, and after a temperature drop of 20°C at design conditions, the return temperature directly goes to the radiant panel system. The corresponding warm floor temperature is 23°C. This satisfies the recommended upper limit of 29°C. Fan-coil winter design load is 32 W/m^2. In winter operation the concrete and sand under layer provide a peak load shifting period of up to 7 h (see Fig. 5).

In summer, q_u is -24 W/m^2 (40 W/m^2 $\times$ 0.60). The required mean cold water (or brine) temperature is 18°C (26°C–8°C from Fig. 7). At a supply–return cold water temperature difference of 5°C, the supply temperature will be about 15°C. The corresponding cooled floor surface temperature is around 23°C. This is well above the dew point temperature. For summer operation, fan coils will be sized according to a sensible load of 1.6 kW and a latent load of 1.5 kW. Fan coils will operate at 8–13°C operation regime. When dehumidification is not required, the fan

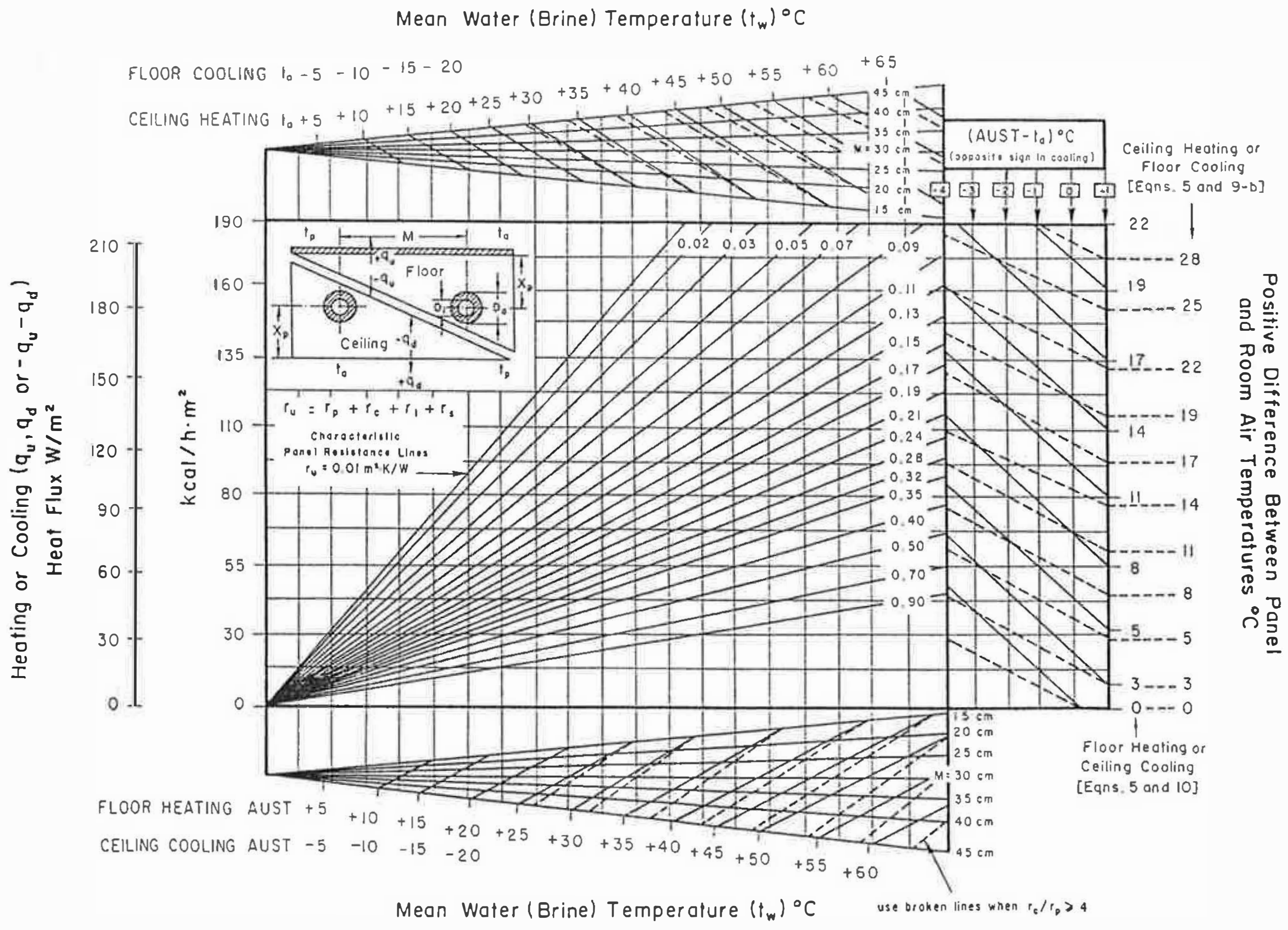

Fig. 7. Radiant panel design nomograph for heating and cooling [10,11].

coil is by-passed in order to increase the cooling COP by switching to 15–18°C from 8–13°C regime. Minimum COP is 0.7.

Other features of the design are: At 55°C supply temperature in winter, the corresponding COP is about 3.5. Therefore the total electric power requirement of the GSHP is around 1.46 kW(e) (6.4 kW total heating load × 0.8/3.5). Here 0.8 is the peak load shaving factor without TES tank in winter. In summer, with latent and sensible cooling loads (3.2 kW(e) × 0.6 + 1.5 kW(e))/0.7, the heat pump requires about 4.9 kW(e) in cooling. Including other parasitic losses, a 6 kW(e) wind turbine was selected for this design. A 7 kW(e) h electric energy storage system is an option. The system may also deliver electrical power to the household at off-peak thermal load periods.

6. Conclusions

A hybrid panel HVAC seems to be a viable solution in order to use wind energy in space conditioning. However, the key issue is the balance between winter and summer loads on the GSHP. In the above example, the wind turbine size is governed by the cooling loads. In summer, the turbine power requirement is 3-fold of the winter requirement. If electrical power demand is not an issue for the household, this represents a surplus capacity in winter. However, electric power is almost all ways on demand in rural or isolated areas. Therefore, it can be generalized that such a system will be more successful in climatic regions where winter and summer loads are similar in magnitude, and electrical power demand exists in the household.

References

[1] M.A. Rosen, I. Dincer, Energy and exergy analyses of sectoral energy utilization: an application for Turkey, Proc. First Trabzon Int. Energy and Environmental Symp. July 29–31, 1996, Vol. 3, pp. 1059–1065.

[2] Danish Energy Agency, Statement of the development in wind turbine technology and economics, Danish Ministry of Energy, Denmark, 1993.

[3] İ.B. Kilkis, Advantages of combining heat pumps with radiant panel heating and cooling systems, Newsletter, IEA Heat Pump Center, Vol. 11, No. 4, Holland, 1993, pp. 28–31.

[4] S. Pashardes, C. Christofides, Statistical analysis of wind speed and direction in Cyprus, Solar Energy, Vol. 55, No. 5, Elsevier, UK, 1995, pp. 405–414.

[5] H. Nfaoi, J. Buret, A.A.M. Sayigh, Stochastic simulation of hourly average wind speed sequences in Tangiers (Morocco), Solar Energy, Vol. 56, No. 3, Elsevier, UK, 1996, pp. 301–314.

[6] C. Unver, Determination of wind energy potential in Nurdagi (Turkiye) (in Turkish), Int. Energy, Electrical, Electronic and Automation Periodical, No. 85 (95-4), Kaynak Yayin, Istanbul, Turkey, 1995, pp. 102–108.

[7] İ.B. Kilkis, Panel heating and cooling with wind energy, In Turkish, Int. Energy, Electrical, Electronic and Automation Periodical, No. 85 (95-4), Kaynak Yayin, Istanbul, Turkey, 1995, pp. 138–143.

[8] H.B. Ramadan, Analysis of an underground electric heating system with short term energy storage, ASHRAE Transactions, paper no. 3784, Atlanta, USA, 1994.

[9] İ.B. Kilkis, S.R. Suntur, M. Sapci, Hybrid HVAC systems when should the sensible, latent and other functions of an HVAC system be decoupled? ASHRAE Journal, Vol. 37, No. 12, Atlanta, USA, 1995, pp. 23–28.

[10] İ.B. Kilkis, COOLP: a computer program for the design and analysis of ceiling cooling panels, ASHRAE Transactions, Vol. 101, Part 2, Atlanta, USA, 1996, pp. 703–710.

[11] ASHRAE Handbook, HVAC Systems and Equipment, Published by the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, USA, 1996.