USE OF DOE-2 TO EVALUATE EVAPORATIVE COOLING IN TEXAS CORRECTIONAL FACILITIES

Namir Saman, Ph.D., P.E.
Visiting Assistant Professor
Energy Systems Laboratory
College Station, Texas

Tarek Bou-Saada
Research Associate
Energy Systems Laboratory
College Station, Texas

Tia Heneghan
Energy Manager
Texas Department of Criminal Justice
Hantsville, Texas

ABSTRACT

This study investigates the feasibility of using direct and indirect evaporative cooling systems for correctional facilities in two different Texas climatic regions with the DOE-2.1E hourly energy simulation program. The analysis is based on adding user defined functions to the DOE-2 SYSTEMS sub-program to simulate direct and indirect evaporative cooling configurations. The DOE-2 program was run with two weather tapes, one for Kingsville, Texas and one for Abilene, Texas during April, July, and October to resemble neutral, summer and winter weather conditions. The results showed that direct evaporative cooling is applicable in April for Abilene and October for Kingsville. The indirect evaporative cooling is feasible in July for Abilene and April for Kingsville.

INTRODUCTION

Evaporative cooling provides an energy efficient, environmentally safe, and cost effective alternative to traditional space conditioning methods for residential homes, industrial facilities, and commercial and institutional buildings. Evaporative cooling may also be used to control dry-bulb temperature and/or humidity levels. Furthermore, it can be combined with mechanical refrigeration to enhance space comfort conditions, or simply as a mechanical refrigeration back-up system. Today's increased attention to rising energy costs, energy conservation issues, indoor air quality questions, and environmental CFC concerns has led engineers to focus on direct and indirect evaporative cooling systems.

The humidification and cooling energy savings of evaporative cooling as compared to conventional economizer cycles were studied by Brown (1989). It was found that when evaporative coolers are employed in conjunction with economizer cycle operation, humidification by evaporation can be achieved for most of the operating hours without expenditure of energy (Brown, 1989). In addition, the use of evaporative cooling is not solely restricted to arid climates. Evaporative cooling can be used over a broad geographical region to provide a quality indoor environment at a reduced annual energy consumption (Brown, 1990). The use of evaporative cooling in the summer to reduce peak cooling loads and cooling ton-hours as well as in the winter to increase building humidification leads to a short payback period (Scofield and Sterling, 1992).

Accordingly, evaporative cooling can be considered as an energy conservation approach that can be used individually or in combination with other cooling systems to save energy and improve the environment. It is not the purpose of this study to consider all evaporative cooling applications for the correctional facilities in Texas. Rather, the purpose is to investigate the feasibility of using DOE-2 as a tool to investigate the possibility of using evaporative cooling to produce a comfortable indoor environment for correctional facilities. Modeling evaporative cooling systems using hourly energy simulation programs makes it possible to analyze a variety of direct, indirect, and multiple-stage evaporative cooling systems (McDonald et al. 1990). The technique described here will enable the user to modify, enhance, or replace the DOE-2 calculations without having to recompile the program. The procedure includes using direct and indirect user-defined functions in the DOE-2 SYSTEMS subprogram together with two different weather conditions in Texas (Kingsville and Abilene).

BACKGROUND

This section introduces direct and indirect evaporative cooling concepts and how they can be used for cooling and humidification processes. Evaporative coolers exchange sensible heat for latent heat (ASHRAE, 1992). In the direct evaporative cooler, water is evaporated into an airstream, and the continuously recirculated water will reach an equilibrium temperature that approaches the entering air wet-bulb temperature. For direct evaporative cooling as shown in Fig. 1, the process of cooling and humidifying the air takes place at constant total heat (i.e. adiabatic process), which nearly coincide with the wet-bulb temperature lines on the psychrometric chart (ASHRAE-Applications, 1995). The effectiveness of an evaporative cooler is defined as the depression in dry-bulb temperature of the air leaving the evaporative cooler divided by the difference between the dry bulb and wet bulb temperatures of the incoming air (ASHRAE, 1995). Evaporative cooling effectiveness can range between 80% to 90%. The indirect evaporative cooling process is also shown in Fig. 1. The cooling process is sensible since there is no direct contact between the supply air and the water.

In this study the direct evaporative cooling is used to cool the incoming outside air as demonstrated in Fig. 2. For the indirect evaporative cooling used in this analysis, and shown in Fig. 3, the evaporative cooler is used to cool the mixed air temperature using outside air.

A combination of indirect and direct evaporator cooling can be used in a system or any of the direct or indirect modes can be used with mechanical refrigeration. In addition, more than one stage evaporative cooling can also be used (ASHRAE, 1995).

EXISTING HEATING/VENTILATING SYSTEMS AT TDCJ

The existing heating and ventilating system at the Texas Department of Criminal Justice (TDCJ) prison facilities consists of a system that is similar to the DOE-2 HVSYS system in the cell buildings. The existing heating/ventilating system pulls outside air directly into the building. There is no means of conditioning the air with respect to cooling the air and controlling humidity. During the heating season, the outside air is pulled and heated using natural gas as the heating medium. In the summer, especially in hot or hot and humid climates, indoor conditions are outside ASHRAE recommended comfort zones.

Two locations in Texas were chosen for this study, namely Abilene and Kingsville. Abilene is located in the upper middle part of Texas with cold and dry weather in the fall, hot and humid weather in the summer and moderate temperature and humidity in the spring. Kingsville is located in the south part of Texas. The weather is hot and humid during the summer and the spring, and with moderate temperature but humid in the winter.

DOE-2 MODELS

The Functional Value features in the DOE-2 LOADS and SYSTEMS sub-programs allow the user to modify the calculations without recompiling the program (DOE-2 manual, 1992). The FORTRAN-like language models employ modifications to the direct and indirect evaporative cooling models developed by McDonald et al. (1990).

The evaporative cooling models are implemented at the air economizer subroutine within the DOE-2 program labeled ECONO-2. The modified ECONO-2 INDIRECT subroutine includes algorithms to calculate the drop in supply air dry-bulb temperature as a result of using indirect evaporative cooling (McDonald et al., 1990). The DOE-2 simulation produced hourly zone temperature for Abilene and Kingsville using weather conditions for April 30, July 31, and October 31. The analysis was carried out with and without the use of evaporative cooling. The format of the model for the indirect evaporative cooling is shown in the Appendix.

The modified ECONO-2 DIRECT subroutine includes an algorithm that calculates the temperature drop and humidity change as a result of using the direct evaporative cooling in the outside air intake (see Appendix). A fixed value for evaporative cooling effectiveness is used. The psychrometric equations used are based on ASHRAE humidity calculation algorithms (ASHRAE 1993) and McDonald et al. (1990).

Two TMY weather tapes were used for the DOE-2 simulation (Kingsville and Abilene). The DOE-2 input file of a typical TDCJ 2,250-bed cell building was run with evaporative cooling and with existing heating and ventilating systems simulated with HVSYS in DOE-2.

The direct and indirect evaporative cooling models were run together with the no evaporative cooling model. The hourly room temperature and relative humidity for April 30, July 31, and October 31 were determined and plotted on the ASHRAE comfort region psychrometric chart (similar to Fig. 11) to determine the comfort level of using evaporative cooling for both locations.

RESULTS

Figures 4 through 14 summarize the results of this work. Figures 4 and 5 show the variation of the difference between the dry-bulb and wet-bulb temperatures for Kingsville and Abilene for April 30, July 31 and October 31, respectively. It was expected that the higher temperature difference would produce the best performance with direct evaporative cooling. However, the high outside dry-bulb temperature and relative humidity during July and April for Kingsville left only October as the suitable direct evaporative cooling month of the three. In Abilene, the high temperature and humidity in July, and the low temperature in October made April to be the suitable direct evaporative cooling month.

Figures 6 and 7 show the variation in the supply air temperatures for Kingsville and Abilene, respectively. Use of the evaporative cooling lowered the supply air temperatures in both locations. Supply air temperatures of less than 68° F was considered for the analysis (McDonald et al., 1990). For Kingsville, shown in Fig. 6, both April and July supply air temperature values were above the 68° F level. October conditions produced supply air temperatures below 68° F. For Abilene, shown in Fig. 7, April proved to be the month with the most suitable supply air temperatures. July supply air temperatures were higher than 68° F and October supply air temperatures were too low for comfort cooling.

Figures 8 and 9 show the variation in room air temperature and humidity ratio, respectively, for Kingsville and Abilene, with October shown in Fig. 8 and April shown in Fig. 9. Both direct evaporative cooling and no evaporative cooling were considered. The combination of temperature and humidity ratio was plotted on the psychrometric chart to compare with ASHRAE's comfort conditions as shown in Fig. 10. The combination of room dry-bulb temperature and humidity ratio fell outside the standard ASHRAE comfort zones. However, the combination fell inside the ASHRAE comfort zones with higher air velocities accompanying evaporative coolers, as demonstrated in Fig. 11.

For the indirect evaporative cooling mode, the supply and room air temperatures for Abilene and Kingsville are shown in Figs. 12 and 13, respectively. The same discussions mentioned above concerning ASHRAE comfort zones applies to indirect evaporative cooling room dry-bulb/humidity ratio combination. In the indirect evaporative cooling mode, no humidification of the supply air takes place. The humidity of the outside air or the mixture of outside and return air will be the humidity of the supply air. For both Abilene and Kingsville, the indirect evaporative cooling reduced room air temperature by 5 to 10° F.

DISCUSSION

The use of direct and indirect evaporative cooling was investigated for three days in three different months for two locations in Texas (Abilene located in upper central Texas, and Kingsville located in southern Texas). The results showed that indirect evaporative cooling is feasible in July for Abilene and in April for Kingsville with room temperatures of 5 to 10° F lower than the case of no evaporative cooling. Direct evaporative cooling is feasible in April for Abilene, and October for Kingsville. The high dry-bulb temperature and relative humidity in Kingsville made October suitable for direct evaporative cooling. The humid weather in April and July led to an inappropriate humidity/dry-bulb temperature combination that was rendered ineffective by the humidification effect of the direct evaporative cooling process. For Abilene, October temperatures are too low and thus not suitable for direct evaporative cooling. On the other hand, July temperatures were high enough to result in supply air temperatures of greater than 68° F. Since there is no direct contact between the supply air and the water for the indirect evaporative cooling, there will be no humidification of the supply air. For Abilene, July was the month with moderate outside temperature that resulted in suitable supply air temperatures. For Kingsville, April was the suitable month.

The performance of the direct and indirect evaporative cooling process can be modified by using chilled water in an evaporative cooling/chiller system that will supply better indoor conditions. This issue will be addressed in a future study. Figure 14 shows the regions on the psychrometric chart where evaporative cooling and evaporative cooling plus mechanical refrigeration systems can be used (reference 9).

CONCLUSIONS

DOE-2 is a useful tool to study the feasibility of using a evaporative cooling process in different regions of the country once weather conditions and building construction information are available. The subroutines in DOE-2 may be modified to accommodate the use of special routines that model the performance of direct and indirect evaporative cooling processes. The use of evaporative cooling during certain times of the year will lead to improved indoor comfort conditions in correctional facilities in Texas. This study is preliminary and will be expanded in a future study to include more regions in Texas as well as yearly simulation with the DOE-2 program. The following is a summary of the conclusions of this work.

1- Although large differences between the dry-bulb and wet-bulb temperatures existed for both locations, the high dry-bulb temperature values led to unsuitably high supply air temperatures. This was especially noticeable for Kingsville in July.

2- For cold weather, using an evaporative cooling process is not appropriate (October in Abilene).

3- The ASHRAE comfort zones are affected by supply air velocity. Therefore, the combination of humidity ratio and dry-bulb temperature resulting from using the direct and indirect evaporative cooling processes fell inside the "moved" comfort zone. This was the case of the indirect evaporative cooling in July for Abilene and April Kingsville. The same was true for the direct evaporative cooling in April for Abilene and October for Kingsville.

4- The regions appropriate for using evaporative cooling are shown in Fig. 14 (Reference 9).

ACKNOWLEDGMENTS

The authors would like to acknowledge Mr. Bryce Munger for preparing some of the figures used in this study. Professor Jeff Haberl also provided helpful review and suggestions.

REFERENCES

1. Brown, W.K., "Humidification by Evaporation for Control Simplicity and Energy Savings," ASHRAE TRANSACTIONS, V. 95, Pt. 1, 1989.

2. Brown, W.K., "Fundamental Concepts Integrating Evaporative Techniques in HVAC Systems," ASHRAE TRANSACTIONS, V. 96, Pt. 1, 1990.

3. McDonald, G.W., M.H. Turietta and R.E. Foster, "Modeling Evaporative Cooling Systems with DOE-2.1D," ASHRAE TRANSACTIONS, V. 96, Pt. 1, 1990.

4. Scofield, C.M. and E. Sterling, "Dry Climate Evaporative Cooling with Refrigeration Backup," ASHRAE Journal, June 1992.

5. DOE-2.1 D users Manual, 1992.

6. ASHRAE Handbook-HVAC Systems and Equipment, 1992.

7. ASHRAE Handbook-Fundamentals, 1993.

8. ASHRAE Handbook-HVAC Applications, 1995.

9. Private communications with Mr. Mehrad Sadrisabet, President, Professional Engineering & Technology Engineering Firm, Tucson, Arizona, 1992.

APPENDIX

Direct Evap-Cooling Routine
(ECONO-2 DIRECT, based on ref. 3)

```
FUNCTION NAME=DIRECT ..
ASSIGN  DBT=DBT      WBT=WBT
        TM=TM        PO=PO
        TR=TR        PATM=PATM
        HUMRAT=HUMRAT WM=WM
        WR=WR ..
CALCULATE ..
C EVAP-COOLER CONTROL, PERFORMANCE,
AND MIXED AIR
C TEMP CALCULATION
   IF(DBT.LT.55) GO TO 80
   PO=1.0
   EFF=0.85
C    CALCULATE EVAP COOLER EXIT DBT
   DBT2=DBT-EFF*(DBT-WBT)
   IF (DBT2.GT.68) GO TO 80
   IF (DBT.LT.68) GO TO 60
60   TM2=(PO*DBT2)+(1.0-PO)*DBT
C    PSYCHROMETRIC CALCULATION FOR
PRE-COOLED AND MIXED AIR
C    HUMIDITY RATIO
C    CALCULATE PARTIAL PRESSURE OF
WATER VAPOR IN MOISTURE
C    SATURATED AIR
   A1=-7.90298
   A2=5.02808
   A3=-0.00000013816
   A4=11.344
   A5=0.0081328
   A6=-3.49149
   TW=(WBT+459.688)/1.8
```

```
    Z=373.16/TW
    P1=A1*(Z-1)
    P2=A2*ALOG10(Z)
    DUMMY1=A4*(1-(1/Z))
    P3=A3*((10**DUMMY1)-1)
    DUMMY2=A6*(Z-1)
    P4=A5*((10**DUMMY2)-1)
    DUMMY3=P1+P2+P3+P4
    PVP=29.921*10**DUMMY3
    GO TO 85
80  TM2=DBT
85  CONTINUE
C   CALCULATE HUMIDITY RATIO OF PRE-
COOLED AIRSTREAM
    HUMRATS=0.622*PVP/(PATM-PVP)
    HUMRAT2=HUMRAT+EFF*(HUMRATS-
HUMRAT)
C   CALCULATE HUMIDITY RATIO OF MIXED
AIR AFTER EVAP COOLER
    WM1=(PO*HUMRAT2)+(1.0-PO)*WR
C   CALCULATE ROOM HUMIDITY RATIO
    WROOM=WM1+(WR-WM)
C   CALCULATE HUMIDITY RATIO OF AIR
AFTER COOLER
    HL=1093.049+0.441*TM2-WBT
    CH=0.24+0.441*HUMRATS
    WH=HUMRATS-CH*(TM2-WBT)/HL
    PV=PVP-(5.704E-04*PATM*(TM2-WBT)/1.8)
    WEVAP=0.662*(PV/(PATM-PV))
    TROOM=TM2+(TR-DBT)
C   CALCULATE THE RELATIVE HUMIDITY
OF ROOM AIR
    RH=RHFUNC(dbt,humrat,press)
    PRINT
1,TM2,DBT,TR,TROOM,WROOM,WEVAP
1
FORMAT(1X,'TM2=',F4.0,'OAT=',F4.0,'TR=',F4.0,
    1
'TROOM='F4.0,'WROOM=',F7.6,'WEVAP=',F7.6)
    END
END-FUNCTION ..
```

Indirect Evap-Cooling Routine
(ECONO-2 INDIRECT, based on ref.3)

```
$ INDIRECT EVAP-COOLER MODEL
SUBR-FUNCTIONS   ECONO-2=*INDIRECT*   ..
$HOURLY REPORTS

    RP-1 = SCHEDULE THRU DEC 31 (ALL)
(1,24) (1) ..

    BLOCK-1 = REPORT-BLOCK
            VARIABLE-TYPE = SYST-1
            VARIABLE-LIST = (3,4,8,35,36) ..

    BLOCK-2 = REPORT-BLOCK
            VARIABLE-TYPE = ZONE-J3-1
            VARIABLE-LIST= (6) ..
    HR-1    = HOURLY-REPORT
            REPORT-SCHEDULE = RP-1
            REPORT-BLOCK=(BLOCK-
1,BLOCK-2) ..

END ..
FUNCTION  NAME=INDIRECT ..
ASSIGN   DBT=DBT      WBT=WBT
    TM=TM        PO=PO
    TR=TR        PATM=PATM
    HUMRAT=HUMRAT  WM=WM
    WR=WR  ..
CALCULATE ..
C EVAP-COOLER CONTROL, PERFORMANCE,
AND MIXED AIR
C TEMP CALCULATION
    IF(DBT.LT.55) GO TO 80
    TM=(PO*DBT)+((1.-PO)*TR)
    EFF=0.65
    TM2=TM-EFF*(TM-WBT)
    IF(TM2.LT.55) GO TO 60
    TM=TM2
    GO TO 80
60  TM=55
80  CONTINUE
    TROOM=TM+(TR-DBT)
    PRINT 1,TM,DBT,TR,TROOM,PO
1   FORMAT(F4.0,F4.0,2F4.0,F4.2)
    END
END-FUNCTION ..
```

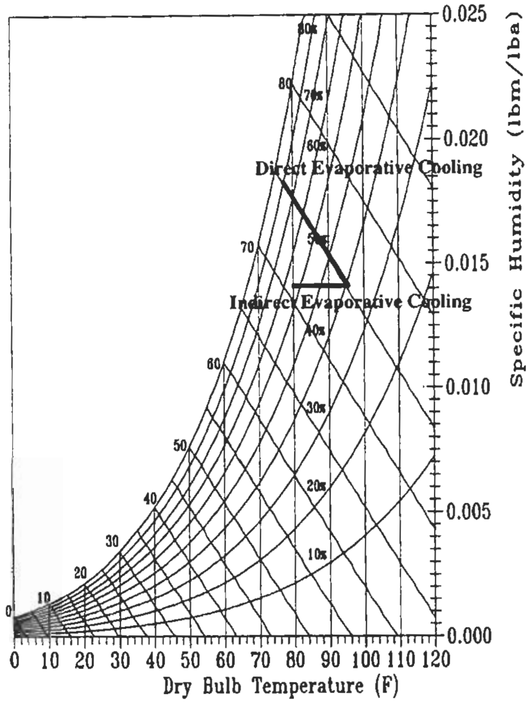

Figure 1 Psychrometrics of direct and indirect evaporative cooling.

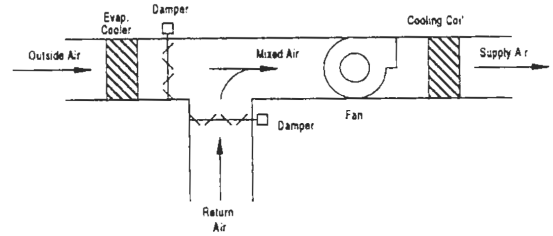

Figure 2 Schematic of direct evaporative cooling.

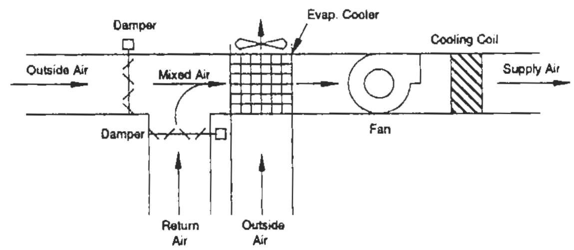

Figure 3 Schematic of indirect evaporative cooling.

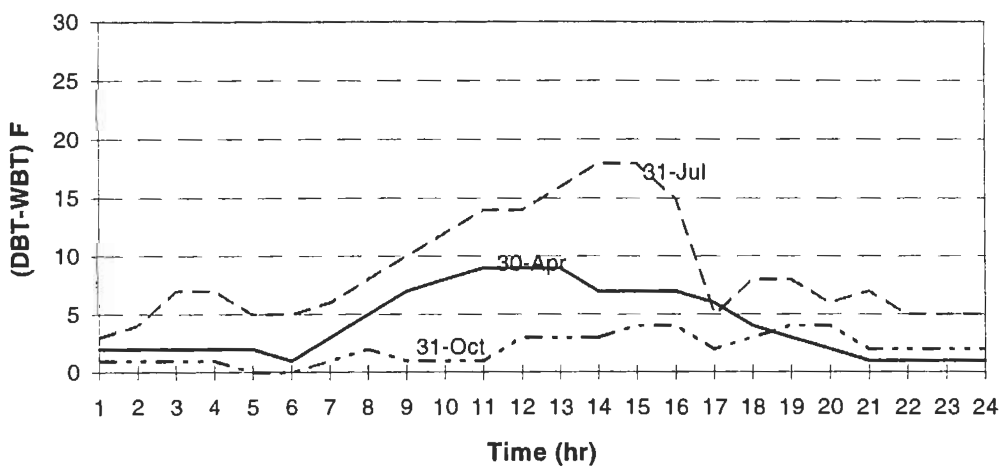

Figure 4 Variation of the difference between outside DBT and WBT with time for April 30, July 31, and October 31 for Kingsville, Texas.

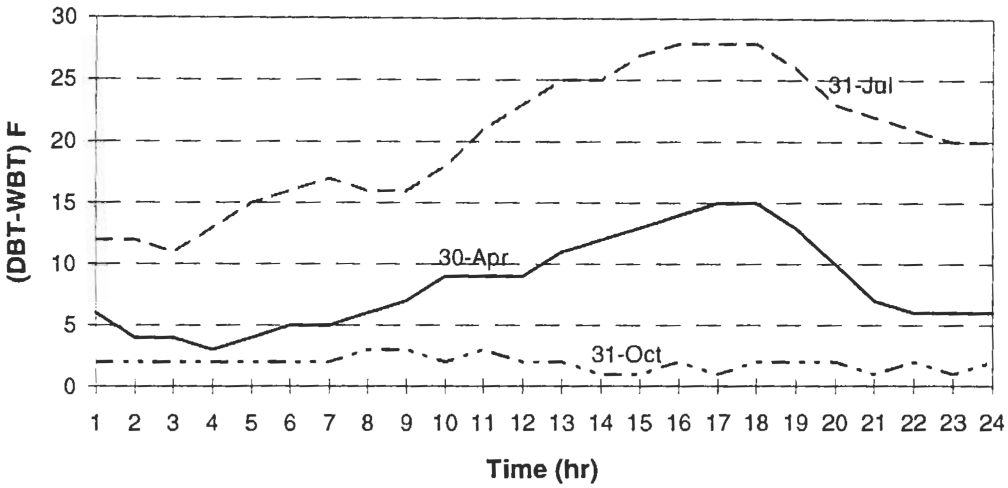

Figure 5 Variation of the difference between outside DBT and WBT with time for April 30, July 31, and October 31 for Abilene, Texas.

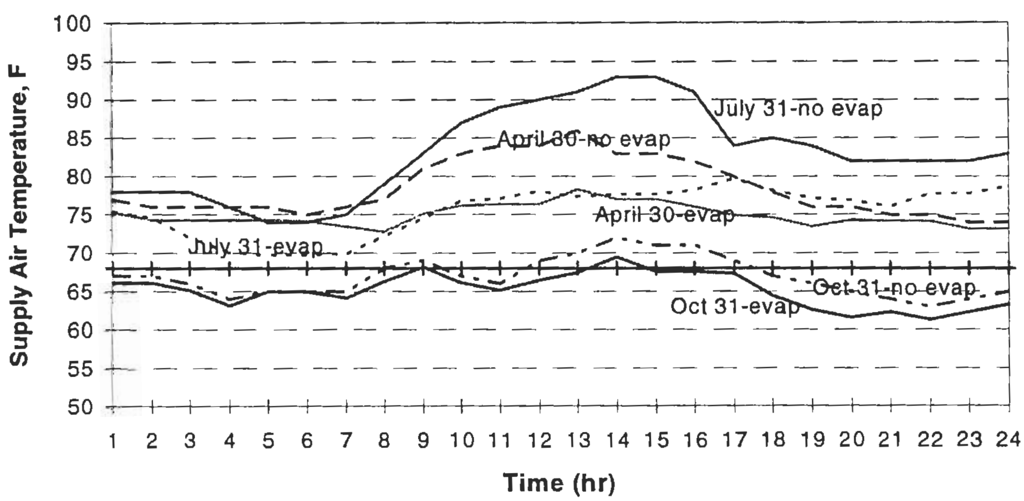

Figure 6 Variation of supply air temperature with and without direct evaporative cooling for Kingsville, Texas.

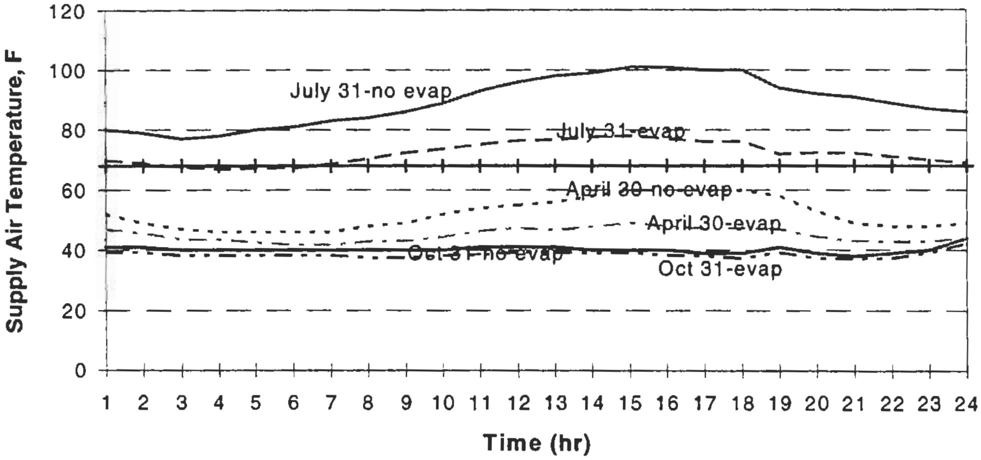

Figure 7 Variation of supply air temperature with and without direct evaporative cooling for Abilene, Texas.

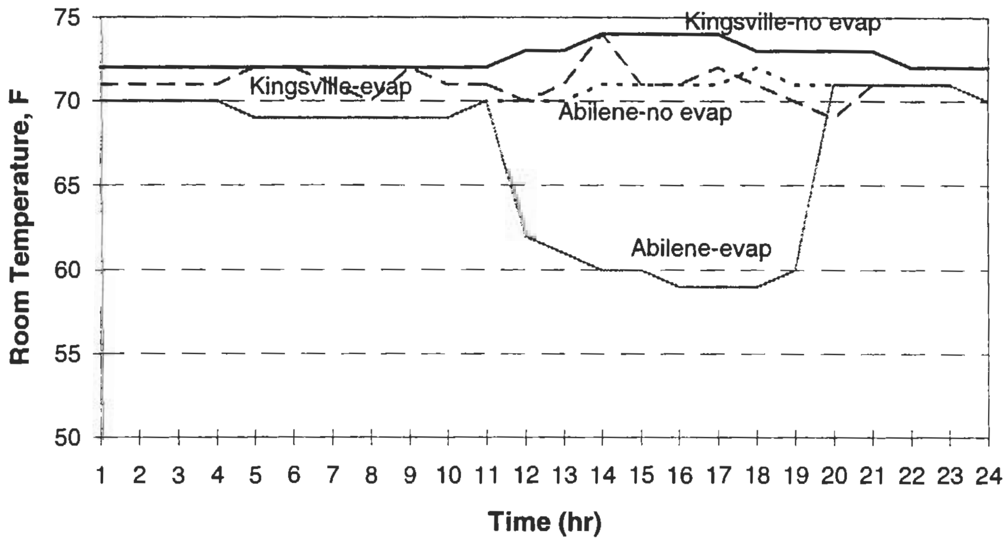

Figure 8 Variation of room air temperature in October with and without direct evaporative cooling for Kingsville and Abilene, Texas

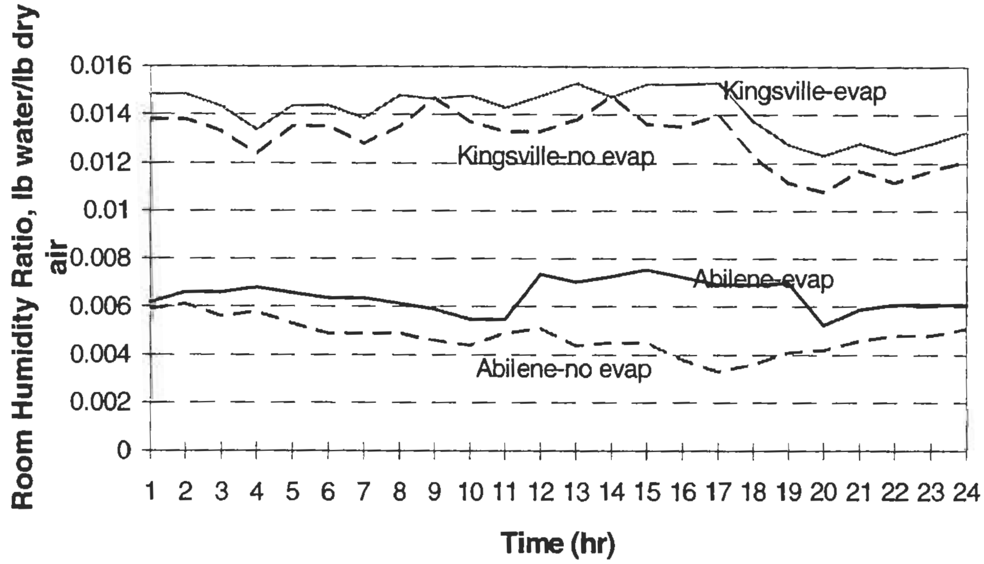

Figure 9 Variation of room humidity ratio in April with and without direct evaporative cooling for Kingsville and Abilene, Texas

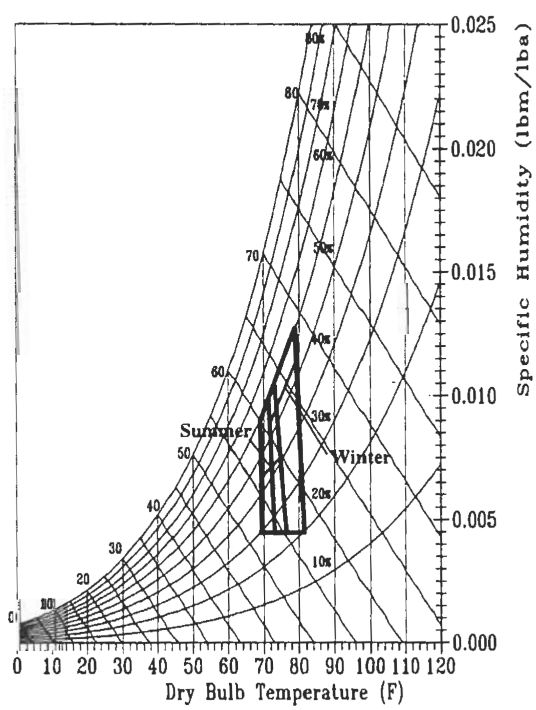

Figure 10 Standard ASHRAE comfort zones.

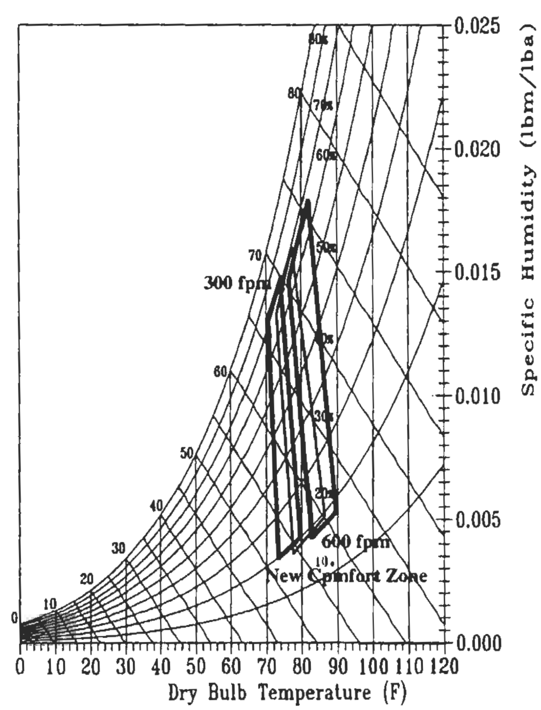

Figure 11 Change of human comfort zone due to higher air velocities.

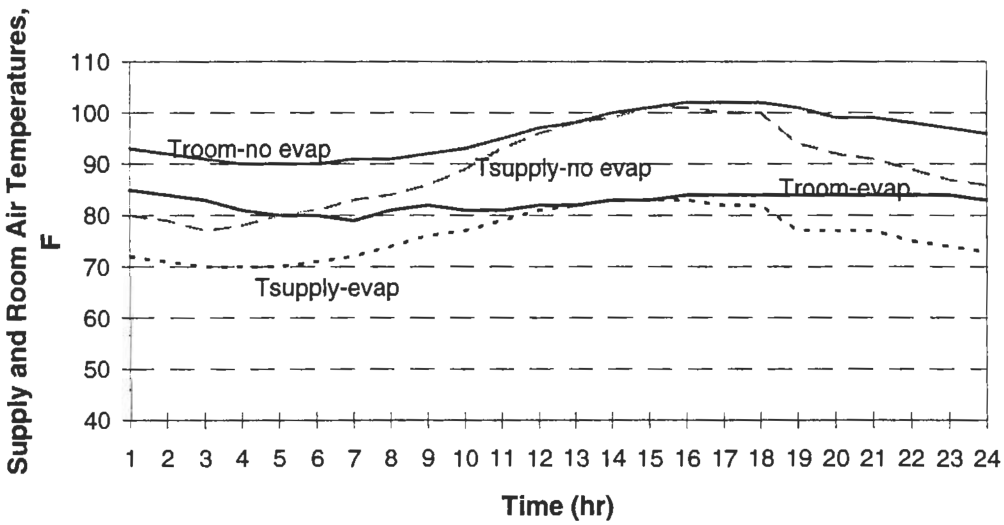

Figure 12 Variation of supply and room air temperatures with and without indirect evaporative cooling for Abilene, Texas.

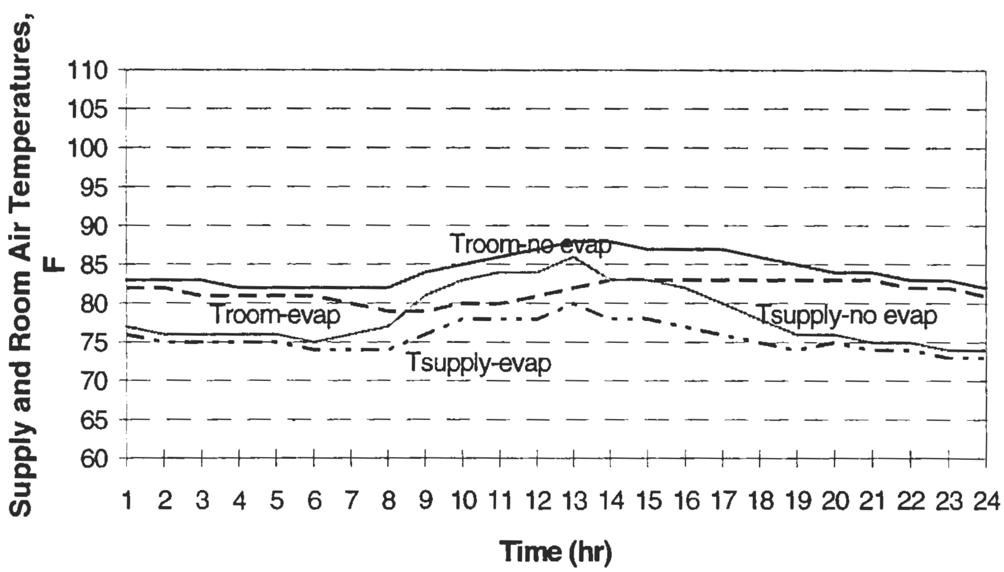

Figure 13 Variation of supply and room air temperatures with and without indirect evaporative cooling for Kingsville, Texas.

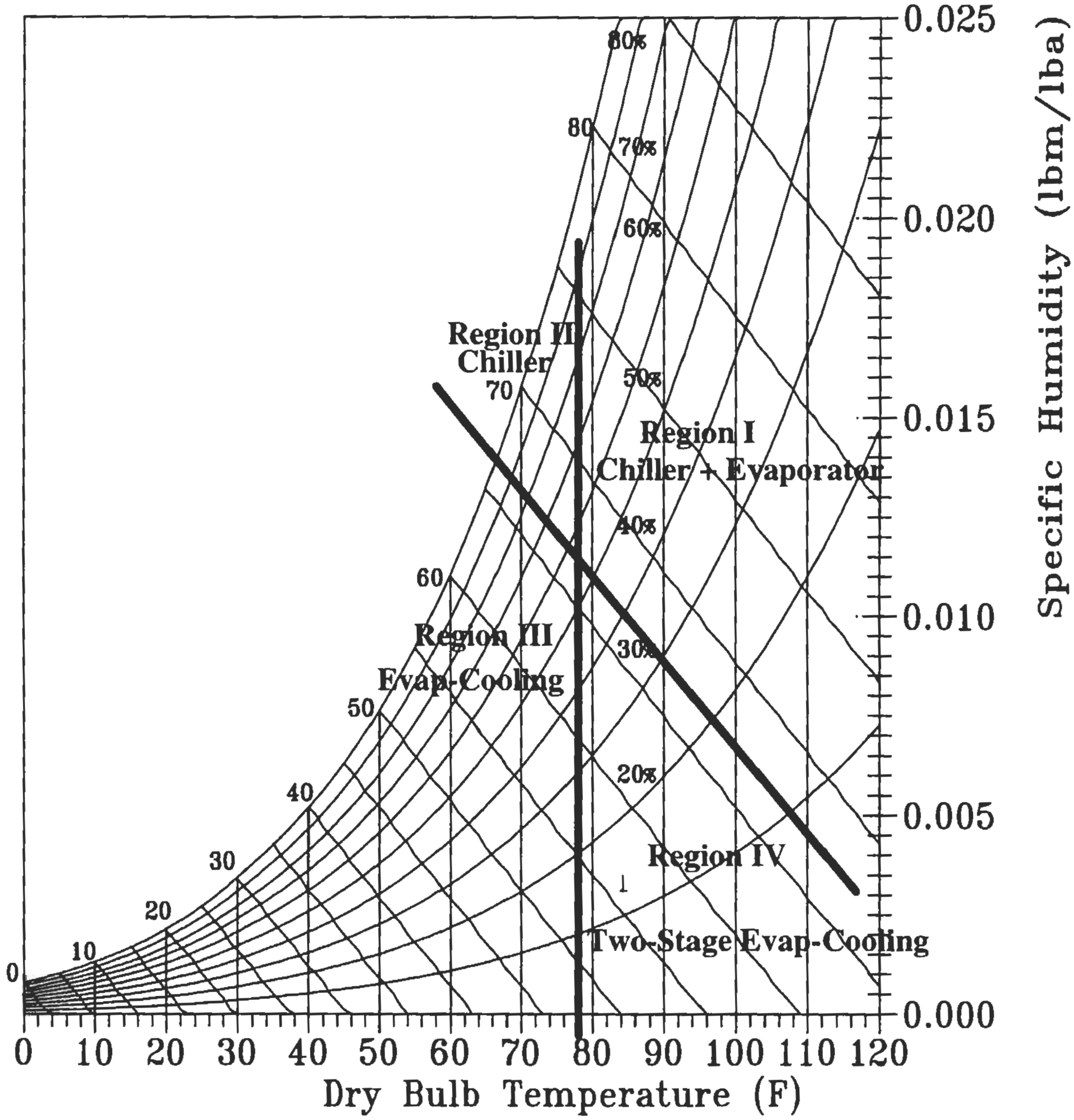

Figure 14 Regions of using evap-cooling or evap-cooling/chiller combination.