CONF-8606431--

SERI/CP-254-3000
DE89009490

October 1989

Desiccant Cooling and Dehumidification Opportunities for Buildings Workshop

T. R. Penney

June 10-11, 1986
Solar Energy Conservation Training Institute
Chattanooga, Tennessee

Sponsored by:
Electric Power Research Institute
Gas Research Institute
Tennessee Valley Authority
Coordinated by:
U.S. Department of Energy
Solar Energy Research Institute

SERI/CP--254-3000

DE89 009490

Desiccant Cooling and Dehumidification Opportunities for Buildings Workshop

T. R. Penney

June 10-11, 1986
Solar Energy Conservation Training Institute
Chattanooga, Tennessee

Sponsored by:
Electric Power Research Institute
Gas Research Institute
Tennessee Valley Authority
Coordinated by:
U.S. Department of Energy/
Solar Energy Research Institute

October 1989

Solar Energy Research Institute
A Division of Midwest Research Institute

1617 Cole Boulevard
Golden, Colorado 80401-3393

Prepared for the
U.S. Department of Energy
Contract No. DE-AC02-83CH10093

ps

NOTICE

This report was prepared as an account of work sponsored by an agency of the United States government. Neither the United States government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States government or any agency thereof.

Printed in the United States of America
Available from:
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, VA 22161

Price: Microfiche A01
Printed Copy A10

Codes are used for pricing all publications. The code is determined by the number of pages in the publication. Information pertaining to the pricing codes can be found in the current issue of the following publications which are generally available in most libraries: *Energy Research Abstracts (ERA); Government Reports Announcements* and Index (*GRA* and I); *Scientific and Technical Abstract Reports (STAR)*; and publication NTIS-PR-360 available from NTIS at the above address.

Preface

Desiccant cooling and dehumidification technology is gaining acceptance as an option for air-conditioning buildings. Over the last few years, significant advances have been made in desiccant cooling and dehumidification in the private and public sectors. Market penetration of desiccant systems in the past has been limited to unique applications.

This workshop was an open forum for discussing these unique and potential applications. Workshop participants discussed the state of the art of desiccant materials, components, systems, and applications, both theoretically and experimentally. This document is a compilation of the papers presented in the workshop sessions.

Table of Contents

Papers for the Workshop on

DESICCANT COOLING AND DEHUMIDIFICATION OPPORTUNITIES FOR BUILDINGS

Title	Author(s)	Organization
Advances in Open-Cycle Solid Desiccant Cooling	Terry R. Penney	SERI
Energy Storage and Heat Pumping with Liquid Desiccants	H. Robinson	Univ. of South Carolina
Energy-Integrated Desiccant HVAC System Applications	Gershon Meckler	Gershon Meckler Associates
Evaluation and Validation of Desiccant Cooling/Dehumidification Systems Models	J.R. Howell	Univ. of Texas-Austin
Analytical Tools for Solid Desiccant Cooling Research	K. Schultz	Univ. of Wisconsin-Madison
ENERGYMASTER - Desiccant Cooling in the Marketplace	J. Coellner	American Solar King Corp.
A Desiccant/Refrigeration Hybrid System for Air Conditioning Buildings	Kennard L. Bowlen	Cargocaire Engineering Corp.
Field Development of a Desiccant-Based Space-Conditioning System for Supermarket Applications		Gas Research Institute
Use of Liquid Desiccant Dehumidification for Cooling Buildings	Wm. C. Griffiths	Kathabar Systems
Dehumidification Systems	Brian Blackburn	Miller-Picking Corp.
Gas-Fueled Desiccant-Based Air-Conditioning	Barry M. Cohen	Tecogan, Inc.
Brief Notes and Sample Cost Calculations on Gas Engine Desiccant Hybrid Cooling for Australian Conditions	I. MacLaine-Cross	Univ. of New South Wales
Integrating Desiccant-Based HVAC Systems	Milton Meckler	The Meckler Group
Desiccant Materials	R.K. Collier, Jr.	Enerscope, Inc.
Polymers: Options as Advanced Desiccant Materials	T. Thomas	SERI

Contents *(cont)*

OTHER PAPERS SUBMITTED

Program

DESICCANT COOLING AND DEHUMIDIFICATION OPPORTUNITIES FOR BUILDINGS

June 10-12, 1986

June 10

7:45 - 8:15	**Registration** (obtain nametag and handouts)	

Session I: Workshop Opening Remarks

8:15 - 8:25	**Introduction/Welcome**	D. Carver, TVA
8:25 - 8:45	**Overview**	T. Penney, SERI
8:45 - 9:15	**Gas Utility Program**	D. Kosar, GRI
9:15 - 9:30	**Electric Utility Program**	M. Blatt, EPRI

Session II: Applications and the Potential for Desiccant Cooling

Chairman/Moderator D. Kosar, GRI

9:30 - 9:45	**Comfort Related Issues**	H. Robison, Univ. of South Carolina
9:45 - 10:00	**Energy Integrated Desiccant HVAC System Applications**	G. Meckler, GMA
10:00 - 10:15	**Gas Utility Perspective**	E. Fodor, American Gas Association
10:15 - 10:30	**Electric Utility Perspective**	G. Chinery, TVA
10:30 - 10:40	**Break**	
10:40 - 11:00	**Open Forum Discussion** -- questions from the audience	

Session III: Methods for Component/System Analysis

Chairman/Moderator: Jeff Moorehouse, Univ. of South Carolina

11:00 - 11:15	**A novel approach for understanding and controlling latent and sensible loads in buildings**	P. Fairey, Florida Solar Energy Center

Program (cont)

11:15 - 11:30	**Evaluation and Validation of Desiccant Cooling/Dehumidification Systems Models**	J. Howell, Univ. of Texas-Austin
11:30 - 11:45	**Analytical Tools for Solid Desiccant Cooling Research**	K. Schultz, Univ. of Wisconsin
11:45 - 12:05	**Open Forum Discussion** -- questions from the audience.	
12:05 - 1:00	**Catered Lunch**	

Session IV: Industry Representatives Talk about State-of-the-Art

Chairman/Moderator: G. Meckler, GMA

1:00 - 1:15	°**American Solar King** - Jim Coellner
1:15 - 1:30	°**Cargocaire Eng. Corp.** - Kennard Bowlen
1:30 - 1:45	°**Kathabar Systems** - Wm. Griffiths
1:45 - 2:00	°**Miller - Picking** - B. Blackburn
2:00 - 2:15	°**Thermoelectron** - B. Cohen
2:15 - 2:30	°**Zeopower** - D. Tchernev
2:30 - 2:45	°**Rotary Heat Exchangers** - I. Maclaine-Cross
2:45 - 3:00	°**Meckler Group** - M. Meckler
3:00 - 3:30	**Open Forum Discussion** -- questions from the audience
3:30 - 4:00	**Break**

Session V: Desiccant Materials Review

Chairman/Moderator: D. Ruthven, Univ. of New Brunswick

4:00 - 4:15	**A Review of Materials Opportune for Desiccant Cooling Applications**	K. Collier, Enerscope
4:15 - 4:30	**Desiccant Polymers -- New Options For the Future**	T. Thomas, SERI
4:30 - 4:45	**Desiccant Manufacturers Point of View**	G. Allen, Kali-Chemie Corp.

Program (cont)

4:45 - 5:00	**Desiccant Contamination - Research Equipment to Assess Capacity Changes**	T. Thomas, SERI
5:00 - 5:15	**Open Forum Discussion** -- questions from the audience	
5:15 - 5:45	**Workshop Objectives -- Ideas/Cross Fertilization**	

JUNE 11

Session VI: Concurrent Workshop Team Discussions

8:00 - 12:00	**Materials:** Chairman, Dr. Douglas Ruthven, Univ. of New Brunswick
8:00 - 12:00	**Analytical Methods:** Dr. Jeff Moorehouse, Univ. of South Carolina
8:00 - 12:00	**Components/Systems:** G. Meckler, GMA
8:00 - 12:00	**Applications:** M. Blatt, EPRI; D. Kosar, GRI
12:00 - 2:00	**Catered Lunch**

Session VII: Workshop Summaries

2:00 - 2:30	**Materials Workshop Summary**	D. Ruthven, Univ. of New Brunswick
2:30 - 3:00	**Analytical Methods Workshop Summary**	J. Moorehouse, Univ. of South Carolina
3:00 - 3:30	**Components/System Workshop Summary**	G. Meckler, GMA
3:30 - 4:00	**Applications Workshop Summary**	M. Blatt, EPRI; D. Kosar, GRI
4:00 - 5:00	**Open Forum Discussion/Closure** - questions/comments from the audience	

Program (cont)

JUNE 12

Optional Hardware Tour

8:00 - 9:30	TVA's Demonstration
9:30 - 11:00	Drive to Murfreesboro, Tennessee
11:00 - 12:00	GRI/Cargocaire's SuperAire II on a Kroger Supermarket

Pre-Registered List of Attendees

1.
Adebiyi, George
Mississippi State University
P. O. Drawer ME
Mississippi State, Miss. 39762
(601) 325-3260

2.
Akin, Joe A.
ALAGASCO
2101 Sixth Ave., No.
Birmingham, AL 35203
(205) 326-9264

3.
Allen, George I.
Kali-Chemie Corp.
41 West Putnam Ave.
Greenwich, CT 06830
(203) 629-7900

4.
Allen, Michael L.
Hussmann Corporation
12999 St. Charles Rock Road
Bridgeton, MO 63044
(314) 291-2000

5.
Anand, Dr. Dave K.
TPI, Inc.
8824 Burningtree Rd.
Bethesda, MD 20817
(301) 469-7115

6.
Askew, Greg
Tennessee Valley Authority
SP2551D-C
Chattanooga, TN 37401
(615) 751-6808

7.
Bacchus, Rocky
Bacchus Industries, Inc.
P.O. Box 465
Sunland Park, NM 88063
(505) 589-5431

8.
Barret, Terry
Bacchus Industries, Inc.
P.O. Box 465
Sunland Park, NM 88063
(505) 589-5431

9.
Bean, Michael
PRI Energy Systems
P. O. Box 22688
Honolulu, Hawaii
(808) 547-3526

10.
Belding, W.A. (Bill)
Kaiser Aluminum & Chemical Corp.
P.O. Box 877
Pleasanton, CA 94566
(415) 847-4402

11.
Bergt, Carl
Trane Company
6200 Troup Highway
Tyler, TX 75711
(214) 581-3334

12.
Bharathan, Desikan
SERI
1617 Cole Blvd.
Golden, CO 80401
(303) 987-1309

Attendees (cont)

13.
Bierwirth, Hank
Hussmann/Alton
4830 Transport Drive
Dallas, TX 75247
(214) 638-6010

14.
Blackburn, Brian
Miller-Picking Corp.
P.O. Box 130
Johnstown, PA 15907
(814) 479-4023

15.
Blatt, M.
EPRI
P. O. Box 10412
Palo Alto, CA 94303

16.
Bowlen, Kennard L.
Cargocaire Eng. Corp.
79 Monroe St.
Amesbury, MA 01913
(617) 388-0600

17.
Brothers, Steve
Tennesses Valley Authority
SP 35 31D
Chattanooga, TN 37402
(615) 751-7369

18.
Bunow,Marc
830 Summit Avenue, N.E.
Chattanooga, TN 37411
(615) 622-8770

19.
Burrows, David L.
Tennessee Valley Authority
SP 35 49D
Chattanooga, TN 37402
(615) 751-7408

20.
Carver, D.
Tennessee Valley Authority
217 Power Board Bldg.
Chattanooga, TN 37401

21.
Chaffin, David
Tennessee Valley Authority
SP2SD
Chattanooga, TN 37402
(615) 751-6847

22.
Cheek, Craig
Advanced Energy Concepts
1333 Meridian Street
Huntsville, Alabama 35801
(205) 533-4545

23.
Chinery, Glenn T.
Tennessee Valley Authority
Signal Place, 1st Floor,
Mail Stop A
Chattanooga, TN 37402
(615) 751-6855

24.
Coellner, James A.
American Solar King Corp.
700 Loop 340 South
Waco, TX 76710

25.
Cohen, Barry M.
Tecogen, Inc.
45 First Avenue
P. O. Box 9046
Waltham, MA 02254
(617) 890-8700

26.
Collier, Kirk
Enerscope, Inc.
4886 W. Port Au Prince
Glendale, AZ 85306

27.
Conlin, Francis
North Carolina Alternative
Energy Corporation
P.O. Box 12699
Durham, NC 27709
(919) 549-9046

28.
Cooley, Donald L.
Tennessee Valley Authority
ST3S63D-C
Chattanooga, TN 37401
(615) 751-7396

29.
Eaves, M.
SCG

30.
Fairey, Philip
Florida Solar Energy Center
300 State Road 401
Cape Canaveral, FL 32920
(305) 783-0300

31.
Fitch, Stephen D.
Vice President-Chief Engineer
Bry-Air, Inc.
P.O. Box 269
Sunbury, OH 43074
(614) 965-2974

32.
Fletcher, Jim
Lockheed
39 Barton Avenue
Rockledge, FL 32955
(305) 867-0549

33.
Fodor, Eben
American Gas Association
1515 Wilison Boulevard
Arlington, VA 22209
(703) 841-8573

34.
Fraioli, Anthony V.
Argonne National Laboratory
Argonne, IL 60439
(312) 972-7474

35.
Goldsmith, John C.
Dept. of Energy
Solar Buildings Division
1000 Independence Ave.
Washington, D.C. 20585
(202) 252-8779

36.
Griffiths, William C.
Kathabar Systems
P.O. Box 791
New Brunswick, NJ 08903
(201) 356-6000

37.
Guinn, Gerald R.
Director
Alabama Solar Energy Center
Univ. of Alabama-Huntsville
Huntsville, AL 35899
(205) 895-6707

38.
Healey, Henry M.
Consultant
Tennessee Valley Solar
2155 Jason Street
Merritt Island, FL 32952
(305) 459-1263

39.
Hines, Prof, Anthony
Oklahoma State University
College of Engineering
Stillwater, OK 74078

40.
Howard, David T.
Tennessee Valley Authority
SP2563D
Chattanooga, TN 37401
(615) 751-6830

Attendees (cont)

41.
Howell, J.
Univ. of Texas - Austin
10100 Burnet Road
Austin, TX 78758

42.
Ira, Stephen
Tennessee Valley Authority
3S 44D SP
Chattanooga, TN 37401
(615) 751-7411

43.
Irwin, William C.
Tennessee Valley Authority
2S39A Missionary Ridge Place
Chattanooga, TN 37402-2801
(615) 751-6118

44.
Jones, Richard
Red Foods
Chattanooga, TN

45.
Joyner, Powell A.
Electric Power Research
Institute
Box 10412
Palo Alto, CA 94303

46.
Kemp, Terry N.
Tennessee Valley Authority
SP 35 52D
Chattanooga, TN 37402
(615) 751-7351

47.
Keresteciogly, Alp
Florida Solar Energy Center
300 State Road 401
Cape Canaveral, FL 32920
(301) 783-0300

48.
Klein, Richard H.
Tennessee Valley Authority
ST1S57A-C
Chattanooga, TN 37401
(615) 751-6858

49.
Knight, Dr. Charles V.
Univ. of Tennessee-Chattanooga
School of Engineering
McCaille Ave.
Chattanooga, TN 37402
(615) 842-7986

50.
Kosar, Doug
Gas Research Institute
8600 West Bryn Mawr Ave.
Chicago, IL 60631
(312) 399-8282

51.
Lannus, Dr. Arvo
EPRI
P. O. Box 10412
Palo Alto, CA 94303

52.
Lavan, Zalman
Illinois Institute of Technology
10 W. 32nd Street
Chicago, IL 60202
(312) 567-3189

53.
MacLaine-Cross, Ian L.
University of New South Wales
P.O. Box 1
Kensington, New South Wales
2033
(612) 697-4112

54.
Matsuki, Kenji
Sharp Corporation
282-1 Hajikami, Shinjo-Cho
Nara, Japan

Attendees (cont)

55.
Maye, Bobby
Kwik-Kold Ref.
Birmingham, AL

56.
Maye, John
Kwik-Kold Ref.
Birmingham, AL

57.
Meckler, Gershon
Gershon Meckler Associates, P.C.
40 Rector Street, 10th Floor
New York, NY 10006
(212) 693-0366

58.
Meckler, Milton
Meckler Group
17525 Ventura Blvd., Suite 307
Encino, CA 91316
(818) 995-7672

59.
Meyer, Paul R.
TVA Energy Use Test Facility
DB PSC 3-C
Chattanooga, TN 37401
(615) 697-4255

60.
Morehouse, Jeffrey H.
University of South Carolina
Mech. Eng. Dept.
Columbia, S.C. 29208
(803) 777-4185

61.
Novosel, Davor
Gas Research Institute
8600 W. Bryn Mawr Avenue
Chicago, IL 60631
(312) 3990-8258

62.
Pash, Joe
Kathabar Systems
P. O. Box 791
New Brunswick, NJ 08903
(201) 356-6000

63.
Peterson, John L.
Center for Energy Studies,
University of Taxas at Austin
10100 Burned Road
Austin, TX 78750
(512) 471-3104

64.
Pesaran, Ahmad
SERI
1617 Cole Blvd.
Golden, CO 80401
(303) 231-7636

65.
Picking, III, Howard M.
Miller-Picking

66.
Rhodes, Barry V.
Tennessee Valley Solar, Inc.
Route 1, Box 595
Chickamauga, GA 30707
(404) 866-3511; 375-2503

67.
Robison, Harry
Professor Emeritus
University of South Carolina
27 Chappel Creek Rd./Litchfield
Plantation
Pawleys Island, SC 29585

68.
Roose, Thomas R.
Gas Research Institute
8600 West Bryn Mawr Avenue
Chicago, IL 60631
(312) 399-8207

69.
Russell, Lynn D.
Mississippi State University
P. O. Drawer ME
Mississippi State, MS 39762
(601) 325-3260

70.
Ruthven, Prof. D.M.
University of New Brunswick
P. O. Box 4400
Fredericton, W.B.
Canada E3B5A3

71.
Sanders, Jr., John W.
Tennessee Valley Authority
Signal Place 2S 32D
Chattanooga, TN 37402
(615) 751-3591

72.
Scharer, Charles R.
Tennessee Valley Authority
Energy Use Test Facility
DB PSC 3-C
Chattanooga, TN 37401
(615) 697-4266

73.
Schultz, Kenneth
Solar Energy Lab
Univ. of Wisconsin-Madison
1500 Johnson Drive
Madison, WI 53706
(608) 263-3346

74.
Shoemake, Ron
Red Foods
Chattanooga, TN

75.
Swanson, John L.
Tecogen, Inc.
45 First Avenue
P. O. Box 9046
Waltham, MA 02254
(617) 890-8700

76.
Tchernev, Dr. Dimiter I.
The Zeopower Company
75 Middlesex Avenue
Natick, MA 01760
(617) 655-4125

77.
Teagan, W. Peter
Arthur D. Little, Inc.
20 Acorn Park
Cambridge, MA 01240
(617) 864-5770

78.
Thomas, Terry
SERI
1617 Cole Blvd.
Golden, CO 80401
(303) 231-7273

79.
Tiller, Jeffrey
Georgia Solar Coalition, Inc.
P. O. Box 5506
Atlanta, GA 30307
(404) 525-7657

80.
Turner, Robert
Univ. of Nevada

81.
Walker, David
Foster-Miller, Inc.
350 Second Avenue
Waltham, MA 02254
(617) 890-3200

82.
Weeks, J.W.
Tennessee Valley Authority
SP 35 45D
Chattanooga, TN 37402
(615) 751-7360

83.
Wood, Joan M.
Tennessee Valley Authority
Signal Place 1564A
Chattanooga, TN 37377
(615) 751-6852

84.
Worek, Dr. William M.
Illinois Institute of Technology
10 West 32nd Street
Chicago, IL 60616
(312) 567-3200

Advances in Open-Cycle Solid Desiccant Cooling

Terry R. Penney
Ian Maclaine-cross

Solar Energy Research Institute
1617 Cole Boulevard
Golden, Colorado 80401

ABSTRACT

Of the solar cooling options available open cycle solid desiccant cooling looks very promising. A brief review of the experimental and analytical efforts to date shows that within the last 10 years thermal performance has doubled. Research centers have been developed to explore new materials and geometry options to improve and validate mathematical models that can be used by design engineers to develop new product lines. Typical results from the Solar Energy Research Institute's (SERI) Desiccant Cooling Research Program are shown. Innovative ideas for new cycles and spinoff benefits provide incentives to continue research in this promising field.

INTRODUCTION

One function of buildings is to maintain comfortable temperatures and humidity. Reducing the nonrenewable energy required to achieve this is a major national goal. In summer, infiltration, ventilation, solar gains, building envelope conduction, internal heat and moisture generated by the occupants, and equipment raise the temperature and humidity of the conditioned space. Better equipment; building design; and insulation, sealing, and ventilation systems can reduce but not eliminate five of these heat loads. The heat and moisture generated by occupants depend on building function and cannot be significantly altered [1]. A cooling system with reduced energy consumption, especially one using renewable energy that can be nationally marketed and retrofitted to existing building stock, is an extremely attractive alternative to the consumer and utility and the industry that supplies and installs the product.

Conventional vapor compression systems using electricity have reached their limits after sixty years of research and development. The cost of electricity increases at a rate depending on supply and demand. New cooling systems using renewable energy sources directly are being developed when the economics show a favorable rate of return. This paper focuses on cooling options using the renewable energy sources available nationally, which are direct solar and a type of indirect solar, the energy of the unsaturated atmosphere.

These energy sources impose constraints on the candidate systems. Closed cooling systems in which the refrigerant does not contact the atmosphere can not use the energy of the unsaturated atmosphere. Open systems evaporating water into the atmosphere as a refrigerant use this energy source and use less direct solar energy; e.g., the simple evaporative cooler requires energy only to move the water and air. Closed systems may be combined with open systems to take advantage of this. The performance of open systems is reduced in humid areas unless dehumidification is used. A solar collector will deliver more heat at low temperatures because heat losses are less [2,3.]. These facts lead us to choose open cycles using dehumidification with low temperature heat as promising options for a national renewable energy cooling system.

A nationally marketed and retrofitted system ideally needs to be compact and have low cost and maintenance with high performance and reliability. High performance may be achieved if the system approaches the limits set by the second law of thermodynamics. This requires that all heat transfer should be across small temperature differences and all mass transfers across small concentration differences. Of commercial dehumidifiers only the adiabatic regenerative dehumidifier does this.

The design of regenerative dehumidifiers is complicated by the mass transfer resistance and poor mechanical and manufacturing properties of available desiccants. New materials, geometries, and manufacturing techniques are capable of reducing size, cost, and maintenance while increasing the performance and reliability of regenerative dehumidifiers. An improved adiabatic regenerative dehumidifier will facilitate national marketing and retrofitting of desiccant open cooling systems.

Open-cycle desiccant systems using adiabatic regenerative dehumidifiers have been referred to as adiabatic desiccant open cooling (ADOC) systems. The Solar Energy Research Institute (SERI) for the Department of Energy's (DOE) Active Heat and Cooling Program has been pursuing basic research and development in the area of solid desiccant solar cooling as one alternative. This paper briefly describes the development of this technology to its current state and work currently in progress at SERI and elsewhere. The likely future trends and opportunities, however, are of greatest interest and are consistent with DOE's long-term goals for renewable energy cooling. Near-term applications for conservation are presented to show how desiccant cooling is practical now, and spinoffs to other technologies are imminent.

A SHORT HISTORY OF THE TECHNOLOGY

The simple evaportive cooler is the oldest open-cycle cooling system. During the 1930s, it was developed to the form that is still widely marketed today [4]. The small amount of energy it consumes for fans and pumps can be reduced considerably by applying modern knowledge of heat and mass transfer and fluid mechanics [5]. Simple evaporative coolers only cool down to the wet bulb temperature, which limits their effective use to the hot dry areas of the Southwest.

Indirect and regenerative evaporative coolers were developed as an improvement to these simple evaporative coolers of the 1930s [4]. Later, however, they did not compete economically with conventional vapor compression or simple evaporative coolers. The large size of the metal plate heat exchangers was an important disadvantage. More compact and economical heat exchangers for these coolers have been developed and marketed [5,6,7] with further improvements possible [8,9,10]. Indirect and regenerative evaporative coolers cool down to the dew point temperature, which, being less than the wet bulb temperature, gives them a broader climatic range than the simple evaporative coolers. This range does not extend to humid climates.

Pennington [11] solved the problem of limited climatic range of open cooling systems. He saturated a rotary heat exchanger matrix with a desiccant solution making an adiabatic regenerative dehumidifier. He coupled this dehumidifier with a heat source to a double regenerative evaporative cooler (Figs. 1 and 2). Pennington's cycle is now routinely referred to as the ventilation cycle. Pennington gave a sound description of the phenomena in a regenerative dehumidifier but did not give the performance of his prototype. He discovered that rotating the regenerative dehumidifier at a reasonably high speed would make it transfer the total heat or enthalpy to the incoming fresh air stream during winter heating.

Munters [12] improved Pennington's regenerative dehumidifier by introducing parallel passages. Regenerative dehumidifiers for drying conditioned spaces are commercially available from a number of suppliers [13]. Total heat or enthalpy regenerators for building exhaust air to fresh air energy recovery are also available [13].

The Institute of Gas Technology (IGT) recognized the potential of the ventilation, or Pennington, cycle for gas-fired air conditioning and built a number of prototypes in the 1960s [14]. Solar energy prototypes were built in the late 1970s by IGT [15], AiResearch Manufacturing Company [16] and Exxon Research and Engineering Company [17]. By 1981 thermal coefficients of performance (COP) (ratio of cooling energy delivered to heat energy supplied) over 0.5 had been achieved experimentally at ARI standard test conditions.

The early IGT effort was hampered by the lack of a satisfactory analytical method for predicting the performance of regenerative dehumidifiers. The available methods [18,19,20] could only predict performance at rotational speeds so low that the whole matrix was in equilibrium with the entering airstream at the end of both dehumidification and regeneration periods.

Dunkle [21], in response to a request from the Australian mining industry, had independently of Pennington [11] developed a proposal for an ADOC cycle. A compact, spirally wound, parallel-plate heat exchanger was developed [5]. Johnston [22] used the results of tests on packed beds of silica gel beads to predict the performance of Dunkle's cycle. The predictions were disappointingly low, and Dunkle's group realized that the regenerative dehumidifier needed to be understood before ADOC cycles could be improved.

Banks, working in Dunkle's laboratory, initiated the development of a comprehensive theory of combined heat and mass transfer in regenerative dehumidifiers and related devices in 1967. He introduced combined potentials, analogous to temperature, and combined specific capacity ratios, analogous to specific heat ratios, in terms of which combined heat and mass transfer can be described [23]. These quantities can be calculated from the vapor pressure isotherms, the specific heat of the matrix, and the thermodynamic properties of moist air. Maclaine-cross and Banks [24] used combined potentials and specific capacity ratios in predicting the performance of the regenerative dehumidifier and enthalpy regenerator. This analogy theory makes the many phenomena of regenerative dehumidifiers understandable and has subsequently been simplified for rapid calculation by Jurinak [25].

Nelson et al. [26] applied the analogy theory of regenerative dehumidifiers to predicting the performance of ADOC cycles. They proposed a regenerative dehumidifier with a matrix consisting of thin parallel plates of silica gel separated by uniform air passages. Their predicted performance was much better than previously measured for ADOC cycle performance.

Maclaine-cross [27] included the effect of a Lewis number other than unity in the analogy theory and found good agreement with a accurate finite difference solution of the partial differential equations. This finite difference algorithm was based on detailed numerical analysis and extrapolation to zero grid size using three carefully chosen grid sizes. His program, MOSHMX, [27] has been widely used by subsequent investigators.

Jurinak and Mitchell [28] used MOSHMX to study the effect of isotherm shape, maximum water content, heat of adsorption, regenerative matrix thermal capacitance, matrix moisture diffusivity, and adsorption hysteresis on dehumidifier performance. They found that the best performance was obtained with a type 1 vapor pressure isotherm shape, characteristic of microporous silica gel desiccant. They concluded that commercially available microporous silica gels were attractive materials for dehumidifier construction.

Jurinak [25] also studied the effect of matrix properties on the ventilation cycle performance. Adding heat capacity to a desiccant regenerative dehumidifier matrix increases COP and, hence, reduces energy consumption slightly. However, it reduces specific capacity greatly, which increases capital costs. Minimum life cycle cost requires a minimum desiccant matrix heat capacity. Jurinak and Mitchell [29] recently used MOSHMX to predict the optimum purge rate for a regenerative dehumidifier. For operating conditions typical of the ventilation cycle the predicted benefit of purge was small.

Effective Lewis numbers measured on packed beds of silica gel spheres have been between 3 and 30, indicating excessive mass transfer resistance [30, 31]. Mass transfer resistance of silica gel has been investigated at UCLA [32,33]. SERI has fabricated and tested parallel plate matrices of desiccant-coated polyester films [34]. The smaller particle size used by SERI greatly reduces the mass transfer resistance, and the dehumidifier approximates the Nelson et al. [26] proposal previously discussed.

A thermal COP of 1.1 was predicted at standard test conditions for ventilation cycles using this dehumidifier [35,36]. This makes a gas-fired ventilation cycle competitive in energy consumption with electric vapor compression [36,37]. However, higher COP and reduced capital costs are necessary for a solar energy cooling system to compete with conventional alternatives. The application of ADOC cycle components to energy conservation in building cooling has been proposed and investigated. Some applications are currently commercial, and others are so promising they deserve mention. By encouraging the manufacture and improvement of ADOC cycle components, the applications in the following two paragraphs are contributing to the development of a national renewable energy cooling system.

Vapor compression systems dehumidify inefficiently, and adiabatic regenerative dehumidifiers dehumidify more efficiently. Combining the two into a hybrid cycle with the condenser providing most of the regeneration heat (Fig. 3) gives substantial overall energy conservation [38,39,40,41]. For supermarkets with open refrigerated cabinets hybrid cycles are economical today using commercially available dehumidifiers [42].

Fresh air introduced into office buildings adds substantially to the cooling loads. The sensible load may be reduced or cooling may be extracted by using an evaporative cooler in the exhaust air stream before a rotary heat exchanger by exchanging with the fresh air [9,10] (Fig. 3). The rotary heat exchanger will work as an enthalpy exchanger [43,44] whenever the outside air is humid. When the outside air is dry, the evaporative cooler and heat exchanger work together as a regenerative evaporative cooler, substantially reducing energy consumption.

Recent theoretical research into advanced ADOC cycles has shown the possibility of substantially higher thermal COPs. Maclaine-cross and Banks [8] proposed a regenerative evaporative cooler that minimized the temperature differences across which heat was transferred and the concentration differences across which moisture was transferred. The second law of thermodynamics would lead one to expect improved performance, and the predicted performance was substantially better than less thermodynamically ideal regenerative evaporative coolers. Lavan et al. [45] calculated the thermal COP for a thermodynamically reversible ventilation cycle at standard test conditions as 4.66. Maclaine-cross [46] has proposed an ideal ADOC cycle requiring no thermal energy (i.e., infinite COP) at standard test conditions.

Reduction of collector cost improves the economics of a renewable energy cooling system in a similar fashion to increasing thermal COP. Solar collector costs will be reduced by economies of scale as solar heating expands. Recent research [47,48] into collector materials, manufacturing methods and concepts has shown considerable promise of reducing costs further.

A renewable energy cooling system can follow various development paths. Fundamental work on solid desiccant dehumidifiers shows immediate promise for advanced ADOC cycles unexplored to date.

THE CURRENT NATIONAL EFFORT ON DESICCANT COOLING

Several organizations (private, utility, and federal sectors) have been working on various desiccant cooling, integrated HVAC options (Fig. 4). Each organization has its own development strategy, depending upon its product line, principal interest, or long-term mission statement. It is beyond the scope of this paper to be comprehensive and knowledgeable about all of the research activities in the field of desiccants; however, we do briefly mention the publicly available larger efforts with special attention to the work funded by the Department of Energy.

The Gas Research Institute (GRI) is conducting one of the larger gas-fired cooling research efforts. GRI's strategy [49] for cooling systems is to develop cost-effective, gas-fired and recovered-heat-driven adsorption and absorption equipment for a range of sizes and applications within the residential and commercial markets. GRI's desiccant dehumidification/cooling program includes both near- and mid-term projects. All hardware development projects currently involve HVAC manufacturers on a cost-shared basis. GRI feels integrated dehumidification/chiller systems that separate latent and sensible load control show good potential for the near term, especially when cost optimized systems are packaged using existing or readily modified technology. These systems are being developed first for supermarkets, which are an ideal early market application because of the high latent-sensible load ratio. The technology will then be extended to the broader commercial market in the 20- to 100-ton size range.

GRI is also looking into the technical and economic feasibility of combining desiccant dehumidification with passive and other sensible cooling techniques for residential buildings. Advanced desiccant cooling technology, based on the early work by Exxon Corporation, is being developed for residential and light commercial applications in the 2- to 20-ton size range. Two parallel development projects are in place, each involving a major HVAC manufacturer, for designing and fabricating laboratory prototype units. Aside from these larger hardware projects, GRI has funded SERI to investigate the use of enhanced geometries to promote the rate of mass transfer [50]. This work just started in February 1985.

American Solar King Corporation, committed to near-term commercialization of a solar regenerated desiccant cooling system, has introduced the first residential solar desiccant cooling unit, which is intended to directly compete [51] with electric vapor compression units. The Solar King desiccant system, called the Sunaire, is packaged with eight, 4x8 collectors and a storage tank with backup heating devices. The company thinks it can meet its target cost of $10,000 with an annual sale of 10,000 units, which is estimated to be only 0.7% of the market penetration for air conditioning. Field testing at eight sites, mostly in the southern part of the United States, has given Solar King the confidence in the systems mechanical reliability. Solar King places a high priority on reliability for their system, which has been designed to be cost-effective on a life cycle basis. This was only achieved after careful engineering tradeoffs without compromising the mechanical integrity of the system.

The Federal Government in the last few years has restructured its role to only fund long-term, high risk research and development that will not be pursued by the private sector under normal incentives. Hence, the current DOE solar desiccant cooling projects are limited to developing high performance, low cost desiccants and other system component materials; researching heat and mass transfer phenomena in absorbent and adsorbent geometries; and analyzing advanced desiccant cooling system concepts as a mechanism for quantifying the impact of material and component progress. A conscious effort is being made to leave engineering development to the private sector, which is investing their own resources to commercialize the technology.

A fundamental research aim in materials and sorption kinetics is to develop predictive analytical models that are improved by validation with experimental data. In some cases defining properties for existing desiccant materials in the range applicable for cooling can provide accurate quantitative data to industry that is immediately beneficial.

SERI has the federally funded national lead in solar regenerated solid desiccant cooling research. A five-year plan with associated long-term technical and programmatic goals consistent with this current philosophy has been developed by Zangrando [52]. Researchers at SERI have focused primarily on material and heat and mass transfer issues. The Argonne National Laboratory (ANL) is concentrating on advanced material options. The University of Wisconsin is supporting both ANL and SERI through systems analysis activities. The Florida Solar Energy Center (FSEC) has recently started looking at novel cooling concepts. Professor Lavan at Illinois Institute of Technology (IIT) is investigating cooling concepts with regard to second law thermodynamic analysis. Each element of this integrated national team effort is discussed later. A simplified flowchart of current experimental and analytical activities (Fig. 5) shows this interaction.

CURRENT MATERIALS RESEARCH ACTIVITIES

The cost-effective deployment of many new renewable energy conversion technologies is currently limited by the durability and life cycle cost of the materials used. Research on desiccant materials and degradation studies of their sorption performance are greatly needed. Fundamental materials research includes efforts to understand and quantify the degradation processes of common desiccants and to provide new material options for improving the sorption kinetics and, thus, the mass transfer in solid desiccant geometries. The ideal goal is to identify a cost-effective material with optimal performance in the temperature range unique to a solar regenerated desiccant dehumidifier. The important parameters for the solid desiccant material would include the sorption capacity, the heat of sorption, the rate of sorption at or near the desiccant bed temperature, the rate of vapor evolution at an elevated temperature, and the cyclic repeatability of the sorption amount and rates.

From sorption isotherms the binding energy can be determined and used to calculate the rate of desorption from the surface. The rate of diffusion through the porous material can be calculated and in principle, the rate of evolution from the solid can be determined and compared with experimental results. Although considerable sorption data are available in the literature for candidate desiccant adsorbent materials, these data have not been analytically scrutinized to determine what maximum rates of water vapor evolution are possible at very low driving potentials. Recently, GRI let a contract to Collier at ENERSCOPE, Inc. to study the effect of adsorption isotherm shape on cooling system performance. The results of this work are specifically aimed at developing a new desiccant material for use in gas-fired cooling machines.

We know that sorption rates will be reduced by pore volume diffusion and intraparticle diffusion in various geometries. Recent papers by Pesaran and others have attempted to validate analytical models of this process from experiments on solid desiccants. Previous work has focused on conventional volumetric and scanning electron microscopic (SEM) techniques to provide a limited understanding of the fundamental sorption processes. To understand the molecular processes for water-solid adsorbent desiccant systems, Czanderna at SERI is combining the use of a quartz crystal microbalance [53], infrared and surface spectroscopies to correlate rates and amount sorbed with the chemical bonds formed (related to the heats of adsorption), and surface composition (effects of chemical modificiation). Any changes in the amount sorbed and rate of sorption should result from a change in the bonding at the surface (infrared).

The experimental data are not sufficiently available for assessing the potential of polymeric materials designed and used as desiccants. The objectives of this research have been developed from the obvious unexplored opportunities for providing scientific understanding and information about candidate water-desiccant materials and rationale for modifications of the adsorbents that might be necessary.

Researchers at SERI, collaborating with Argonne National Laboratory (ANL), are assessing the use of desiccants in candidate devices. They are concerned with the form of the desiccant materials; quantification of performance of fundamental geometries with desiccants attached, including total desiccant geometries, vapor flows, and heat and mass exchange characteristics in these geometries; sorption hysteresis; and desiccant material contamination/degradation effects. Recent quantitative results (Fig. 6) show how the materials capacity changes with time using field samples of silica gel regenerated by gas and electric sources. Specifically, the ANL work is concerned with securing isotherms for water on nonconventional desiccants, such as manganese oxides; analyzing for surface area and pore size distribution; separating interparticle and intraparticle diffusion processes; and studying new material combinations with a low heat of adsorption and less diffusional resistance.

Regular density silica gel is the preferred desiccant for solar application by universities and industry, yet there is considerable scatter (±15%) in the equilibrium isotherm data published in the literature. The SERI sorption test facility (Fig. 7) was used by Pesaran and Zangrando [54] and Zangrando et al. [55] to quantify and refine the equilibrium properties of regular density silica gel, molecular sieve, and one form of manganese dioxide under temperature and humidity conditions typical of those occurring in solar desiccant cooling system operations. Typical results from that work are shown in Figure 8.

HEAT AND MASS TRANSFER RESEARCH

The SERI sorption test facility was also used to generate mass transfer rate data under isothermal conditions for a packed bed geometry and a single channel of parallel passage configuration [54]. The mass transfer rate data were obtained by monitoring the outlet water vapor concentration as a function of time after a step change in the inlet water vapor concentration to the desiccant. This set-up provided initial isothermal data for comparison with mass transfer models of simple geometries. This flow of information from a facility output to a model validation path is shown on Fig. 5.

Another facility at SERI, called the single blow test facility (see Fig. 9), provides heat and mass transfer rate data under adiabatic conditions, which are more realistic of practical dehumidifier operations in a static (nonrotating) mode. The first geometry considered for commercial desiccant dehumidifiers was a packed bed. Kutscher and Barlow [31] found that the packed bed provided low heat-transfer-to-pressure-drop ratios (St/f = 0.05) and was economically unattractive for use in solar desiccant cooling applications. Further testing by Barlow and Schlepp [34] indicated that the parallel passage geometry generated a larger heat-transfer-to-pressure-drop ratio (St/f = 0.49). Typical experimental results from a prallel passage geometry in a single blow operation is shown in Fig. 10.

Since performance is not the only relevant parameter for a cost-effective desiccant system, other configurations, such as honeycombs, air-side enhanced geometries, solid desiccant impregnated surfaces, extrudable polymeric geometries, etc., are all providing incentives to various industrial suppliers, which further enhances the opportunity for near-term industrial marketability. These configurations are being tested in the single-blow test facility to investigate their performance under conditions typical of solar air conditioning.

These trade-offs (between cost, performance, and reliability) are best answered by the private sector who then completes the engineering design and product development for satisfactory residential and commercial applications. The basic analytical tools developed and validated by the research groups can be used by industry to encourage critical technical review of new options extrapolating to other physically similar geometries and processes.

Van Leersum and Close [30] showed that the dynamic rotating behavior of desiccants in the adsorption/desorption cycle can significantly affect the performance of a dehumidifier. To characterize this dynamic behavior SERI constructed a cyclic test facility (see Fig. 12) [56] to characterize the performance of desiccants on geometries in cyclic operation at conditions typical of those that would occur in a solar regenerated desiccant cooling system.

Unvalidated analytical dehumidifier models predict optimum ventilation cycle performance for a flow ratio (regeneration to process air flow rate) near 0.8. Testing is currently being performed for a range of 0.6 - 1.0 for parametric study and to verifiy this prediction. Rotation speed can have a 10% to 15% effect on performance. SERI's present test matrix calls for a wide range of speeds. Low speeds will allow us to check the equilibrium properties used in the math model, while high speeds will allow us to check the effective Lewis number (importance of solid-side diffusional resistance to gas-side mass transfer).

In the recirculation cycle process inlet conditions are cooler and somewhat drier than in the ventilation cycle. The regeneration humidity ratio is significantly higher, which is typical. Model predictions of optimum flow ratios near 0.6 show the effect of these changes. Purging the hot, wet wave of air from the dehumidifier can have a 8% to 12% effect on performance. A range of purge angles will be tested and the model predictions verified.

The typical examples cited, although concentrating on steady-state results, will be taken so as to produce useful transient data. Figure 13 shows data taken on a typical dehumidifier geometry. Notice how there is a direct relationship between the rotational speed and transient response. Steady-state values are reached after fifty (50) minutes of testing on this particular device.

Validation of steady-state models is of first importance because of its usefulness in system simulations, numerical investigations of design variation, and in providing feedback to materials and transfer coefficient research. However, accurate transient models are important to more accurately determine "real" system performance and in developing efficient system control strategies. Once again, validated theoretical models are intended to provide industry with tools to design desiccant dehumidifiers with improved confidence.

SYSTEMS ANALYSIS ACTIVITIES

As shown in Fig. 6, all of the experimental data is reduced in a form suitable to validate property, heat and mass transfer, and component and system models. It is at the system modeling level that fundamental research directions can be parametrically analyzed.

The University of Wisconsin has utilized the general system analysis computer code, TRANSYS, to provide more credible estimates of the impact on performance by changing various material, component, and system parameters. To the degree of uncertainty by which we can model the physical processes in a desiccant cooling system, this type of analysis can set performance goals for the material and components that reduce the gap between conventional technologies and the alternatives. For example, the payoff to develop a new material with a certain isotherm shape can be assessed with respect to overall system performance, size reduction, parasitic power requirements, and so on. These tradeoffs can show where the greatest sensitivities are, and hence, define new challenges for the research community. Without this feedback it would be unclear and hard to defend research opportunities.

Systems analysis efforts also provide the opportunity to look at creative ideas, such as hybrid (dehumidification coupled to vapor compression) cooling options or entirely new concepts.

The Florida Solar Energy Center (FSEC) [57] under funding from DOE is investigating a novel cooling concept (Fig. 14) that couples a desiccant dehumidification technique with a night-sky cooling concept. Night-sky cooling works efficiently where buildings can radiate daytime generated heat to the night sky. To do this, the temperature difference between the building surface (usually the roof) and the clear, deep night sky must be as great as possible. This is a cooling technique that works well in the southwest, where air temperatures and solar heat loads are high during the day and where the night sky is very clear and cold.

In places like Florida nighttime air is quite humid, and the sky temperature is generally much closer to that of the buildings roof surface. Researchers at FSEC state that the humid air acts as a blanket between the roof surface and night sky, preventing the roof from radiating heat efficiently to deep space. To overcome the effects of humidity and small temperature differences the roof surface temperature would have to be raised to achieve significant cooling potential.

FSEC's concept uses a desiccant material to adsorb moisture from the air to drive up the temperature of a metal roof radiator. Conceptually, they feel a building would be designed structually with well-insulated concrete block walls with internal cores as the cooling ducts. During the day, the massive walls would store internally generated heat and moisture from the building. At night air would be forced through the ducts carrying the stored heat and moisture to the roof plenum (attic). Within this space, a thin bed of silica gel would adsorb this moisture from the humid air. During the day, the roof would transfer solar radiation to the room plenum and dry the silica gel for the next night's cycle of adsorption. The moisture from the silica gel would be carried away by an air current between the metal roof and the desiccant bed. A radiant barrier under the silica bed would prevent radiant heat from crossing the attic and penetrating into the building itself. FSEC is analyzing this concept to see if the system is technically feasible.

PROJECTIONS FOR THE FUTURE

There are several promising outlooks for desiccant cooling applications in the future. GRI development projects are expected to yield 5- to 20-ton gas regenerated desiccant cooling systems with thermal COPs at an ARI design condition of over 1.1. On a seasonal basis thermal COPs of 2.0 or higher will be achieved by using of the evaporative cooling feature at less humid, off-design conditions. An EER of 35 or higher (0.34 kW/ton) for electric parasitic power requirements of fans, pumps, and wheel motors is projected.

GRI's cooling system package is projected to be larger than the equivalent capacity electric vapor compression units. The installed costs of these systems are also projected to be higher than the electric vapor compression alternative, but if the project goals are met, it should provide attractive payback periods of under three years, depending on climate and local energy prices.

The first gas regenerated desiccant cooling systems introduced to the marketplace will be light commercial units of 5 to 10 tons. These should be commercially available June 1988. Field tests of several prototypes will be carried out during the summer of 1986.

The substantial progress over the past 20 years in developing adiabatic desiccant open cooling (ADOC) cycles, as reviewed in this paper, can be measured by the increases in thermal COP with the time shown in Fig. 15. The state of the art is a thermal COP of 1.0 at ARI standard test conditions.

SERI's planned research in materials and component heat and mass transfer will raise the thermal COP to 1.3 by 1990 while enabling manufacturers to reduce costs and increase market penetration (Table 1) [49]. After 1990, emphasis will shift from the dehumidifier to other components and from the classical ventilation cycle to new cycles currently the subject of academic debate [44,57]. This will enable the target [49] of a thermal COP over 2.0 to be achieved for an ADOC cycle by the year 2000. The ratio of gas to solar collector cost is expected to have increased to a level at which more than half the thermal energy will be solar.

CONCLUSIONS

The feasibility of a solar desiccant cooling system is here today. Many choices and compromises must be made to arrive at an acceptable system design. One design team may be excited by a new material breakthrough, while another team could be satisfied with existing materials but interested in increasing mass transfer to reduce dehumidifier size. Others, whose product line may be evaporative coolers, could be inspired by a promising new cycle using their product and would use existing technology to accomplish the dehumidification. With these conflicting requirements SERI's facilities are generic to most of these concerns. The research is performed on a logical path with experimental data providing input to validate math models intended for industry to use as engineering design tools.

Research goals based on an objective analysis of the material, component, and system's ideal limits provide the rational behind DOE's Cooling Program. As technology transfers are made from the basic research community to industry, further market penetration by renewables is imminent. Going from a research stage to a product line involves risk. The private sector knows that both the industrial and residential consumer will ultimately make their decision based primarily upon the cost-effectiveness of the product or system. Successful businesses know when to take this risk.

ACKNOWLEDGMENTS

The authors would like to thank several individuals who provided direct input and constructive criticisms for this paper; specifically Doug Kosar and Keith Davidson of the Gas Research Institute, Barry Cohen of Thermo Electron, Anthony Fraioli of Argonne National Laboratory, Jim Coellner of American Solar King, Kennard Bowlen of Cargocaire, Al Czanderna, Larry Flowers, Ahmad Pesaran, Kenneth Schultz, and Federica Zangrando from the SERI Desiccant Research Team. The preparation of the manuscript and typing was patiently and meticulously handled by Ms. Deborah Chroniak.

REFERENCES

1. Mitchell, J. W., Energy Engineering, Wiley, New York, 1983.
2. Lunde, P. J., Solar Thermal Engineering, Wiley, New York, 1980.
3. Duffie, J. A., and Beckman, W. A., Solar Engineering of Thermal Processes, Wiley, New York, 1980.
4. Watt, J. R., Evaporative Air Conditioning, The Industrial Press, New York, 1963.
5. Dunkle, R. V., Banks, P. J., and Maclaine-cross, I. L., "Wound Parallel Plate Exchangers for Air-Conditioning Applications," R. K. Shah et al. Compact Heat Exchangers. ASME HTD-10, 1980 pp. 65-72.
6. Pescod, D., "An Advance Inplate Heat Exchanger Geometry Giving Increased Heat Transfer," R. K. Shah et al. Compact Heat Exchangers. ASME HTD Vol. 10, 1980, pp. 73-78.
7. Pennington, N. A.,"Heat-Exchanger for Air-Conditioning Unit," U. S. Patent 2,563,415, 1951.
8. Maclaine-cross, I. L., and Banks, P. J., "A General Theory of Wet Surface Heat Exchangers and its Application to Regenerative Evaporative Cooling," J. Heat Transfer, Vol. 103, 1981, pp. 579-585.
9. Ambrose, C. W., Maclaine-cross, I. L., and Robson, E. B., The Use of Rotary Regenerative Heat Exchangers for the

Conservation of Energy in Buildings. Final Report Project 109. National Energy Research Development and Demonstration Committee, Australian Government, Canberra, 1983.

10. Maclaine-cross, I. L., "Optimizing Rotary Exchanger Matrices for Conditions CLose to Ambient." ASME Paper 83-WA/HT-87, Boston, MA, 1983.

11. Pennington, N. A., "Humidity Changer for Air Conditioning," U. S. Patent 2,700,537, 1955.

12. Munters, C. G., "Inorganic, Fibrous, Gas-Conditioning Packing for Heat and Moisture Transfer," U. S. Patent 3,377,225, 1968.

13. ASHRAE Handbook, Equipment volume. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, 1983.

14. Rush, W. F., and Macriss, R. A., "Munters Environmental Control System," Appliance Engineer 3, 3, 1969, pp. 23-28.

15. Macriss, R. A., and Zawacki, T. S., "High COP Rotating Wheel Solid Desiccant System," 9th Energy Technology Conference, Washington, D. C., 16-18, Feb. 1982.

16. Rousseau, J., "Development of a Solar Desiccant Dehumidifier." Final Summary Report, U. S. Department of Energy Contract No. EG-77-C-03-1591, Washington, D. C., 1982.

17. Huskey, B., et al., Advanced Solar/Gas Desiccant Cooling System. Final report. GRI contract 5081-343-0477. Gas Research Institute, Chicago, 1982.

18. Bullock, C. E., and Threlkeld, J. L., "Dehumidification of Moist Air by Adiabatic Adsorption." Trans. Am. Soc. Heat. Refrig. Air-Condit. Eng. 72, 1966, pp. 301-313.

19. Chi, C. W., and Wasan, D. T., "Fixed Bed Adsorption Drying," AIChE 16, 1970, pp. 23-31.

20. Chase, C. A., Gidaspow, D., and Peck, P. E., "Adiabatic Adsorption in a Regenerator." Chem. Eng. Prog. Symp. Ser. 65, No. 96, 1969, pp. 34-47.

21. Dunkle, R. V., "A Method of Solar Air Conditioning." Mech. Chem. Eng. Trans. Inst. Engrs. Aust. 1, 1965, pp. 73-78.

22. Johnston, R. C. R., "Regenerative Evaporative Air Conditioning with Dehumidification," Aust. Refrigeration, Air Conditioning and Heating 21, 2, 1967, pp. 29-34 and 65.

23. Banks, P. J., "Coupled Equilibrium Heat and Single Adsorbate Transfer in Fluid Flow through a Porous Medium - I Characteristic Potentials and Specific Capacity Ratios," Chem. Eng. Sci., 27, 1972 pp. 1143-1155.

24. Maclaine-cross, I. L., and Banks, P. J., "Coupled Heat and Mass Transfer in Regenerators - Prediction Using an Analogy with Heat Transfer," Int. J. Heat Mass Transfer, Vol. 15, 1972, pp. 1225-1242.

25. Jurinak, J. J., Open Cycle Solid Desiccant Cooling-Component Models and System Simulations, Ph.D. Thesis, Solar Energy Laboratory, University of Wisconsin, Madison, 1982.

26. Nelson, J. S., et al., "Simulations of the Performance of Open-Cycle Desiccant Systems Using Solar Energy," Solar Energy, Vol. 21, 1978, pp. 273-278.

27. Maclaine-cross, I. L., A Theory of Combined Heat and Mass Transfer in Regenerators, Ph.D. Thesis, Department of Mechanical Engineering, Monash University, 1974.

28. Jurinak, J. J., and Mitchell, J. W., "Effect of Matrix Properties on the Performance of a Counterflow Rotary Dehumidifier," J. Heat Transfer, 106, 1984, pp. 638-645.

29. Jurinak, J. J., and Mitchell, J. W., "Recirculation of Purged Flow in an Adiabatic Counterflow Rotary Dehumidifier," J. Heat Transfer, 106, 1984, pp. 369-375.

30. van Leersum, J. G., and Close, D. J., "Experimental Verification of Open-Cycle Cooling System Component Models." Report to the Solar Energy Laboratory, University of Wisconsin, Madison, 1982.

31. Kutscher, C. F., and Barlow, R. S., Dynamic Performance of Packed Bed Dehumidifiers: Experimental Results from the SERI Desiccant Test Laboratory, SERI/TR-252-1529, Solar Energy Research Institute, Golden, CO, 1982.

32. Pesaran, A. A., "Experiments on Sorption Characteristics of Solid Desiccant Materials for Solar Desiccant Cooling Systems," Proceedings of the 1985 ASME/ASES Conference, Knoxville, TN, March 1985, in press.

33. Pesaran, A. A., and Mills, A. F., "Modelling of Solid-Side Mass Transfer in Desiccant Particle Beds," Solar Engineering, Proceedings of the ASME Solar Energy Division 6th Annual Conference, Las Vegas, NV, April 1984, pp. 177-186.

34. Schlepp, D., and Barlow, R., "Performance of the SERI Parallel-Passage Dehumidifier," SERI/TR-252-1951, Solar Energy Research Institute, Golden, CO, 1984.

35. Schlepp, D., "A High-Performance Dehumidifier for Solar Desiccant Cooling Systems." Paper presented at the 19th Intersociety Energy Conversion Engineering Conference, Orlando, FL, August 1983.

36. Jurinak, J. J., Mitchell, J. W., and Beckman, W. A., "Open-Cycle Desiccant Air Conditioning as an Alternative to Vapor Compression Cooling in Residential Applications," J. Solar Energy Eng., 106, 1984, pp. 252-260.

37. Booz-Allen and Hamilton, Inc. Competitive Assessment of Desiccant Solar/Gas Systems for Single Family Residences. Final report, GRI contract 5081-343-0466, Gas Research Institute, Chicago, 1982.

38. Schlepp, D., and Schultz, K. J., "High Performance Dolar Desiccant Cooling Systems," SERI/TR-252-2497, Solar Energy Research Institute, Golden, CO, 1984.

39. Sheridan, J., and Mitchell, J. W., "Hybrid Solar Desiccant Cooling Systems," Proc. 1982 Annual Meeting of AS/ISES, Houston, TX, 1-5 June, 1982.

40. Howe, R. R., Beckmann, W. A., and Mitchell, J. W., "Commercial Applications for Solar Hybrid Desiccant System," Proc. 1983 Annual Meeting of ASES, Anaheim, CA, 5-9 June, 1983.

41. Howe, R. R., Beckman, W. A., and Mitchell, J. W., "Factors Influencing the Performance of Commercial Hybrid Desiccant Air Conditioning Systems." Paper presented at the 18th Intersociety Energy Conversion Engineering Conference, Orlando, FL, August 1983.

42. Cargocaire Engineering Co., "Superaire Cuts Supermarket Cost by Cutting Frost," Leaflet, Cargocaire, Amesbury, MA, USA, 1984.

43. van Leersum, J., Heat and Mass Transfer in Regenerators, Ph.D. Thesis, Department of Mechanical Engineering, Monash University, 1975.

44. van Leersum, J. G., and Ambrose, C. W., "Comparisons Between Experiments and a Theoretical Model of HEat and Mass Transfer in Rotary Regenerators with Nonsorbing Matrices," J. Heat Transfer, 103, 1981, pp. 189-195.

45. Lavan, Z., Monnier, J.-B., and Worek, W. M., "Second Law Analysis of Desiccant Cooling Systems," J. Solar Energy Eng., 104, 1982, pp. 229-236.

46. Maclaine-cross, I. L., "High Performance Adiabatic Desiccant Open Cooling Cycles," J. Solar Energy Eng., 107, 1985, pp. 102-104.

47. Kutscher, C. F., et al., Low-Cost Collectors/Systems Development Progress Report, SERI/RR-253-1750, Solar Energy Research Institute, Golden, CO, 1984.

48. Wilhelm, W. G., "Brookhaven National Laboratory's Low Cost Solar Technology." CAL/SEIA Annual Meeting, Anaheim, CA, September 15, 1984.

49. Kosar, D., Letter to first author, 19 Feb. 1985.

50. Kreith, F., "Innovative Dehumidifier Development for Desiccant Cooling System," Proposal for GRI contract 5084-243-1076, Gas Research Institute, Chicago, 1985.

51. Venhuizen, D., "Solar King's Cooling Gambit," Solar Age 9, 10, 1984, pp. 25-27.

52. Zangrando, F., "Five-Year Research Plan for the Solar Desiccant Cooling Program: FY 1985-1989," Solar Energy Research Institute, Golden, CO, 1985.

53. Lu, C., and Czanderna, A. W., "Applications of Piezoelectric Quartz Crystal Microbalances," Elsevier, New York, 1984.

54. Pesaran, A. A., and Zangrando, F., "Experiments on Sorption Hysteresis of Desiccant Materials," SERI/TR-252-2382, Solar Energy Research Institute, Golden, CO, 1984.

55. Zangrando, F., and Pesaran, A., "Experiments on Sorption Properties of Manganese Dioxide under Isothermal Conditions," Draft Report, Solar Energy Research Institute, Golden, CO, 1984.

56. Schlepp, D., Schultz, K., and Zangrando, F., "Facility Design for Cyclic Testing of Advanced Solid Desiccant Dehumidifiers," SERI/TR-252-2464, Solar Energy Research Institute, Golden, CO, 1984.

57. FSEC, "Novel Cooling Concept Developed," The Florida Solar Energy Center, Solar Collector 10, No. 1, 1985 p. 4.

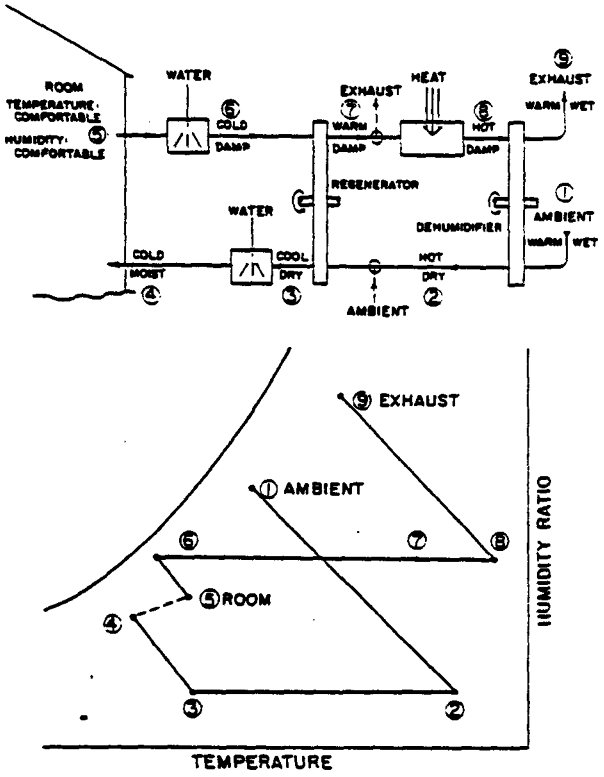


Fig. 1. Schematic and Psychrometric Chart for Pennington's Cycle

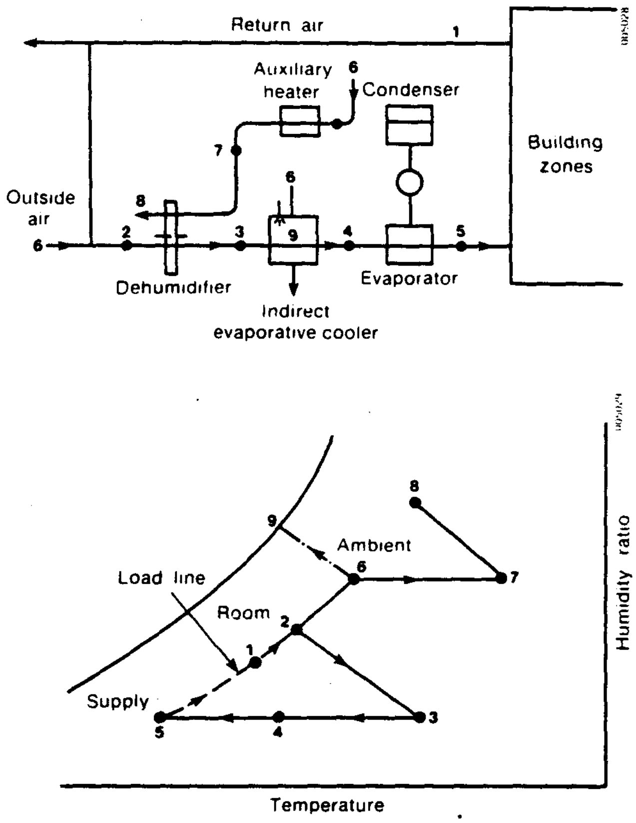


Fig. 2. Schematic and Psychrometric Chart for Hybrid Desiccant Cooling Cycle

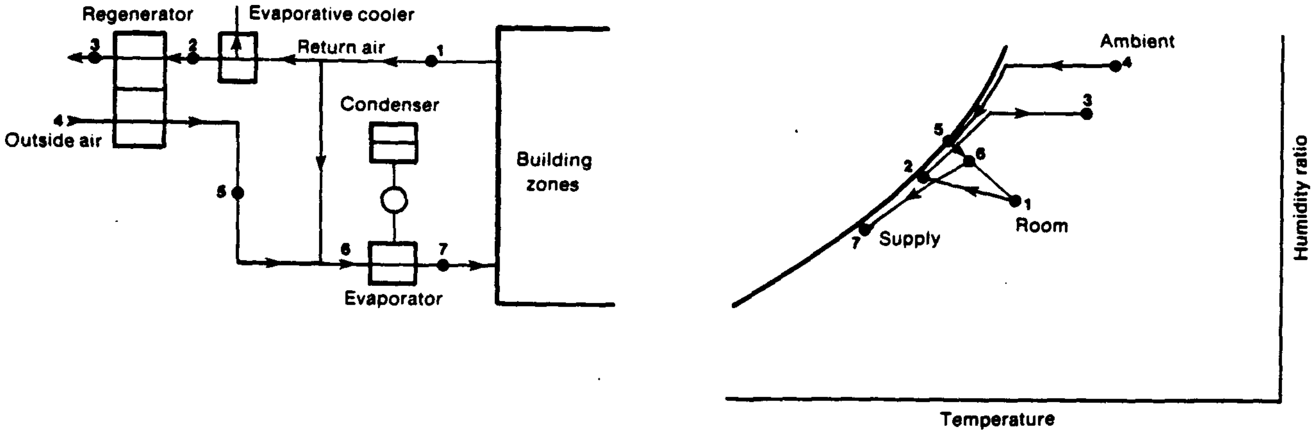


Fig. 3. Schematic and Psychrometric Chart for Hybrid Exhaust Air Regenerative Evaporative Cooling Cycle

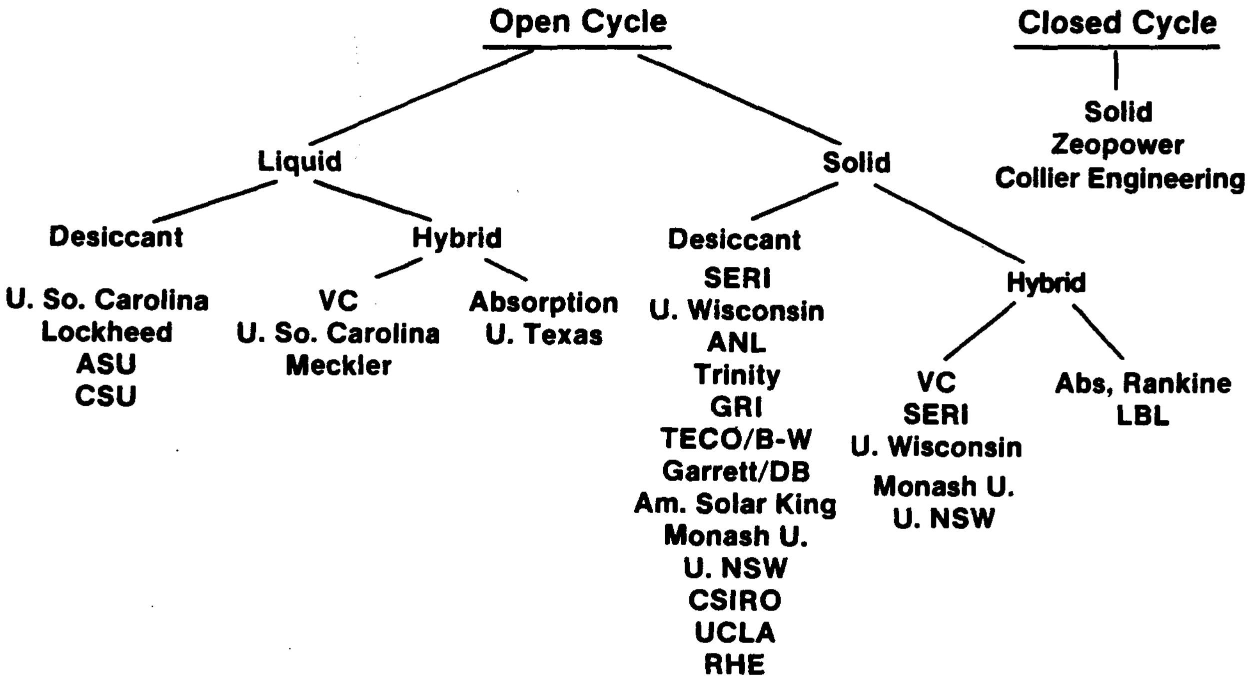


Fig. 4. Organizations Contributing to Desiccant Cooling Research

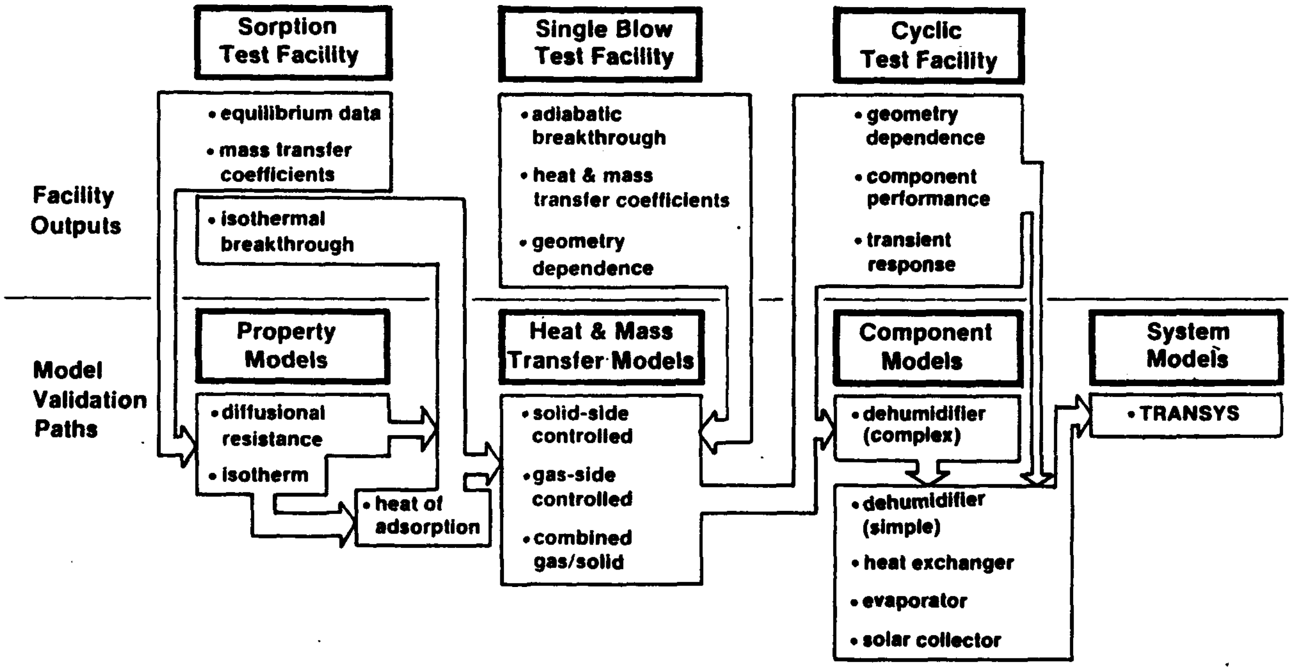


Fig. 5. Research and Development Activities Under the SERI Solar Desiccant Cooling Program

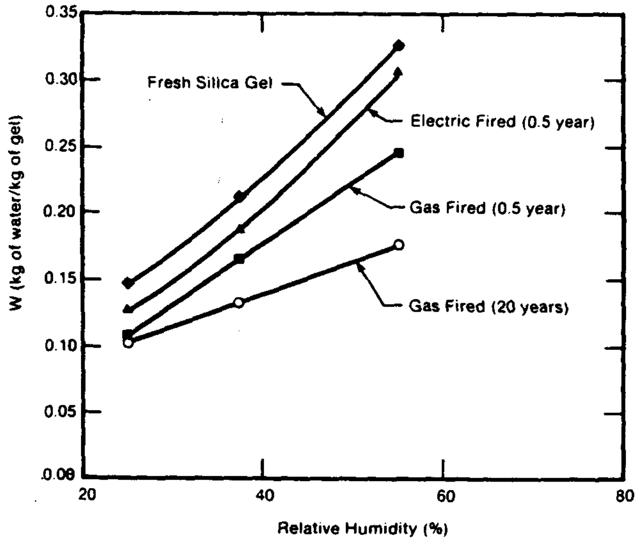


Fig. 6. Degradation of Silica Gel Vapor Pressure Isotherms

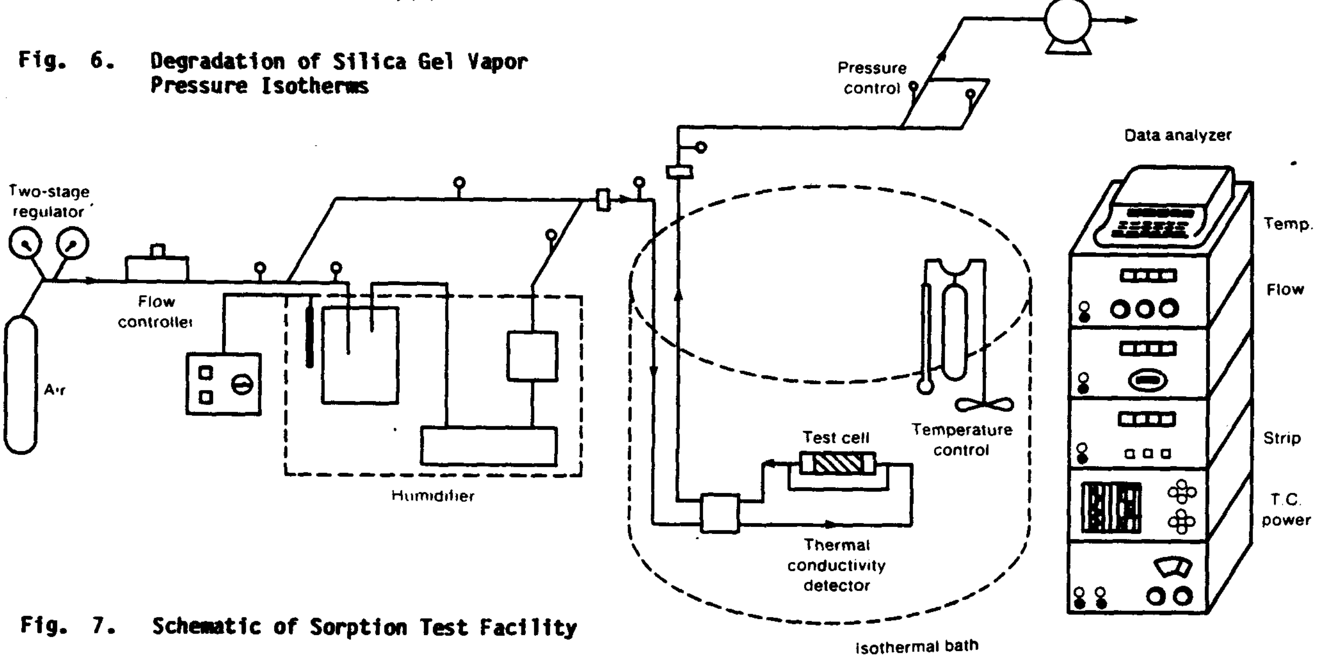


Fig. 7. Schematic of Sorption Test Facility

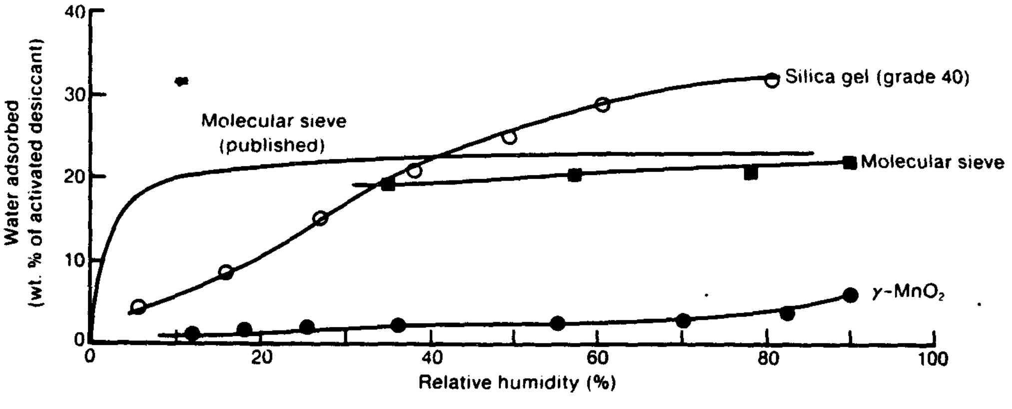


Fig. 8. Typical Adiabatic Breakthrough Data Results from Sorption Test Facility

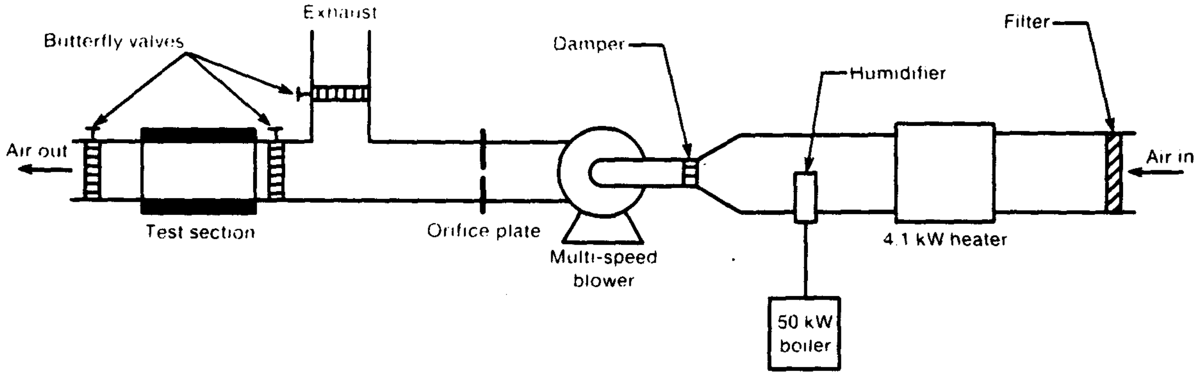


Fig. 9. Schematic of Single Blow Test Facility

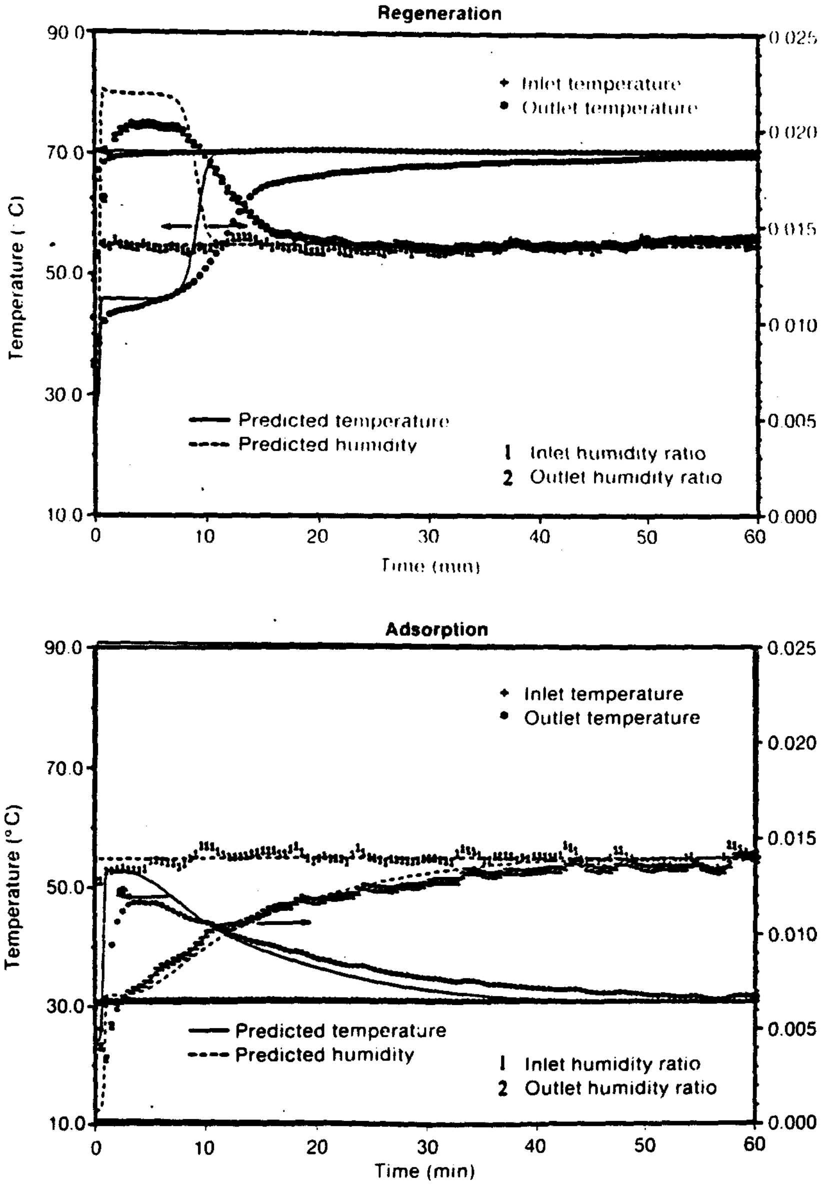


Fig. 10. Typical Results from Single-Blow Test Facility

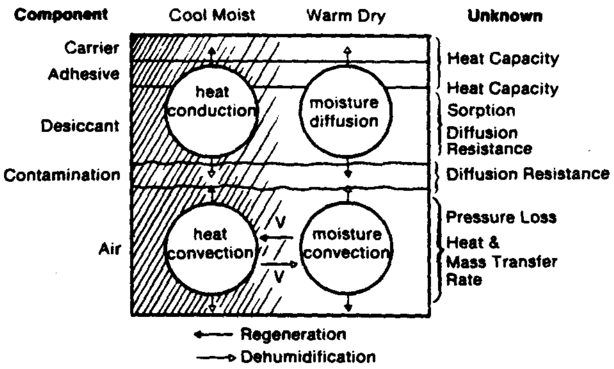


Fig. 11. **Combined Heat and Mass Transfer Processes in a Dehumidifier**

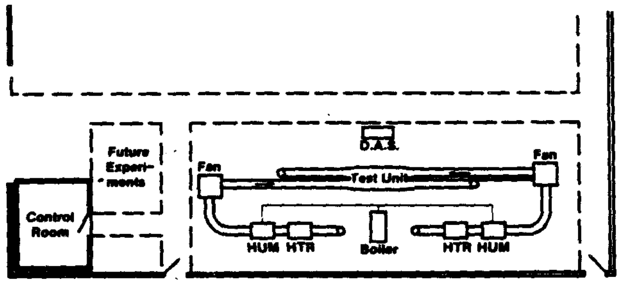


Fig. 12. **Schematic of Cyclic Test Facility**

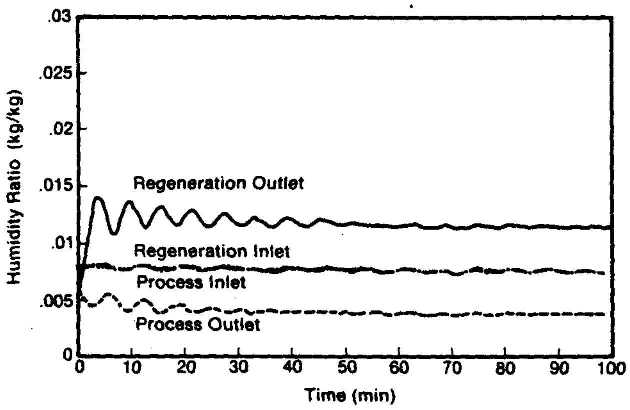


Fig. 13. **Preliminary Results from Cyclic Test Facility**

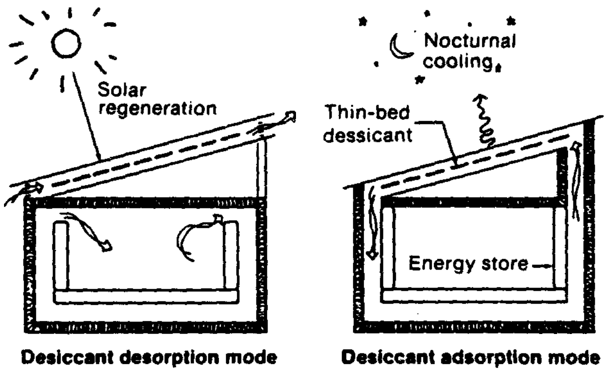


Fig. 14. **A Novel Cooling System Proposed by FSEC**

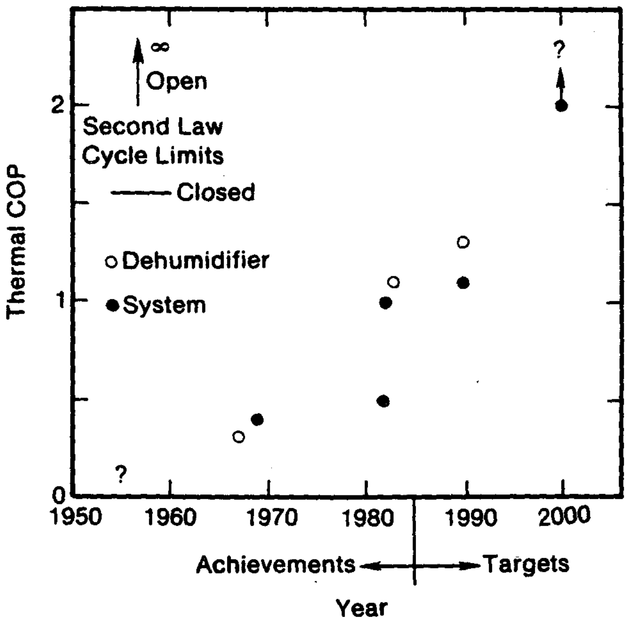


Fig. 15. **Progress in ADOC Cycle Thermal Coefficient of Performance at ARI Conditions**

ENERGY STORAGE AND HEAT PUMPING WITH LIQUID DESICCANTS

H. Robison
CCC/USC Energy Lab
Conway, SC 29526 USA

W. Griffiths
Kathabar Division
Midland-Ross Corporation
New Brunswick, NJ 08903 USA

ABSTRACT

Coastal Energy Laboratory has designed and constructed a second-generation liquid desiccant/vapor compressor hybrid air conditioner and energy storage system based upon two-and-one-half years' experience with a desiccant heat pump in the CCC/USC Energy Lab research house. Unlike the original desiccant heat pump, which uses a shallow well as a heat sink and source and is thus limited to regions where surface water is available, the new desiccant air conditioner has no such restriction; it can discharge heat to the outside air. The unique energy storage capability of the liquid desiccant allows dehumidification and cooling at any time of day or night without simultaneous operation of the electrically powered vapor compressor.

LIQUID DESICCANT HEAT PUMP

A liquid desiccant heat pump that can heat, cool, humidify, and dehumidify, as well as heat domestic water, has been designed, developed, and tested over a six-year period. The desiccant heat pump (DHP) requires no compressor, no gas-fired generator, no condenser or evaporator coils, no expansion valve, no vacuum or pressure systems, no large insulated energy storage, and no freon. The process air stream is the energy transfer medium. Water is the refrigerant. During the cooling cycle, the device uses a salt solution made from water and calcium chloride to cool the process air to a temperature lower than that of a shallow-well heat sink. During the heating cycle, the concentrated salt solution is used to achieve a process air temperature higher than that of the shallow-well heat source. Triethylene glycol, calcium chloride, and mixtures of calcium chloride and lithium chloride have all been tested as absorbers in the machine. Solar energy, waste heat, and off-peak electricity have been used to reconcentrate the desiccant solution so that it can be used again. References (1–4) describe the design and development phase of the research program.

Successful operation of the DHP in the CCC/USC Energy Laboratory research house has demonstrated that a heating cycle utilizing the heat of sorption of a concentrated desiccant solution could be added to a desiccant cooling system, thus creating an open-cycle liquid-desiccant heat pump. See figure 1.

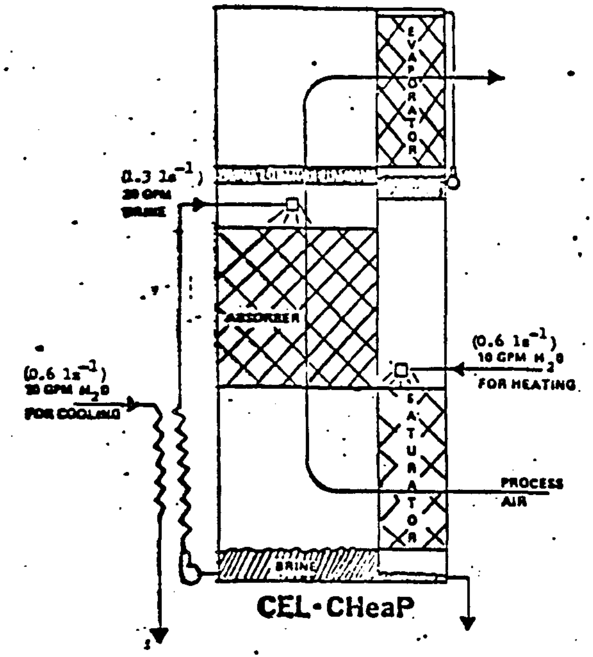
The calcium chloride liquid desiccant was shown to possess a unique capability: the ability to store energy — not as sensible heat but as chemical potential energy — in an uninsulated storage tank with a volume that is an order of magnitude smaller than the uninsulated volumes needed for water or rock-bed storage systems. The spent absorbent solution discharged from the heat pump was reconcentrated in a simple, inexpensive, roof-top solar-collector/reconcentrator.

During the first year of operation, the heat pump heated, cooled, and provided humidity control for the 2800 ft^2 (260 m^2) research house. No back-up equipment was installed. None was needed. References (5–9) describe two-and-one-half years of heat pump operating experience in the research house.

DAILY AND SEASONAL PEAK ELECTRICAL LOAD

In many southern states, the summer electrical load is significantly influenced by the use of electrically powered air conditioners. See figure 2. The energy required to operate air conditioners creates a daily late-afternoon peak load, and these electrical demand curves reach even higher peaks during the hottest summer days. The electrical energy used to drive vapor compression cooling units that create the daily summer electrical peak loads is discharged as waste heat from the systems' condensers to the outside air. Additionally, the heat that is pumped from the building is also discharged as waste heat. If this waste heat were recovered and used to cool the building during the hours of peak demand, a substantial leveling effect on the electrical demand curve could be accomplished.

This reduction of the peak electrical demand would make it unnecessary for many utilities to build additional facilities.



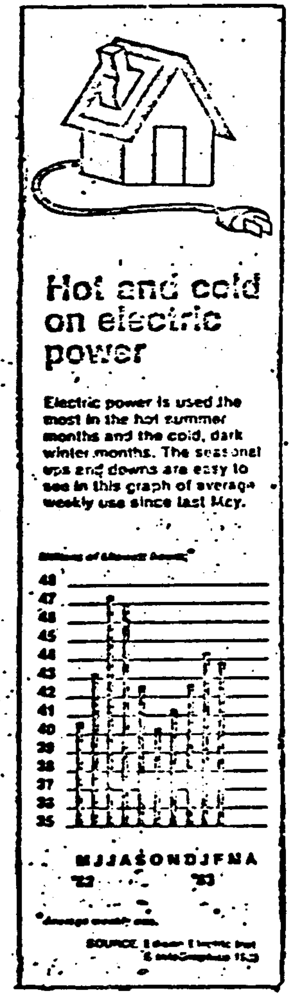
LIQUID DESICCANT HEAT PUMP
OR
OPEN CYCLE CHEMICAL
HEAT PUMP
POWERED BY ANY
LOW GRADE ENERGY SOURCE
- SOLAR ENERGY
- OFF-PEAK ELECTRICITY
- VAPOR COMPRESSOR WASTE HEAT

NO NEED FOR THESE COMPONENTS
USUALLY FOUND IN HEAT PUMPS
COMPRESSOR
CONDENSER COIL
EVAPORATOR COIL
FREON
VACUUM SYSTEM
PRESSURE SYSTEM
GAS-FIRED GENERATOR

ENERGY STORAGE
AS
CHEMICAL POTENTIAL ENERGY
(300 GAL OF CONCENTRATED SOLUTION)
(1136 l)

TWO METHODS
BRINE RECONCENTRATION
- DIRECT BRINE RECONCENTRATION IN ROOFTOP SOLAR COLLECTOR/CONCENTRATOR
- BRINE HEATED IN EPDM SOLAR COLLECTORS AND RECONCENTRATED IN PACKED COLUMN STRIPPER

Figure 1. Liquid Desiccant Heat Pump



Hot and cold on electric power

Electric power is used the most in the hot summer months and the cold, dark winter months. The seasonal ups and downs are easy to see in this graph of average weekly use since last May.

Figure 2 Seasonal Electric Load

SECOND-GENERATION ENERGY STORAGE/COOLING SYSTEM

Using operational data from research conducted by both the Kathabar Division of Midland-Ross Corporation and the CCC/USC Energy Laboratory, Coastal Energy Laboratory has designed and constructed a second-generation liquid desiccant/vapor compressor hybrid air conditioner and energy storage system suitable for electrical load management. Unlike the original liquid desiccant heat pump, which uses a shallow well for a heat sink and source and thus is limited to regions where surface water is available, the new hybrid air conditioner has no such restrictions; it can discharge heat to the outside air.

The second-generation desiccant air conditioner (DAC) was designed to operate in a student activities building at Louisiana State University. During periods of peak demand, the air conditioner is driven by a salt brine that has been concentrated by waste heat from a five-ton vapor compressor cooling system, which operates only during non-peak-demand periods. The waste heat recovered as chemical energy in the form of a concentrated brine is stored for later use in a small uninsulated tank. During the period of peak electrical demand in the afternoon, the vapor compressor is shut down and the DAC assumes the dehumidification/cooling load. An all-plastic crossflow heat exchanger acts as absorber, heat exchanger, and regenerative evaporative cooler. The unique energy storage capability of the liquid desiccant allows dehumidification and cooling at any time of day or night without simultaneous operation of the electrically powered vapor compressor.

The DAC was designed to be a marketable unit that can be powered by concentrating the absorbent solution through the use of any source of low-grade energy such as waste heat, solar energy, or a combination of the two. Additionally, the DAC is capable of utilizing off-peak electricity to concentrate the dilute brine for operation during either the heating or the cooling season.

In order to design the second-generation DAC, a packed-column concentrator was constructed and installed at the research house. See reference (10) for concentrator details. Brine concentration is accomplished by heating the liquid desiccant and allowing the heated liquid to flow down through the packed-column stripper. Outside air is circulated upward through the packing. The heated brine, having a vapor pressure higher than the partial pressure of the water vapor in the scavenger air, desorbs water to the scavenger airstream and is thus concentrated.

Figure 3 shows the concentrator test system. A Fedders rotary compressor with identical copper-nickel liquid-to-freon heat exchanger for condenser and evaporator was installed to test the feasibility of using the waste heat from a vapor compressor system to concentrate the brine. An electrical boiler was constructed and used to calibrate the system and to gain experience in operating the packed column using off-peak electricity as a heat source. The test system also allowed the use of a single-glazed EPDM-rubber solar collector/brine heater.

The energy available from the solar collector and the energy available from the vapor condensation are both available in similar temperature ranges (100–130°F) (38–54°C). When the brine is heated by the waste energy recovered from superheat and from the heat of condensation from the vapor compressor and then passed through the EPDM collectors, the output of the solar collectors drops drastically. The net concentration rate is only slightly greater than the output from either the solar collector or the heat of condensation alone.

The energy from the waste heat from the vapor compressor is available for the cost of a heat exchanger. The solar energy is available for the cost of the EPDM collector. Because the collectors are more expensive than the heat exchanger, it was recommended to LSU that they use only the compressor waste heat and that they not use the solar collectors for this application. To utilize solar energy, it was suggested that LSU draw air from the building attic for the scavenger airstream. As much as 40% of the reconcentration energy can be lost as sensible heat and radiation to the surroundings and to the scavenger airstream. Hot or even warm scavenger air can lower this loss significantly.

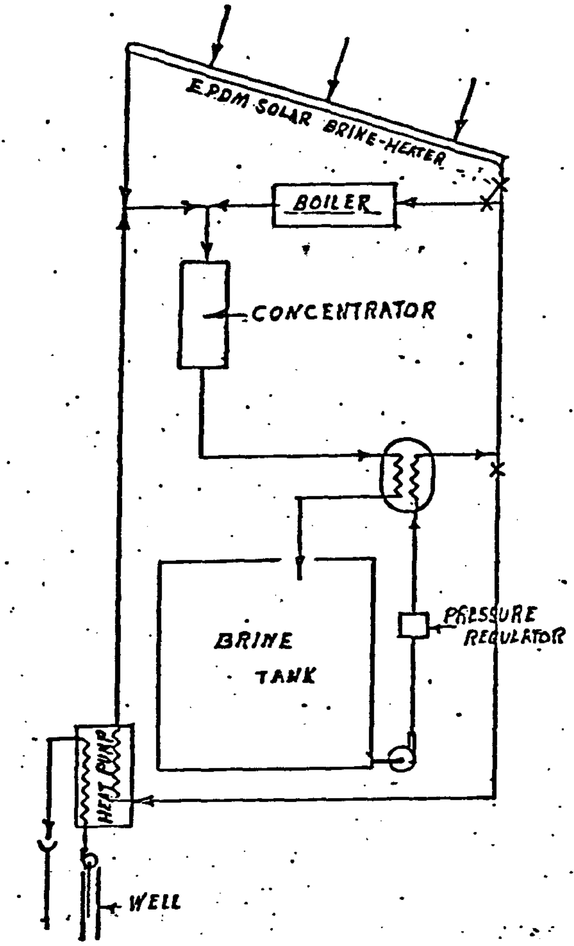


Figure 3. Concentrator Test System

The vapor compressor in the LSU test building was rated at 62,000 BTU hr^{-1} (18.2 kW) at 45°F (7°C) evaporator temperature and 130°F (54°C) condenser temperature. Operating with 120°F (49°C) condenser temperature, the following enthalpy values are obtained:

	BTU/lb	(kcal kg^{-1})
At 220°F (104°C)	135	(75)
Saturated vapor	113	(63)
Superheat	22	(12)
Saturated vapor	113	(63)
Saturated liquid	46	(26)
Condensation	67	(37)

22+67 BTU/lb (49 kcal kg^{-1}) total enthalpy

①② ½ GPM to storage tank (0.03 l s⁻¹)
Liquid level in both units replenished by return lines from storage and controlled by float valves

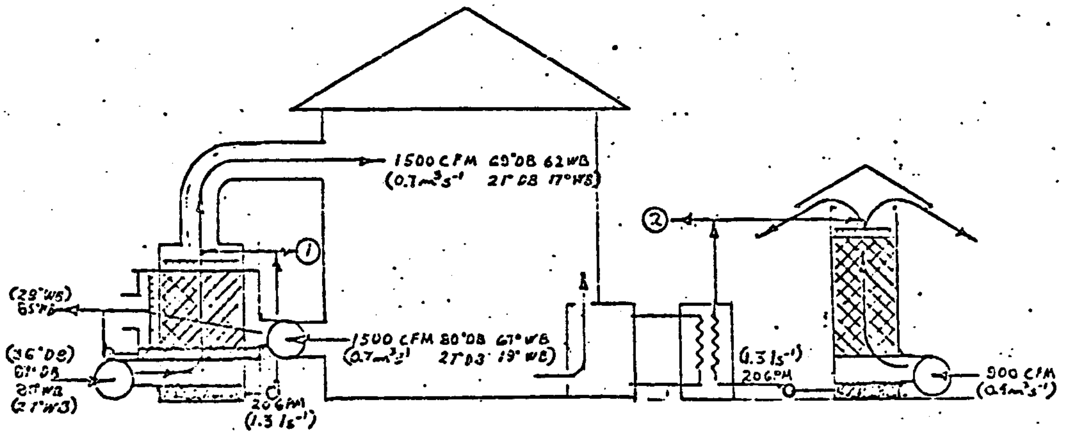


Figure. 4 Louisiana State University System Layout

Seventy-five percent of enthalpy is available at only 120°F (49°C). With reasonable surface area, the heat of condensation should yield 115°F (46°C) brine. The superheat should heat the brine to 120°F (49°C) at discharge from the tube-in-tube counterflow heat exchanger. The peak energy from this source could be as high as 83,000 BTU/hr (24.3 kW).

DESCRIPTION OF LSU-DAC

The LSU-DAC consists of the concentrator, the evaporatively cooled conditioner, the freon heat-recovery brine-heater, the storage tank, and associated plumbing. Figure 4 shows details of the system.

The brine concentrator is a 2 ft x 2 ft x 8 ft (61 cm x 61 cm x 244 cm) column made from fiberglass-reinforced resin. The packing is 2 ft x 2 ft x 3 ft (61 cm x 61 cm x 91 cm) thick. The demister consists of a 6-in (15-cm) layer of ceramic saddles.

The evaporatively cooled conditioner is a 1 ft x 4 ft x 5½ (30 cm x 122 cm x 168 cm) ft fiberglass-reinforced resin column. See figures 5 and 6. The packing is a Midland-Ross "Ekonoblok", a hand-made prototype.

Figures 7 and 8 show the actual cost of the materials for the concentrator and for the conditioner of the DAC. The "Ekonoblok" is a prototype, and no price has yet been set for this unit. The DAC has been delivered and is now in operation on the campus of Louisiana State University in Baton Rouge, LA. No performance data are available at this time.

SUMMARY

Adequate energy storage is an expensive necessity for any solar-driven dehumidification, cooling, or heating system. When dehumidification is necessary, it is usually needed 24 hours a day. While sensible cooling demand is only a few hours out of phase with available insolation, maximum heating demand lags behind maximum insolation by about 18 hours. The liquid desiccant system has been shown to have a unique capacity: the ability to store energy, not as sensible heat but as chemical potential energy. The uninsulated storage volume required is an order of magnitude less than the insulated volumes needed for water or rock-bed storage systems.

A second-generation liquid desiccant energy storage/cooling system was constructed. The design was based upon 2½ years' operating experience with the original DHP, which used a shallow well for a heat sink and source. The new hybrid system, constructed by Coastal Energy Lab in cooperation with Midland-Ross Corp., discharges heat to the outside air. Waste heat from a five-ton (17.6 kW) vapor compressor is used to concentrate the salt brine during off-peak hours. The unique energy storage capability of the liquid desiccant allows the energy recovered to be stored in a small tank in the form of chemical energy for later use. During the period of peak electrical use in the afternoon, the DAC assumes the dehumidification/cooling load, and the vapor compressor is shut down.

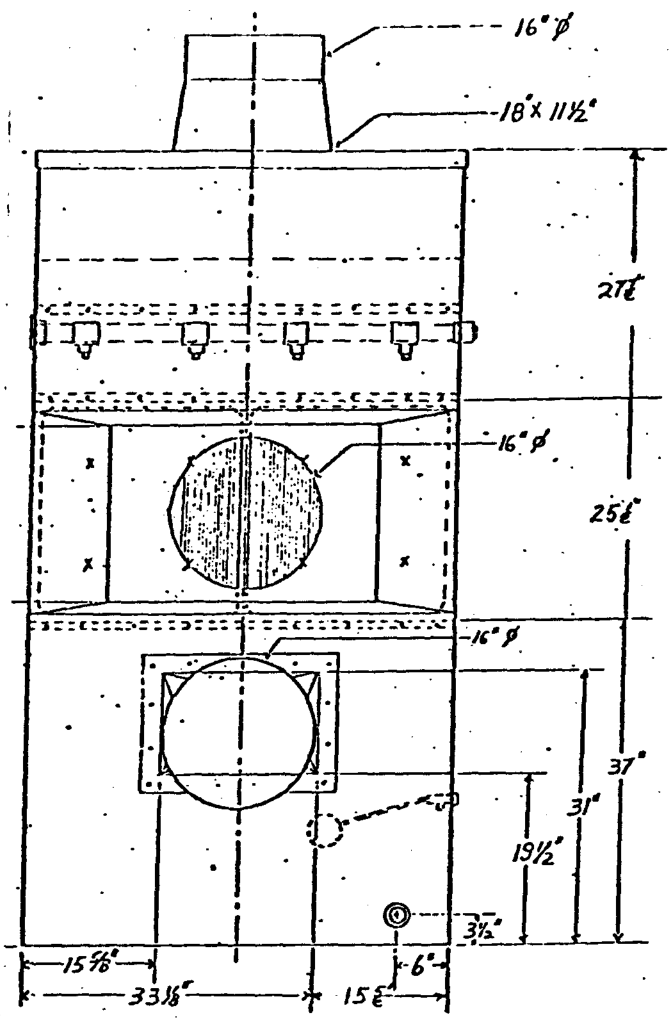


Figure 5. Evaporatively Cooled Conditioner

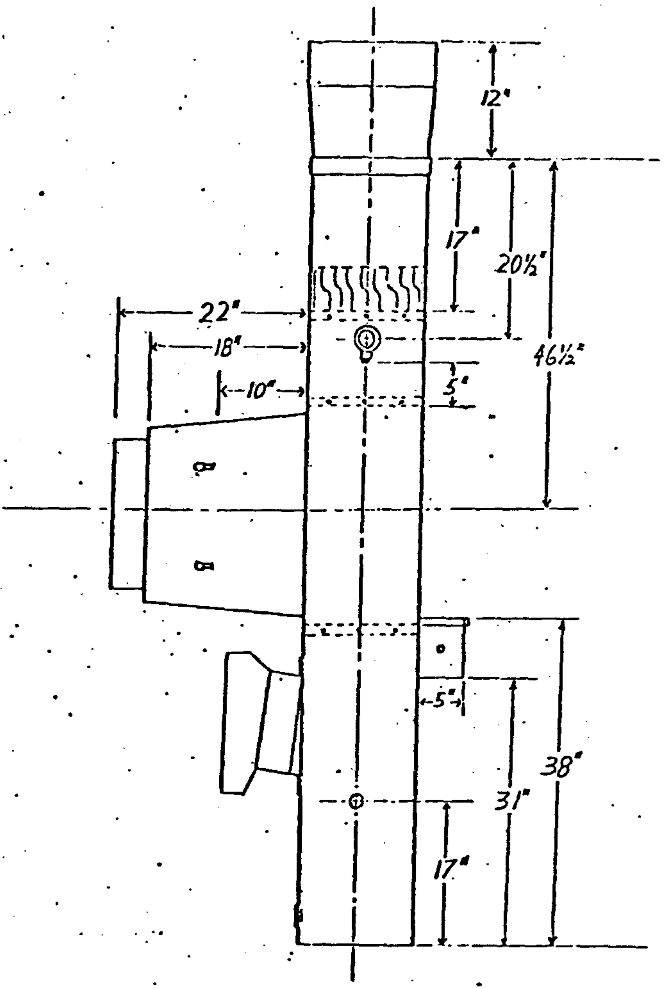


Figure 6 Evaporatively Cooled Conditioner

FINAL CONCENTRATOR DESIGN AND
PROJECTED MATERIAL COST IN SMALL QUANTITIES

Housing-reinforced fiberglass hand layup, 24"x24"x96" Bolton Boat Works, Inc.	$224.00
Rain cover-3/4" exterior plywood 28"x30"	6.00
Rain cover supports (4) 2"x2"x10"	1.00
Locknut fittings (2) SP1023 1½" and (1) SP1025 1½" long body, Hayward Mfg. Co., Inc.	7.20
Brine pump-Power-Flo II SP1705, ½ HP, 1½" inlet and outlet, Hayward Mfg. Co., Inc.	97.00
Scavenger air blower 1140 CFM @3/4" SP, ½ HP, 115 v Dayton Mfg. Co.	229.00
PVC pipe, elbow, nipples, reducers, and adaptors 1½", 1", and 3/4"	8.50
Spray nozzle, wide-angle square spray 1½H290WSQ cast iron female connection, Spraying Systems Co.	25.72
Demister- 2 ft^3 (90#) of 1" unglazed chemical stoneware beryl saddles	60.00
Reinforced fiberglass angle Extron 1"x1"x1/8"x20' Joseph Ryerson & Son	17.50
Freon to brine heat exchanger (6) RXX 140075 DAYB' Noranda Metal Industries	255.00
Tower packing 2'x2'x3' Humi-Kool Celdek CEL09xFC560 Munters Corp.	152.40
Stainless machine screws and nuts	12.00
Float valve 2X525-3/4" 2F009 4"x5" W. W. Grainger	14.50
Scavenger air inlet duct, reinforced fiberglass	15.00
Total	$10[illegible].92

Figure 7 Materials List for Concentrator

FINAL EVAPORATIVELY COOLED CONDITIONER DESIGN
AND PROJECTED MATERIAL COST IN SMALL QUANTITIES

Housing-reinforced fiberglass hand layup, 14"x49"x96" Bolton Boat Works, Inc.	$256.00
Locknut fittings (3) SP1023 1½" Hayward Mfg. Co.	6.00
Brine pump-Power Flo II SP1705, ½HP, 1½" inlet & outlet Hayward Mfg. Co., Inc.	97.00
Process air blower-variable speed 1450 to 2100 CFM @ 1.6" SP, 3/4 HP; Dayton Mfg. Co.	174.34
Spray nozzles (4); Spraying Systems Co.	37.16
Drift eliminator-5"x14"x48" type D-15 Munters Corp.	18.50
Demister 4"x14"x48" Midland Ross Corp.	22.00
Reinforced fiberglass angle, Extron 1"x1"x½"x32' Joseph T. Ryerson & Son	28.00
Econoblok- (2) 24 3/8"x25¼"x13¼" Midland Ross Corp.	Price not available
Stainless machine screws and nuts	15.00
Float valve 2X525-3/4" W.W. Grainger	11.97
Float 2F009 4"x5" W.W. Grainger	2.53
Cooling water recirculating pump McGraw-Edison	37.44
Cooling water spray nozzles and manifold	40.80
Inlet and outlet air plenum, galvanized iron	25.00
Inlet air plenum, reinforced fiberglass	20.00
PVC pipe, tees, bushings, and adaptors 1½"	8.50

Figure 8 Materials List for Conditioner

The development by Midland-Ross Corp. of an all-plastic crossflow heat exchanger that acts as an absorber, heat exchanger, and regenerative evaporative cooler made it possible to design and build the new unit. Operation of the original DHP is limited to regions where surface water is available. The new DAC has no such restrictions; it operates as an air source unit.

Based upon costs of the machines built by Coastal Energy Lab, the costs of three-ton (10.6-kW) machines in small quantitites should be as follows:

Water-source desiccant heat pump	$1085
Air-source desiccant air conditioner	1800
Packed-column concentrator	830

These costs indicated that the desiccant heat pump or the desiccant air conditioner can be economically competitive with conventional heating and cooling systems.

ACKNOWLEDGMENTS

Over the last six years, this work has been supported by the San Francisco Office of DOE, the Solar Energy Research Institute, the Coastal Carolina College of the University of South Carolina, the Coastal Energy Laboratory, the Louisiana Department of Natural Resources, the Coastal Educational Foundation, the Kathabar Division of Midland-Ross Corporation, and over fifty other U.S. corporations.

Technical direction has been furnished by Dennis Schlepp, Kirk Collier, Bill Griffiths, Stan Slabinski, and Karl Finch.

Sam Houston, P. Gandhidasan, Howard Harris, and H. Robison conducted this research.

REFERENCES CITED

1. Robison, H. I., "Liquid Sorbent Solar Air Conditioner," Alternative Energy Sources, Veziroglu, T. N., ed., Hemisphere Publishing Corp., New York, NY, 1978.

2. Robison, H. I. and Houston, S., "Passive Solar Heat Pump," Proceedings of the 4th National Passive Solar Conference, Kansas City, MO, Franta, G. F., ed., American Section of International Solar Energy Society, Newark, DE, 1979.

3. Robison, H. I. and Houston, S. H., "Thermochemical Energy Storage for Heating and Cooling," Proceedings of Solar Energy Storage Options, San Antonio, TX, McCarthy, M. B., ed., Trinity University, San Antonio, TX, 1979.

4. Robison, H. and Houston, S., "Solar-powered Saline Sorbent-Solution Heat Pump/Storage System," Alternative Energy Sources II, Veziroglu, T. N., ed., Hemisphere Publishing Corp., New York, NY, 1980.

5. Gandhidasan, P. et al., "Analysis and Simulation of Forced-Flow Solar Collector/Regenerator," Passive Cooling, Bowen, A., Clark, E., and Labs, K., eds., American Section of International Solar Energy Society, Newark, DE, 1981.

6. Robison, Harry I., Open-Cycle Chemical Heat Pump and Energy Storage System, Final Report for SERI, Subcontract No. ZE-O-9185-1, CCC/USC Energy Lab, Conway, SC, 1982.

7. Robison, Harry and Harris, Howard, "Year-Round Operational Test Results of a Chemical Heat Pump Powered by Waste Heat, Off-Peak Electricity, or Solar Energy," Energex '82, Curtis, F. A., ed., Solar Energy Society of Canada, Regina, Saskatchewan, 1982.

8. Robison, Harry I., "Operational Experience with a Liquid Desiccant Heating and Cooling System," Proceedings of the 18th IECEC, American Institute of Chemical Engineers, New York, NY, 1983.

9. Robison, Harry I., Open-Cycle Chemical Heat Pump and Energy Storage System, Final Report for DOE, Contract No. DE-AC03-82SF11657, CCC/USC Energy Lab, Conway, SC, 1983.

10. Robison, Harry, Griffiths, William, and Arnas, Ozer, "Liquid Desiccant/Vapor Compressor Hybrid Air Conditioner and Energy Storage System for Load Management," Solar Energy in the Southeast, Colvin, D. P., ed., North Carolina Solar Energy Association, Raleigh, NC, 1984.

ENERGY-INTEGRATED DESICCANT HVAC SYSTEM APPLICATIONS

Presented by
Gershon Meckler, P.E.
Gershon Meckler Associates, P.C.
40 Rector Street
New York, N.Y. 10006
(212) 693-0366

DOE/SERI Desiccant Cooling Workshop
Chattanooga, Tennessee

June 10-12, 1986

ABSTRACT

Energy-integrated desiccant HVAC systems have been designed and incorporated by the author in several large building applications to shift a major portion of the air-conditioning load from standard vapor compression refrigeration to more efficient selective-energy systems. Latent cooling is separated from sensible cooling to remove all or most of the dehumidification task from the electrically driven vapor compression machine. A moisture-absorbing desiccant is used to dehumidify. Energy required by the desiccant dehumidification process is provided by (a) nonrefrigerated cooling tower water or chilled water at an elevated temperature (55-65 F), which removes the latent heat released by the absorption process; and (b) an integrated selective-energy system, which provides heat to regenerate the desiccant from a cogenerator, a heat pump, off-peak electric utility energy via thermal storage tanks, and solar energy where appropriate. Four applications are described: the 110,000-sq ft Monroe County Courthouse in Stroudsburg, Pa.; the 765,000-sq ft Social Security Administration Metro West facility in Baltimore; a 168,000-sq ft addition to the Veterans Administration Hospital in Washington, DC.; and the Science Museum of Virginia Planetarium and Exhibit Halls.

INTRODUCTION

Historically and currently energy for air conditioning has been supplied almost exclusively by utility-generated electricity energizing vapor compression refrigeration. Alternative methods and energy sources are being actively sought by consumers seeking lower energy costs and by electric utilities who economically cannot continue to meet increased demand by building new generating capacity.

Over the past 20 years, the author has developed, designed, and installed energy-integrated desiccant HVAC systems responsive to this need that significantly reduce utility energy requirements. This approach can eliminate the need for using refrigeration to dehumidify by shifting latent cooling to a desiccant dehumidification system and regenerating the desiccant with heat from a selective-energy system incorporating appropriate combinations of the following: cogeneration, heat pumping, solar energy, thermal storage, and evaporative cooling.(1) Four installations are described below: the 110,000-sq ft Monroe County Courthouse and Government Center in Stroudsburg, Pennsylvania; the 765,000-sq ft Social Security Administration Metro West facility in Baltimore; the 168,000-sq ft nursing home/clinic addition to the Veterans Administration Hospital in Washington, D.C.; and the Science Museum of Virginia Planetarium and Exhibit Halls.

ROLE OF DESICCANT DEHUMIDIFICATION

Desiccant dehumidification is not a new process. In the past, however, it was limited to special-purpose applications where stringent moisture control was essential. It was not considered competitive for general use because of the utility energy required to dry the desiccant for reuse. New techniques developed by the author increase the efficiency of desiccant dehumidification in several ways: (a) by directly reducing the quantity of energy required, through a heat recovery technique on the regenerator; (b) by reducing the temperature level of the heat required, so that the COP of the regeneration process is increased and less energy is required for a given amount of regeneration; and (c) by integrating cogenerated heat, solar energy, and off-peak utility energy in an efficient selective-energy system providing heat for desiccant regeneration.

Lowering the temperature level of regeneration has another important consequence: Usable heat sources now include low-temperature solar energy (130-140 F), which is available via flat plate collectors, and recoverable waste heat, such as cogenerated heat and latent heat from the desiccant conditioner. With the broader range of energy sources, selective-energy systems can effectively minimize utility energy consumption by using other sources optimally before switching to purchased energy. A thermal storage tank integrates the heat sources used to support desiccant regeneration. A storage tank also permits night use of electric resistance heating, to take advantage of off-peak rates and to ease the pressure on utility generating capacity during the annual demand peak - the summer air conditioning season.

Based on efficient integration of systems overall, desiccant dehumidification provides a competitive alternative to conventional air conditioning.

DESICCANT SUBSYSTEM

Desiccants are of two types, solid and liquid. Both are used in the examples described, either in solution form or as a salt impregnated on a honeycomb wheel. The desiccant used is lithium chloride, which is a nontoxic, noncorrosive, bactericidal, inorganic material often used in hospitals.

The liquid desiccant subsystem includes two chambers: a conditioner (dehumidifier) and a desiccant regenerator (concentrator). Outside air passes through the conditioner where moisture is absorbed by a spray of cool desiccant. The moisture-laden desiccant cycles to the regenerator to be dried for reuse. The desiccant is heated and sprayed into the regenerator, where the moisture is removed in the warm, dry, exhaust airstream passing through the chamber. (Heat in the exiting airstream is recovered and used to preheat the air entering the regenerator.)

In the solid desiccant rotary wheel system, a desiccant-impregnated honeycomb wheel rotates within a cylindrical casing. The cylinder is divided into a dehumidifier section and a regenerator section, through which the wheel continuously rotates. Separate airstreams pass through the two sections in opposite directions. One is the outside airstream, from which moisture is removed by desiccant absorption; the other is the warm, dry building exhaust air, which carries moisture out of the wheel, regenerating the desiccant.

The desiccant dehumidification process requires cooling to remove the latent heat released by the moisture absorption process, which can be accomplished using nonrefrigerated cooling tower water, as well as heating to dry the desiccant. Subsequent sensible cooling can be at an efficient elevated temperature (55 F instead of 42 F) since it is not necessary to chill more deeply to condense out moisture (Figure 1).

MONROE COUNTY COURTHOUSE

In order to minimize energy costs at the Monroe County Courthouse in Stroudsburg, Pennsylvania, an energy-integrated HVAC system that incorporates desiccant dehumidification (Figure 2) replaced an old steam heating system for the 19th-century courthouse and its adjoining new wing. The three-story, reinforced concrete structure contains office space, courtrooms, judges' chambers, and hearing rooms. The facility totals 110,230 sq ft, with 77,580 in the old courthouse and 32,650 in the addition.

Heat is used in place of refrigeration to energize latent cooling/ dehumidification. This is made possible by shifting the dehumidification task to a moisture-absorbing desiccant that requires heat for desiccant regeneration.

Selective-Energy System and Thermal Storage

Heat to dry the desiccant is provided by a selective-energy system that blends off-peak night electric energy, cogenerated heat, and solar energy. An 8,000-gallon thermal storage tank that is maintained at 150-180 F is heated at night by electric resistance heaters, to take advantage of off-peak utility rates, and during the day by solar energy (now being added) and back-up heat from a 175-kW diesel cogeneration system. Both the cogenerator and the solar system reduce electric demand, and the electric heaters shift demand off-peak.

With the addition of the solar collectors, which will provide about 35% of the total heat required for regeneration, the cogeneration system will be used only on cloudy days and when humidity exceeds design levels. When it operates to provide regeneration heat, the system simultaneously generates electricity to power sensible cooling. Cogeneration reduces the collector area requirement and also provides standby power for use during utility power outages.

Located in a central plant are the thermal storage, diesel cogeneration, and desiccant dehumidification systems. All sensible cooling and heating are done at terminal unitary heat pumps, which operate at an efficient 55 F evaporative temperature since there is no requirement to chill deeply to dehumidify by condensation.

The thermal storage tank serves as a heat source for the unitary heat pump system in winter and for the desiccant system in summer. Storage provides heat to the terminal unitary heat pumps via a closed-loop water system that links the heat pumps and has its own closed-loop storage tank.

Unitary Heat Pumps and Thermal Storage

The terminal unitary heat pumps are linked to both the thermal storage tanks and the cooling tower. Unitary heat pumps are small, terminal reverse-cycle air conditioners that recirculate air locally and heat or cool as needed. They have a self-contained heating and cooling capability due to a reversing valve that regulates the flow of hot and cold refrigerant gas.

When dehumidification is by condensation in a standard vapor compression refrigeration system, a central refrigeration plant is more efficient than terminal unitary heat pumps. However, when dehumidification is by desiccant absorption and the heat pumps can operate at a higher dry-coil temperature, their overall efficiency increases significantly. When unitary heat pumps are combined with desiccant dehumidification, the cooling COP of the unitary heat pumps increases by about 25%.

The terminal heat pumps are joined by a closed water loop that is maintained at 70-90 F. The heat pumps draw heat from or reject heat to the closed water loop; some can be in a heating mode while others are cooling or off. An example is a winter day when solar gains through south-facing windows require heat removal, while north wall losses require heating. In this case, heat is simultaneously added to and drawn from the water loop by different unitary heat pumps.

When the temperature of the water loop drops below 70 F, it draws heat from the solar system or the high-temperature storage tank. When the temperature of the water loop rises above the upper limit (80-90 F), heat is rejected via the cooling tower.

With a unitary heat pump system, there is no need for both hot and chilled water piping. Additional savings in pipe and space are obtained by using the return side of the system's water loop as the fire sprinkler system (2).

Desiccant Dehumidification

A liquid desiccant is used in the Monroe County system. Moisture absorption and desiccant regeneration are as described in the "Desiccant Subsystem" section above.

Air Distribution in Desiccant/Terminal Cooling System

The minimum quantity of outside air required for ventilation is distributed at a constant volume from the central plant to terminal unitary heat pumps. This minimum air distribution is practical when a desiccant is used, as it is not in conventional systems, since the air is drier and the ventilation quantity is adequate to handle the space humidity load.

The unitary heat pumps mix the dry air with a constant volume of local recirculated air, cool the air sensibly and supply it in a constant volume to the rooms. The constant flow maintains uniform air flow patterns and ventilation to maintain healthful air quality throughout the occupied spaces.

SSA METRO WEST FACILITY

The Social Security Administration Metro West complex in Baltimore, designed in 1976, posed unusual environmental control problems for which desiccant dehumidification provided a particularly appropriate solution with the lowest life-cycle bid. The facility is very large - 786,000 sq ft of conditioned space with long distribution runs - and has heavy internal heat gains from computer terminals.

In any selected occupied area, the facility had to be able to handle 13 Btu/sq ft of heat gain from electronic equipment such as computer terminals (eight times the previous standard for office equipment). Although only 40% of the total area would receive this load, the precise location was unknown at the time and subject to changing space use; therefore, flexibility to handle the equipment throughout the facility was essential. The refrigeration plant could be sized for the specified heat gain from electronic equipment (13 Btu/sq ft in only 40% of the total area); but due to the unspecified location of the heat concentrations, the distribution network had to be designed to handle the maximum load throughout the facility.

Since the facility is very large, a key cost consideration was the size of the distribution ductwork. The use of central-plant desiccant dehumidification, with terminal sensible cooling (Figure 3), permitted significant savings in this area: Desiccant absorption dries the air very deeply, so it was possible to reduce primary-air distribution to the minimum quantity of outside air required for ventilation (0.1 cfm/sq ft) and to reduce ductwork and fan power proportionately. That quantity of primary air, dried to 31 gr/lb (grains of moisture per pound of dry air) takes care of the internal humidity load. One direct result was a 30% reduction in the floor-ceiling spaces between occupied floors. The dehumidified air is distributed uniformly on a modular basis by way of air cells in the cellular steel floor deck, which is connected wherever needed to terminal fan-coil units, as shown in Figure 4.

Sensible cooling is then handled at the fan-coil terminals, which mix the primary dehumidified air with recirculated room air and cool with coils circulating 55 F chilled water. The 55 F chilled water is distributed on a modular, uniform basis through an integrated chilled water/sprinkler piping system (2). This arrangement achieves flexibility for changing space use: When more cooling capability is needed, additional fan-coil units can be plugged into both the modular sprinkler piping and the modular air cells in the cellular steel deck.

With a large latent cooling load, and an unusually large sensible load due to heat from electronic equipment, the size and efficiency of the refrigeration plant was also a major cost concern. Desiccant dehumidification permitted a 33% reduction in the size of the refrigeration equipment (compared with a standard air-conditioning system), based on the shift of a major part of the latent cooling load from the chiller to the desiccant system. The efficiency (COP) of chiller operation was improved also, by 20%, based on higher-temperature cooling (55 F). Consequently, 20% less utility energy is required for a given amount of sensible cooling. In addition, a chilled water thermal storage tank permits a shift of electric utility demand from day to night, when rates are cheaper.

The SSA facility has a two-stage desiccant plant (Figure 5). There are two desiccant conditioners, the first cooled by nonrefrigerated cooling tower water, and the second by 55 F chilled water. When outside air conditions are 94.7 F and 124 gr/lb absolute humidity, and cooling tower water is at 85 F, outside air leaves the first conditioner at 94 F and 42 gr/lb. After second-stage conditioning with 55 F latent cooling, this outside (primary) air is distributed at 75 F and 31 gr/lb. When outdoor humidity drops low enough so that first-stage nonrefrigerated conditioning dries to this level, the second conditioner is deactivated.

When this system was designed, the technology of low-temperature desiccant regeneration had not been developed. A steam boiler provides regeneration heat.

VA HOSPITAL ADDITION

The VA Hospital's recently completed 168,000-sq ft addition in Washington, D.C., includes a 120-bed nursing home unit that requires 24-hour air conditioning and a daytime clinic facility. Based on design data, the electric utility energy required for cooling is greatly reduced by removing dehumidification from the refrigeration system to a desiccant dehumidification system and by using cogenerated electricity to power the 220-ton chiller plant approximately 50% of the time during the cooling season. On an annual basis, it is estimated that cogenerated electricity will provide 35% of the HVAC system's total electrical requirements. The system conditions 19,000 cfm of outside air by day and about half that amount at night.

Heat for desiccant regeneration comes from solar energy, about 45%, and a diesel engine cogenerator, 55% (1). When operating to provide recovered heat, the 350 kW diesel engine simultaneously generates electricity to power the desiccant system's fans and pumps as well as the chiller plant. Nonrefrigerated cooling tower water removes the latent heat released during the moisture absorption process.

The hospital addition's HVAC system (Figure 6) includes the first application of desiccant dehumidification using low-temperature solar energy (140 F) to regenerate the desiccant. Some 4,500 sq ft of flat plate collectors will produce approximately 4.5 million Btu per day at 140 F in July and August. The desiccant regeneration task requires about 10 million Btu per day on average over the cooling season, 7 million during daytime, and 3 million for night operation of the nursing home. An additional 2 million Btu used to preheat domestic hot water also comes from the solar/cogeneration selective-energy system.

In order to evaluate the effectiveness of the solar/cogeneration/desiccant dehumidification HVAC system and to establish the most efficient operating plan for the selective-energy system, an energy management computer system (EMCS) has been installed and equipped to perform data acquisition, analysis, and reporting. Initially the selective-energy system is configured to operate as follows:

1. All available solar energy will be used.
2. The diesel engine will not operate at night.
3. On sunny days, heat sources for desiccant regeneration (10 million Btu) and preheating hot water (2 million Btu) will be:

   4.5×10^6 Btu - solar
   7.5×10^6 Btu - cogenerator (operating 5 hours)

   12.0×10^6 Btu - 24-hour requirement for desiccant regeneration and preheating hot water.

4. On sunless days, the cogenerator, which produces 1.5 million Btu per hour, will operate 8 hours to meet the 12 million Btu requirement.
5. Day/Night Btu Usage and Sources:

	Day Usage	Night Usage
REQUIRE-MENT:	7×10^6 Btu - desic. regen. $2 \text{ X } 10^6$ Btu - preheat hot water	3×10^6 Btu - desic. regen.
	9×10^6 Btu	3×10^6 Btu
SOURCES: (sunny days)	4.5×10^6 Btu - solar 4.5×10^6 Btu - cogeneration	3×10^6 Btu - stored cogenerated heat
	9.0×10^6 Btu	3×10^6 Btu

The system incorporates three 10,000-gallon hot thermal storage tanks: one low-temperature solar tank that can store 1 million Btu at 140 F and two high-temperature tanks that have the capacity to store 6 million Btu of 170 F cogenerated heat.

VA System Dehumidification Process

Outside air - the minimum quantity required for ventilation - is dehumidified in the central plant in two stages in two rotary wheel-type dehumidifiers composed of a honeycomb structure mounted within a cylindrical casing. The honeycomb structure is impregnated with lithium chloride. The cylinder is divided into a dehumidifier section and a regenerator section. Separate airstreams pass through the two sides in opposite directions. One is the outside airstream, from which moisture is removed by desiccant absorption; the other is the warm, dry building exhaust air, which carries moisture out of the wheel, regenerating the desiccant.

Outside Air (design condition)	1st-Stage Dehumidifier	2nd-Stage Dehumidifier	Nonrefrigerated-Water Cooling Coil
Entering condition:	93 F DB, 105 gr/lb		
Leaving condition:	84 F DB, 78 gr/lb	104 F DB, 56 gr/lb	91 F DB, 56 gr/lb

The exhaust air flowing through the dehumidifier wheels in the counter direction to carry off moisture for desiccant regeneration enters the first-stage conditioner at 81 F DB, 70 gr/lb and leaves it at 90 F DB, 97 gr/lb. Before entering the second-stage dehumidifier, the exhaust air is heated first by heat recovered from the exhaust airstream leaving the second dehumidifier, to 97 F DB, then in series by solar energy and cogenerated heat. This air enters the second dehumidifier wheel at 130 F DB, 97 gr/lb, and leaves at 107 F DB, 126 gr/lb.

VA System Sensible Cooling and Air Distribution

An all-air system was a design requirement at the hospital addition. Therefore all sensible cooling of the dehumidified air leaving the desiccant wheels is done in the central plant by the chiller powered primarily by cogenerated electricity and off-peak utility electricity. The primary air is delivered in variable volume to fan induction terminals (3) where it is mixed with recirculated room air for a constant volume delivery at the required temperature. Only perimeter terminals require heating coils, which are heated from the hot thermal storage. With the latent cooling load handled by the cooling tower rather than the chiller, and with the solar/cogeneration integrated heat source plus utility consumption peak-shaving, the system is expected to save up to 30% of total purchased energy costs compared with a refrigeration/condensation system.

SCIENCE MUSEUM OF VIRGINIA, RICHMOND

The Science Museum system (Figure 7) conditions air for a planetarium, exhibit halls, workshops, offices and mechanical equipment room. The principal design challenge was efficient handling of the humidity and cooling loads generated by people passing through the museum as well as from outside air. The summer load profile makes this clear: Dehumidification (latent cooling) is 40% of the total summer conditioning load; another 24% is people-related sensible cooling. The well-insulated building envelope, consisting largely of the windowless planetarium dome, represents only 23% of the summer load, and the luminaires another 13%.

Not only is latent cooling the principal load component, the load is also highly variable due to the varying occupancy. A summer influx of 300 people into the planetarium can produce a humidity rise of 9.6 gr/lb as well as a 13.5 degree temperature rise. Dehumidifying by condensation in a 42 F refrigerated system would require extensive reheat, to maintain comfort in the planetarium when occupancy is light and there is little people-generated heat. Given these conditions, and a requirement for a constant volume delivery of air to the space, desiccant dehumidification is a very significant factor in reducing energy consumption. Both latent cooling and sensible cooling are done at a higher temperature (55 F) which requires less utility energy for a given amount of cooling.

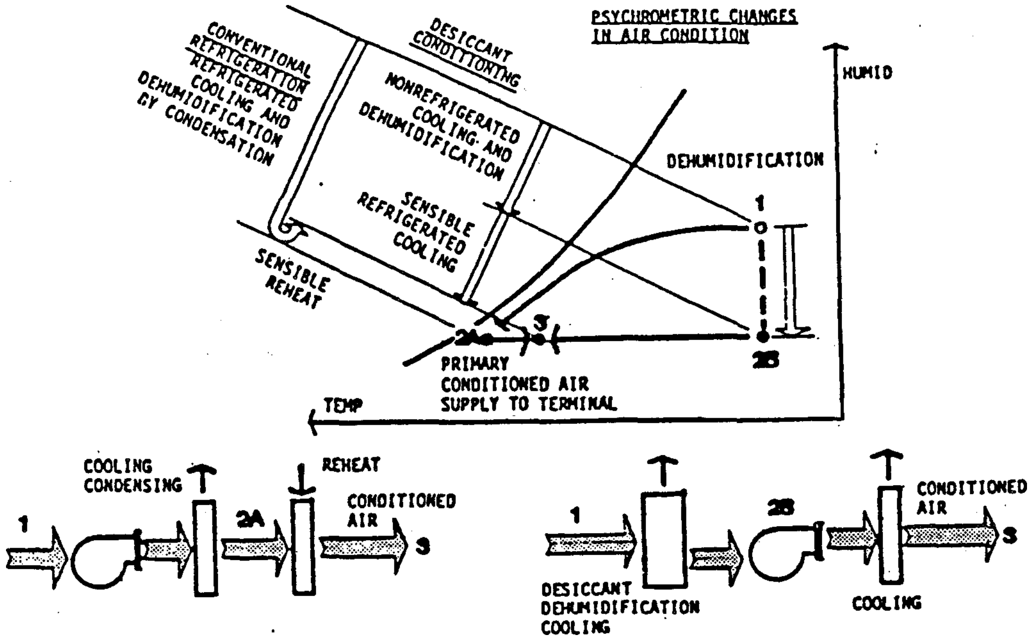
The heat for desiccant regeneration (and winter heating) and the chilled water for cooling are provided by an integrated selective-energy system which incorporates the following: (a) a heat pump that pumps latent heat released in the conditioner to the thermal storage tank and, simultaneously, produces chilled water that meets one-fourth of the chilled water requirements; (b) a cogenerator (with 75 kW gas engine) that starts up when the heat pumped energy is insufficient, at which time cogeneration provides both electricity to run the heat pump, and jacket-and-exhaust heat to charge the high-temperature storage tank; (c) a chiller operating at a 5.2 COP, due to the thermodynamic efficiency of high-temperature cooling; and (d) solar energy (planned addition).

In summer outside air is cooled and dehumidified by a preconditioning coil that handles a substantial portion of the outside air conditioning load. Whenever the occupancy humidity load is high, the preconditioned air is mixed with return air from the planetarium and exhibit halls. The mixed air is further cooled and dehumidified in a central dehumidifier, where a cool desiccant spray absorbs the moisture. The dehumidified air is distributed to three air handlers. Two supply public spaces where internal people-generated loads are heavy and highly variable. In these cases a variable volume of the dehumidified primary air is supplied, as needed to handle the internal humidity load, to fan induction terminals. The terminals mix it with a variable volume of recirculated air, coil heat or cool the air if required, and supply a constant volume of the mixed air to the spaces. In the non-public areas, a minimum ventilation quantity of dry air is distributed to fan coil terminals. Dry coil cooling increases fan efficiency by reducing air flow resistance.

When the humidity load permits, the desiccant conditioner and regenerator are bypassed in an economizer cycle. Based on projected operation of approximately 40 hours per week of public use, the on-site building energy end-use consumption is projected to total 32,600 Btu's per year per square foot, composed of the following: heating, 4000 Btu/y/sq ft ; cooling, 8600 Btu/y/sq ft; lighting 8000 Btu/y/sq ft; fans, 10,000 Btu/y/sq ft; and other, 2000 Btu/y/sq ft. (Reference 4 includes an analysis and comparison of the energy consumption characteristics of six of the system's alternative operating modes: one relying on the electric source only - chiller and heat pump - and the others various combinations with solar and/or cogeneration.)

REFERENCES

1. U.S. Patent No. 4,222,244, issued to Gershon Meckler.
2. U.S. Patents No. 3,918,525 and 4,286,667, issued to Gershon Meckler.
3. U.S. Patent No. 4,315,412, issued to Gershon Meckler.
4. Meckler, Gershon. "Designing Energy Integrated HVAC Systems Based on Thermodynamic Efficiency (Science Museum, Richmond, Va.)." Proceedings of the Designing and Managing Energy Conscious Commercial Buildings Workshops, Denver, July 1982. Prepared for U.S. Department of Energy. Published February 1983.



CONVENTIONAL REFRIGERATION USING CHILLED WATER COOLING AND DEHUMIDIFICATION PLUS REHEAT.

DESICCANT DEHUMIDIFICATION PLUS SENSIBLE COOLING.

Figure 1. Psychrometric changes: desiccant versus condensation dehumidification process

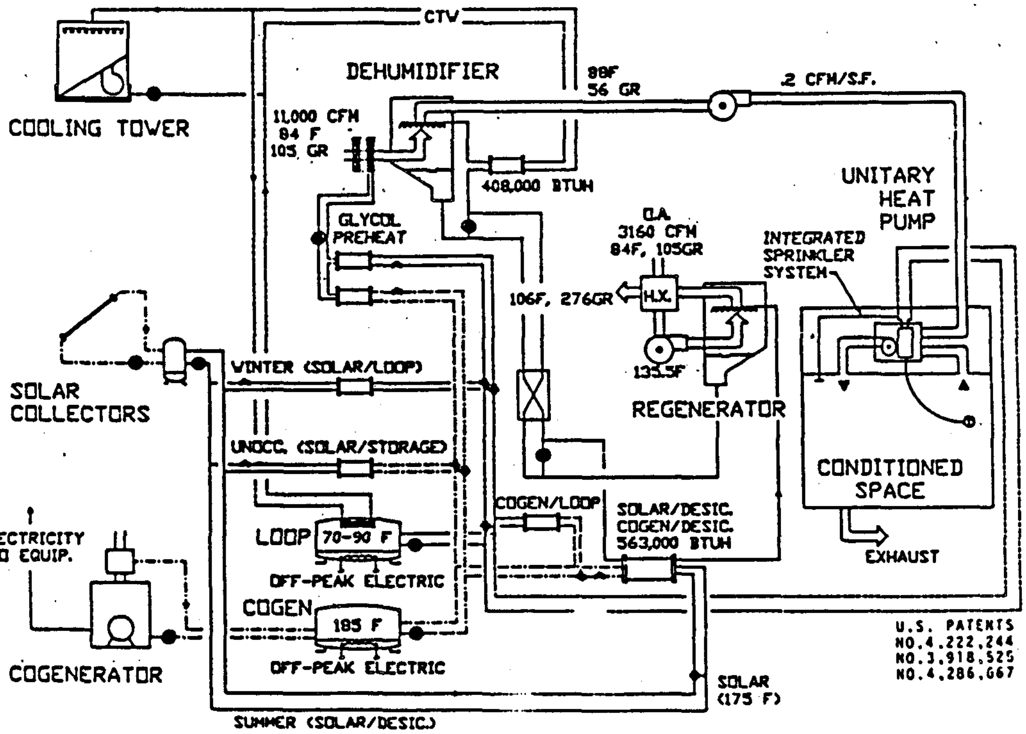


Figure 2. Desiccant dehumidification HVAC system in which selective-energy system and hot thermal storage support both summer cooling and winter heating (Monroe County courthouse, Stroudsburg, PA)

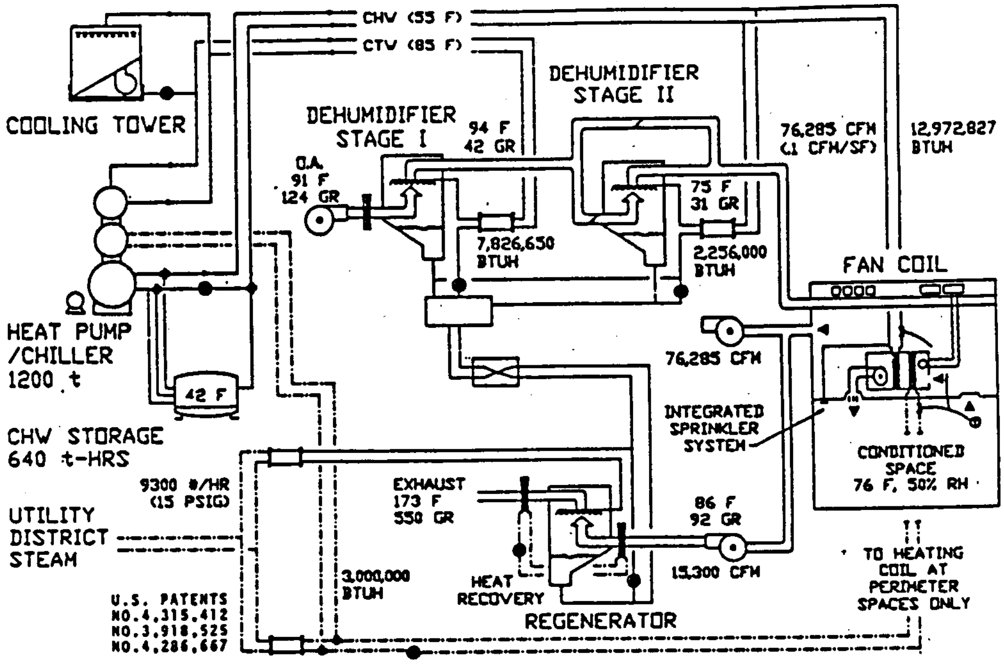


Figure 3. Central desiccant dehumidification with terminal sensible cooling. Minimum ventilation quantity of outside air is dried very deeply (31 grains/lb) permitting minimum primary air distribution, ductwork size, and fanpower. Fan-coil terminals mix primary air with recirculated air and cool (55F) or heat sensibly. (Social Security Administration Metro West Facility, Baltimore)

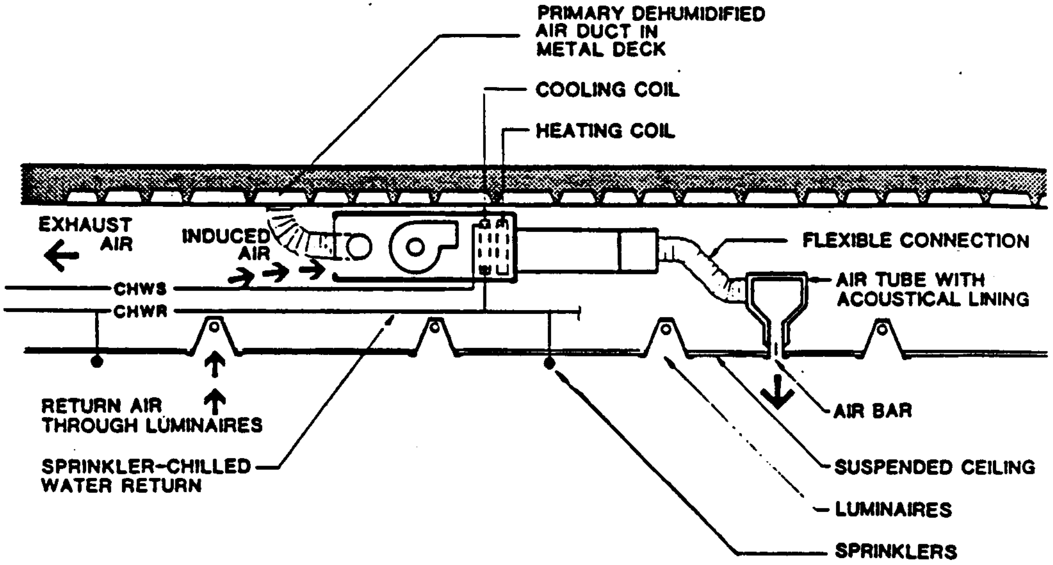
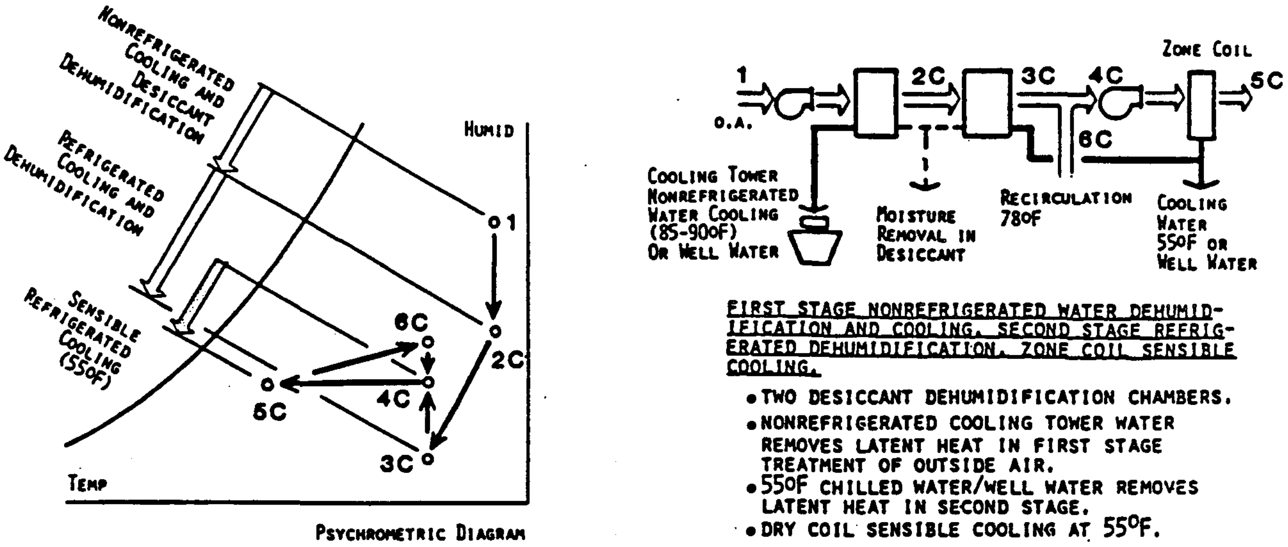


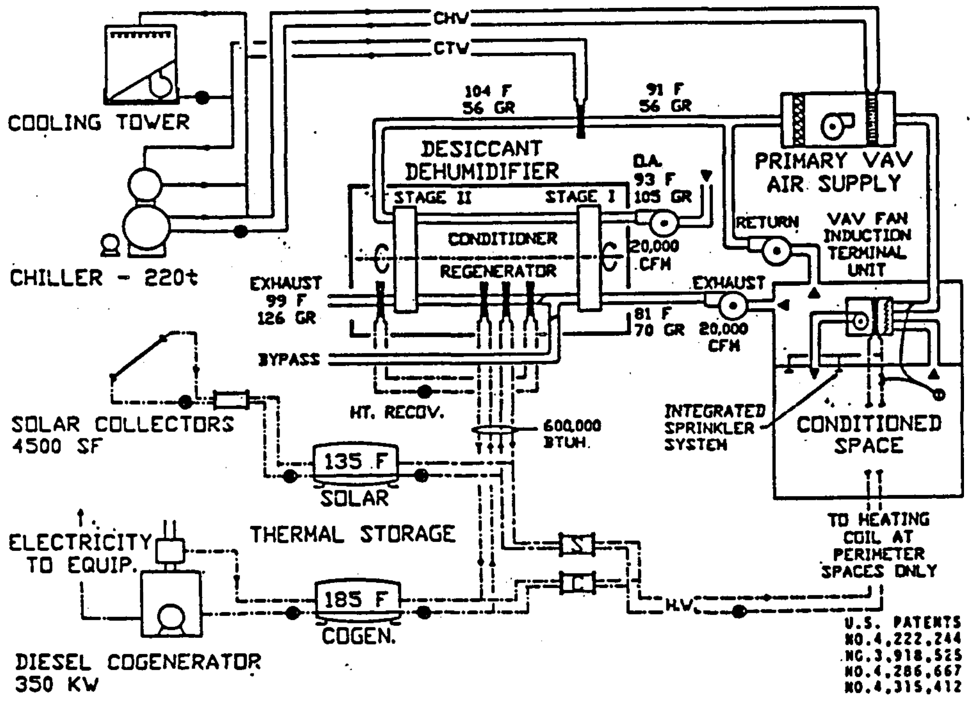
Figure 4. Integrated floor/ceiling sandwich. Connected to the fan-coil units and accessible to be tapped wherever future fan-coil units are needed are (a) primary dehumidified air flowing through modular cells in the cellular steel floor deck and (b) chilled water circulating in modular chilled water/sprinkler piping in the plenum. (Social Security Administration Metro West Facility, Balimore)



FIRST STAGE NONREFRIGERATED WATER DEHUMIDIFICATION AND COOLING. SECOND STAGE REFRIGERATED DEHUMIDIFICATION. ZONE COIL SENSIBLE COOLING.

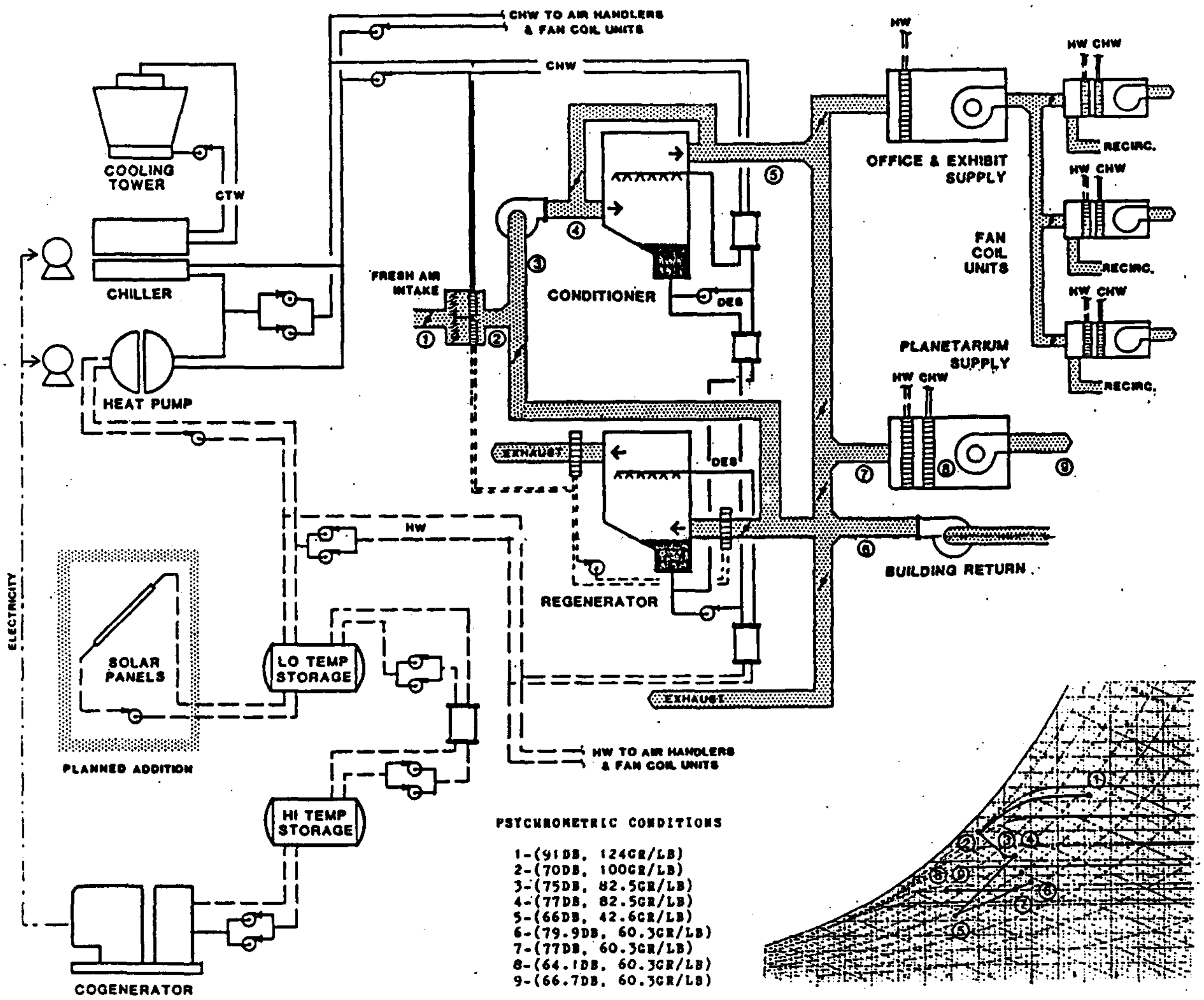
- TWO DESICCANT DEHUMIDIFICATION CHAMBERS.
- NONREFRIGERATED COOLING TOWER WATER REMOVES LATENT HEAT IN FIRST STAGE TREATMENT OF OUTSIDE AIR.
- 55OF CHILLED WATER/WELL WATER REMOVES LATENT HEAT IN SECOND STAGE.
- DRY COIL SENSIBLE COOLING AT 55°F.

Figure 5. Two-stage desiccant dehumidification process



U.S. PATENTS
NO.4,222,244
NO.3,918,525
NO.4,286,667
NO.4,315,412

Figure 6. Solar/cogeneration desiccant dehumidification HVAC system (Veterans Administration Hospital)



PSYCHROMETRIC CONDITIONS

1-(91DB, 124GR/LB)
2-(70DB, 100GR/LB)
3-(75DB, 82.5GR/LB)
4-(77DB, 82.5GR/LB)
5-(66DB, 42.6GR/LB)
6-(79.9DB, 60.3GR/LB)
7-(77DB, 60.3GR/LB)
8-(64.1DB, 60.3GR/LB)
9-(66.7DB, 60.3GR/LB)

Figure 7. Heat pump/cogeneration desiccant dehumidification HVAC system (Science Museum of Virginia, Richmond)

EVALUATION AND VALIDATION OF DESICCANT COOLING/DEHUMIDIFICATION SYSTEMS MODELS

by

J. R. Howell

B. D. Hunn

J. L. Peterson

Center for Energy Studies

The University of Texas at Austin

Austin, Texas 78758

A briefing paper
to be presented at the workshop on
Desiccant Cooling and Dehumidification
Opportunities in Buildings

June 10-11, 1986

Chattanooga, Tennessee

ABSTRACT

This paper reviews the status of fundamental process, component, and systems models for desiccant cooling/dehumidification system analysis, with a focus on experimental data collection and validation needs for these models. These needs are outlined in the context of analytical tool development for solid and liquid desiccant systems. A full-scale test facility that is under construction is described; this facility will be available to support the model evaluation and validation tasks that are identified.

I. INTRODUCTION

Over the past few years, a large research effort has been devoted to improving the performance of desiccant cooling systems. In these systems, humid air is contacted with a liquid or solid desiccant that adsorbs or absorbs water vapor from the conditioned air. The dry air is then cooled either by evaportive cooling or by contact with some sensible cooling source external to the desiccant cycle. The desiccant soon becomes saturated with the water extracted from the air and must be regenerated by heating to drive off the water, usually by desorbing the water to the ambient air.

Such systems in their simplest form can operate with the energy input to the desiccant regenerator at relatively low temperatures (140-158 F for liquid lithium chloride/water solutions, for example). However, simple desiccant air-conditioning systems also tend to have very low Coefficient of Performance (COP) values. On the other hand, desiccant systems, of both the liquid and solid type, reduce energy use through the application of renewable energy resources in the regeneration process. Furthermore, in applications where latent loads are dominant, such as supermarkets, low humidities can be obtained more efficiently with desiccants, as compared with vapor compression refrigeration [1].

To combine the best features of different cooling/dehumidification cycles, a variety of hybrid systems have been developed. Hybrid desiccant/vapor compression cycles, which combine a vapor-compression cycle for sensible cooling with a desiccant cycle for latent energy removal, have been the focus of recent research. A hybrid vapor-compression/liquid desiccant system has been proposed [2], as has a hybrid absorption/desiccant system using a liquid desiccant [3]. However, to achieve significantly more efficient practical air conditioning, improvements in the simple systems must be made, or different and more efficient systems must be developed. This requires a systematic research and development program covering the areas of desiccant properties, fundamental heat and mass transfer processes, improved component design, and better system integration. In such a program, analytical tools, guided and validated using experimental data, are fundamental to the advancement of desiccant technologies. These tools are required to evaluate new system and new component configurations. Comparisons of predicted and measured results not only are used to formulate analytical models, but are needed to establish their credibility among researchers and industry.

Thus, this paper reviews the status of fundamental process, component, and systems models for desiccant cooling/dehumidification analysis, and outlines experimental data collection and validation needed for both solid and liquid systems. Because they show considerable promise where latent loads are high, the development of liquid desiccant systems is emphasized.

II. ANALYTICAL TOOL DEVELOPMENT AND VALIDATION

Analytical tools play a crucial role in the development of desiccant technologies. Basic property models are used to define the behavior of desiccants; fundamental heat and mass transfer models are used to explore potential improvements in the processes and components; and comprehensive systems and building models support the integration of technological improvements into the marketplace. Of equal importance are the contributions of experimental research to the development of the fundamental relationships that are represented by the algorithms in these analytical tools.

A recent Solar Energy Research Institute (SERI) survey paper on desiccant cooling stresses the need for experimental facilities to evaluate and validate property, heat/mass transfer, component, and systems models [4]. Before experimental data needs are outlined at each model level, two general validation issues need to be addressed.

A. Model Validation Issues

First, although component and systems models developed by industry to characterize their equipment have in some instances been validated, those validations are proprietary and do not contribute to establishing the general credibility of the models. Generic component and systems models, whose validation is reported in the open literature, are essential to establish confidence within the technical community. Furthermore, models are often validated against experimental data under design conditions. Comparisons with part-load performance data are required to evaluate models over the full operating range.

Second, at each model level models are developed either from basic principles or from empirical correlations. Detailed hourly (or subhourly) models require carefully controlled laboratory experiments for their validation. On the other hand, empirical component or systems models sometimes can be validated by field data that confirm basic parameter behavior. Thus, the nature of the validation data to be collected is determined by the nature of the model (basic principles or correlations). Whereas hourly or monthly data are needed to validate hourly systems simulations, monthly or annual data are sufficient for bin or other simplified methods.

B. Model Level Hierarchy

The relationship among the several levels of analytical tools is illustrated in Fig. 1. Each level forms the building blocks for the next higher level. Thus component models are built from validated property/process models or algorithms. However, component validation against experimental data tests not only the correct representation of each heat/mass transfer and hydrodynamic process, but also the interaction among processes. Likewise, a system validation tests the correctness of its constituent components, as well as the component interactions. However, we cannot have full confidence in a system- level validation, without validation at each lower level as well. Experimental data needs at each level are discussed below.

C. Model Development at Each Model Level

1. Properties/Fundamental Process Models.

For liquid desiccants, such as triethylene glycol (TEG), lithium chloride, and lithium bromide, adequate data are available to characterize equilbrium vapor pressure, enthalpy, specific heat, and heat of mixing/solution. The sorption and thermophysical properties of solid desiccants commonly used in air-conditioning systems (silica gel, molecular sieve, and manganese dioxide) are fairly well characterized by reliable experimental data taken at universities and national laboratories and in industry. However, data on polymeric desiccant materials, additional data on hysteresis over cyclic operation, and empirical data on fouling are needed [4].

Basic principles models of the simultaneous heat and mass transfer processes in liquid desiccant absorbers and regenerators have generally been based on finite-difference solutions of one-dimensional, quasi-steady-state flows [5-7]. The equations for mass and energy can be written for each fluid stream, with the assumption that the heat and mass transfer coefficients are related through the heat/mass transfer analogy. The following boundary conditions are specified: the air inlet temperature and humidity (or other convenient psychrometric variables); the inlet coolant temperature (in the case of the absorber) or the inlet heating fluid temperature (in the case of the regenerator) -- in both cases equal to the inlet desiccant temperature; and the inlet desiccant concentration. The equations are solved numerically for the desiccant concentration, temperatures of each stream (desiccant, air, and water), and air humidity ratio at each point in the absorber or regenerator. Because the desiccant and water are flowing in the opposite direction from the air, conditions for one stream must be assumed and then the conditions for the other stream are computed. The computed result is then used to redefine the conditions in the first stream, which replace the assumed conditions. This procedure is continued until the variables on each stream converge.

In most studies the analytical model solutions have been compared successfully with experimental data, indicating the validity of these basic principles models [5-9]. However, the wetting fraction, or other global parameter, must be assumed to achieve a good match with experimental data. Additional bench-scale tests are needed here.

Sorption process models for solid desiccants have been developed from basic principles applied to the detailed transfer mechanisms, but generally have a free parameter or two to account for unknown phenomena[10]. Although measured mass transfer rates in the adsorption process meet predictions fairly well, existing desorption models do not predict experimental data accurately. Thus, validation of solid desiccant transfer processes is not considered complete.

Both steady-state and transient experimental heat and mass transfer rates and pressure drops are available for packed beds [11] and for parallel passages [12]. Additional data are being collected for other flow passage geometries, although other geometries and flow configurations need to be tested. The Sorption Test Facility and the Single Blow Test Facility at SERI seem adequate for the additional testing required.

2. Component Models.

Liquid Desiccants. The most important component in a liquid desiccant system is probably the absorber, where water vapor is removed from the humid air, and cooling may also occur. References [5,8,9] provide analyses of this component, and the latter two give comparisons with experimental data for different absorber designs.

Consider an absorber composed of a fin-tube surface, cooled by a liquid flowing within the tubes (Fig. 2). A liquid desiccant film flows downward by gravity over the fin surface and is contacted with air flowing upward through the absorber. To simplify the analysis it is assumed that the desiccant film and the fin temperatures are locally equal, that all properties are uniform at any vertical position (one-dimensional problem), and that the process is steady state.

Peng [5] has carried through the iterative heat and mass transfer process calculations using TEG as the desiccant. Measurement was made of temperature, concentration, and air humidity profiles by Ameringer et al. [9]. They ascribed the differences between prediction and measurement to channeling of the desiccant into certain fin passages, with a resultant high air flow in other passages, causing a very low wetting fraction.

Queiroz, Orlando, and Saboya [8] analyzed and performed experiments on a similar system in which the coolant flow was parallel rather than in counterflow to the air stream. Measured values of the isothermal and adiabatic mass transfer coefficients, as a function of air and desiccant solution mass flow rates, absorber cross-section, and packing length (i.e., mass transfer area per unit volume), were found to be equal within the experimental error. Some work has also been done on the use of packed towers for liquid desiccant absorbers [6]. The regenerator is the other liquid desiccant component in which both heat and mass transfer occur simultaneously. An open-flow regenerator, using direct input of solar energy for regeneration, has been analyzed [13, 14] and several configurations have been tested. Testing results are available in Refs. [15-17].

The governing heat and mass transfer relations for a closed regenerator are presented by Peng [7,18] and are similar to those for the absorber. Using these relationships the temperatures for all fluids, and the air humidity ratio and desiccant/water-vapor concentrations can be determined. Results for the outlet solution concentration as a function of the important parameters are presented in Ref. [7]. Refs. [19-20] compare the predictions and experimental results for a closed regenerator and discuss some of the practical design problems of these devices. Predictions of a detailed model of a packed tower regenerator are successfully compared with experimental results in [21].

Thus, basic principles models of liquid desiccant falling-film absorbers have been developed. Although these models have been compared with experimental data, simplified models usable in hourly simulation codes have yet to be developed and validated. Again, a few proprietary models have been compared to test data, but these models are not generic. A similar situation exists for closed liquid regenerators [7]. Therefore, simplified semiempirical models need to be developed, and experimental facilities are needed in which to evaluate and validate these simplified liquid component models.

Solid Desiccants. The solid desiccant components of interest are the solid bed dehumidifier, the regenerative heat exchanger, and the evaporative heat exchanger. For each of these components, the component performance (effectiveness, pressure drop, etc.), its transient response, and its geometry dependence are the important parameters to be modeled and validated.

Simplified hourly models of dehumidifiers and regenerative counter- or cross-flow heat exchangers have been developed by Jurinak et al. [22], Van den Bulck et al.[23], and Maclaine-cross and Banks [24]. These semiempirical models are based on the effectiveness-NTU characterization, developed from finite-difference solutions of the governing equations. Similarly, most hourly evaporative cooler models are based on effectiveness-Number of Transfer Units (NTU) relationships. Although only limited laboratory test data are available for validation of these models, steady-state and transient validation tests are underway in the Single Blow Test Facility and the Cyclic Test Facility at SERI [4]. However, additional validation work is needed because flow nonuniformities in the rotary wheels tested have so far produced inconclusive results [10].

3. Systems Models. Analysis of the overall performance of desiccant systems, based on detailed differential analysis of individual components, was first reported in Refs. [3, 25]. Experimental behavior and other analyses are given in [25-31].

By combining the differential models for a desiccant absorber and closed regenerator outlined above, Peng [3] developed a model for the overall behavior of desiccant HVAC systems. He found that the system shown in Fig. 3 gave the best performance of the configurations studied.

In this system, a portion of the return air, which has a low dew point in comparison with ambient air, is used to feed an evaporative cooling tower. The chilled water from the tower is used to cool the desiccant absorber, thus increasing the absorber efficiency. The system COP versus evaporator packing factor (dimensionless ratio of mass transfer to capacity rate) plot shown in Fig. 4 is representative of the results obtained when systems models are used for parameter optimization.

Analyses and experimental results for other liquid desiccant cooling system designs are found in [25, 27-33]. The system described in [31] is especially interesting, since it is relatively large scale and provides significant operating experience for these systems.

All-in-all, very limited experimental data have been collected for closed-cycle liquid desiccant systems (e.g., the Kathabar system). These data are generally proprietary and are useful in validating only specialized absorber/generator configurations. TRNSYS simulations are currently being run to determine annual performance of a liquid desiccant system that includes a semiempirical effectiveness-NTU absorber model. However, the limited experimental data available for validation of these models are of poor quality. Simplified models of this type need to be developed for other liquid desiccant system components.

Simulations of solid desiccant systems, where the load is prescribed rather than simulated, have been conducted using TRNSYS [1] or similar hourly programs [34]. Such simulations are used to characterize dynamic system performance over the full range of operation and to determine the effects of design parameter variation. For example, hourly simulations have predicted that the optimum flow ratio (regeneration to process air flow rate) is near 0.8 for the ventilation cycle and about 0.6 for the recirculation cycle. [4].

Building energy analysis programs, such as DOE-2, and simplified tools, such as the bin method, have not been applied to either liquid or solid desiccant systems because appropriate component models are not available. The bin method would require correlational component models to characterize the desiccant and heat exchanger performance for each outside air temperature/humidity bin. These high-level programs include comprehensive loads and HVAC systems models and thus are appropriate to study system designs and to determine the effects of desiccant systems on building energy use. The necessary models need to be developed in order to determine the full promise of these technologies.

Validation of systems-level models is virtually nonexistent. A few comparisons of predictions and experimental data have been made, but these were for very specialized systems and models. Thus, the validation of generic systems models is an urgent need to establish confidence in their use. Both well-controlled laboratory experiments and field experiments that can test for the effects of leakage, controls, and system degradation over time are needed. To address these needs, a full-scale cooling system test facility, described in the next section, is being built at the Center for Energy Studies at The University of Texas at Austin.

III. DESCRIPTION OF A FULL-SCALE COOLING TEST FACILITY

A Dual-Air-Loop Cooling Test Facility is being built at The University of Texas at Austin to test the performance of advanced cooling/dehumidification components and systems at full scale (up to 20 ton cacacity) in a controlled experimental environment. Performance testing of evaporative cooling equipment within a variable-air-volume HVAC system sized for a small office building (approximately 10,000 sq. ft.) is to be conducted initially. Thus, the apparatus will accommodate direct (DEC) and indirect evaporative cooling (IEC) equipment in reconfigurable test sections. In the future, the facility will be used to test desiccant dehumidification, vapor compression, and cool storage equipment.

Figure 5 is a schematic of the test equipment configured for open sorption dehumidification. Both of the air loops, one for primary air and one for secondary air, will have independently controlled heating/cooling and humidification/dehumidification stations to simulate summer ambient or return air conditions for virtually any U.S. climate. The primary air loop also includes provisions for humidification and heating to simulate sensible and latent loads and a chilled water coil to simulate auxiliary cooling. A bypass will be provided in the primary loop to permit blending of air from the load simulator with air from the conditioning station. An air flow rate capacity of 5,000 scfm is to be provided by each air loop. Continuous air flow modulation down to 1,000 scfm will be provided in each loop by controlling the speed of the primary and secondary loop blowers.

The apparatus will condition primary air to a state within the thermal envelope bounded by 50 to 155 F dry bulb and 50 to 73 F dew point temperature, when supplying air at 5,000 scfm. At lower supply air flow rates, the thermal envelope expands to a maximum of 180 F dry bulb and 106 F dew point temperature from the 50 F dry bulb and dew point temperature base.

The cooling test system will be operated to simulate quasi-steady-state hourly time steps. A typical test period will be a series of seven 24-hour load and climate profiles simulating 1- week each in summer, spring, and fall. After steady-state conditions have been established (approximately 15 minutes) at each time step, some 150 data points will be recorded. All significant temperatures, pressures, humidities, air flows, pressure drops, and rotating equipment speeds will be measured. Each loop will operate independently, and both will be controlled by a common, dedicated control and data acquisition system. Several hours of real-time data will be collected, reduced, and stored locally until blocks of reduced data can be sent to the VAX 11-750 computer located elsewhere in the building for further processing and output graphics.

When configured for testing of open sorption dehumidification equipment, the facility can be used to validate mathematical models that relate the performance of the dehumidifier, the evaporative cooler, the regenerator, and the system as a whole, to such important variables as:

1. Ratio of process to regeneration airflow,
2. Process air inlet temperature/humidity,

3. Regeneration air inlet temperature/humidity,
4. Process air outlet temperature/humidity, and
5. Regeneration heat source temperature.

IV. CONCLUDING REMARKS

The discussion and results presented here show that analytical tools play a crucial role in the development of desiccant technologies. Furthermore, it is shown that additional models, and the experimental data for their evaluation and validation, are needed.

Properties/fundamental process algorithms are available and have been validated with some exceptions. Existing test facilities at SERI appear adequate for solid desiccant process models. For liquid desiccants, validated first principles component models are available, but no simplified models have been developed; simplified models are needed for liquid system components. Component models, both first principles and simplified types, are available for solid desiccant components. However, these have not been adequately validated. At the systems level, both solid and liquid models are available, but they are not comprehensive. Moreover, these systems models have yet to be fully validated or incorporated into building energy analysis computer programs. However, a full-scale test facility is being built at The University of Texas at Austin that will facilitate such validation.

Significant improvements in system COP can probably still be made for liquid desiccant cooling systems. The absorbers can be improved through innovative design for improved heat and mass transfer in smaller volumes and at smaller pressure drops. Such improvements in absorber design can probably be incorporated into closed regenerators. Improved system configurations, including the use of integrated components, hold promise for cost reduction.

V. REFERENCES

1. Burns, P. R., Mitchell, J. W., and Beckman, W. A., "Hybrid Desiccant Cooling Systems in Supermarket Applications," ASHRAE Transactions, Vol. 91, Pt. 1, 1985.

2. Howell, J. R., and Peterson, J. L., "Preliminary Performance Evaluation of a Hybrid Vapor-Compression Liquid Desiccant Air-Conditioning System," paper submitted for presentation at the ASME 1986 Winter Annual Meeting, New York, NY, December 1986.

3. Peng, C. S. P., and Howell, J. R., "Optimization of Liquid Desiccant Systems for Solar-Geothermal Dehumidification and Cooling," AIAA Journal of Energy, Vol. 5, No. 6, November-December 1981, pp. 67-74.

4. Penney, T.R., and Maclaine-cross, I., Promising Advances in Desiccant Cooling, Solar Energy Research Institute paper SERI/TP-252-2683, May 1985.

5. Peng, C. S. P., and Howell, J. R., "Analysis and Design of Efficient Absorbers for Low-Temperature Desiccant Air Conditioners," ASME Journal of Solar Energy Engineering, Vol. 103, No. 1, Feb. 1981, pp. 67-74.

6. Gandhidasan, P., Ullah, M. R., and Kettleborough, C. F., "Analysis of Heat and Mass Transfer between a Desiccant Air System in a Packed Tower," ASME Paper 85-WA/SOL-6, November 1985.

7. Peng, C. S. P., and Howell, J. R., "The Performance of Various Types of Regenerators for Liquid Desiccants," ASME Journal of Solar Energy Engineering, Vol. 106, No. 2, May 1984, pp. 133-141.

8. Queiroz, A.G., Orlando, A.F., and Saboya, F.E.M., "Performance Analysis of an Air Dryer for a Liquid Dehumidifier Solar Air Conditioner System", ASME Paper 84-WA/Sol-6, ASME Winter Annual Meeting, New Orleans, November 1984.

9. Ameringer, G., et al., "Validation of Analytical Model of Absorbers for Triethylene Glycol Liquid Desiccant Dehumidification/Cooling Systems," Proc. Japan/US Joint Heat Transfer Conference, Honolulu, March, 1982.

10. Verbal communication with J. W. Mitchell, University of Wisconsin, Madison, Wisconsin, May 1986.

11. Kutscher, C. F., and Barlow, R. S., *Dynamic Performance of Packed Bed Dehumidifiers: Experimental Results from the SERI Desiccant Test Laboratory*, SERI/TR-252-1529, Solar Energy Research Institute, 1982.

12. Schlepp, D., and Barlow, R. S., *Performance of the SERI Parallel-Passage Dehumidifier*, SERI/TR-252-1951, Solar Energy Research Institute, 1984.

13. Lof, G.O.G., Lenz, Terry G., and Rao, Sham, "Solar Regeneration of Lithium Chloride in Open Cycle Absorption Cooling", ASME Paper 82-WA/Sol-6, November 1982.

14. Peng, Chungshiang Patrick, and Howell, John R., "Analysis of Open Inclined Surface Solar Regenerators for Absorption Cooling Applications- Comparison Between Numerical and Analytical Models", *Solar Energy*, Vol. 28, No. 3, 1982, pp. 265-268.

15. Novak, K. S., Wood, B. D., and Nelson, D. J., "Experimentally Determined Correlations for Solar Collector/ Regenerator Heat and Mass Transfer," ASME Paper 85-WA/Sol-7, November 1985.

16. Grodzka, P.G., Rico, S.S., and McCormick, P.O., *Performance and Operational Analysis of a Liquid Desiccant Open Flow Solar Collector*, Lockheed Missiles and Space Company Report LMSC-HREC TR D784488, November 1981.

17. Applebaum, M. A., and Wood, B. D., *Heat Transfer Characteristics of a Glazed Collector/Regenerator for Open-Cycle Solar Absorption Cooling*, Report ERC-R-84035, Department of Mechanical and Aerospace Engineering, Arizona State University, Tempe, Arizona, June 1984.

18. Peng, C.S.P., "The Analysis and Design of Liquid Absorbant/Desiccant Cooling/Dehumidification Cycles for Low Grade Thermal Energy Applications," PhD Dissertation, The University of Texas at Austin, December, 1980.

19. Petty, J.,"The Design, Testing, and Computer Model Comparison of a Liquid Desiccant Regenerator," MS Thesis, The University of Texas at Austin, August 1983.

20. Hunt, I., "Comparison of Experimental Data with Model Predictions for the Performance of a Liquid Desiccant Regenerator," MS Thesis, The University of Texas at Austin, August 1983.

21. Factor, H.M., and Grossman, G., "A Packed Bed Dehumidifier/Regenerator for Solar Air Conditioning with Liquid Desiccants," *Solar Energy*, Vol. 24, 1980, p.541

22. Jurinak, J. J., Mitchell, J. W., and Beckman, W. A., "Open Cycle Desiccant Air Conditioning as an Alternative to Vapor Compression Cooling in Residential Applications," *ASME Journal of Solar Energy Engineering*, August 1984.

23. Van den Bulck, E., Mitchell, J. W., and Klein, S. A., "The Design of Dehumidifiers for Use in Desiccant Cooling and Dehumidification Systems," ASME Paper 84-HT-32, ASME-AIChE National Heat Transfer Conference, Niagara Falls, New York, 1984.

24. Maclaine-cross, I. L., and Banks, P. J., "Coupled Heat and Mass Transfer in Regenerators - An Analogy with Heat Transfer," *International Journal of Heat and Mass Transfer*, 1972.

25. Kakabaev, A., and Khandurdyev, A., "Absorption Solar Refrigeration Unit with Open Regeneration of Solution", *Geliotekhnika*, Vol.5, No.4, 1969, pp.28-32.

26. Siebe, D. A., and Wood, B. D., *Modeling Cooling and Heat Pumping Systems Using Liquid Desiccants*, Report CR-83027, College of Engineering and Applied Science, Arizona State University, Tempe, Arizona, June 1983.

27. Collier, R.K., "The Analysis and Simulation of An Open Cycle Absorption Refrigeration System," *Solar Energy*, Vol.23, 1979, p. 357.

28. Kakabaev, A., et.al.,"Experience in Operating a Solar Absorption Cooling Plant with Open Solution Regenerator", Geliotekhnika, Vol.13, No.4, 1977, pp.73-76.

29. Ko., S.M., and Merrifield, D.V., "Energy-Efficient Desiccant Drying/Dehumidification Using Solar or Fossil Fuel Energy", Proceedings of the 12th Intersociety Energy Conversion Engineering Conference (IECEC), Paper 779070, 1977, pp.434-441.

30. Robison, Harry I., "Open-Cycle Chemical Heat Pump and Energy Storage System", Final Rept. of Research Prog. for Period July 1980-December 1981, Submitted to SERI, Golden, CO, January, 1982.

31. Johannsen, A., "Performance Simulation of a Solar Air-Conditioning System with Liquid Desiccant," International Journal Ambient Energy, Vol.5, No.2, April, 1984, pp.59-88.

32. Gandhidasan, P.,"Heat Flux Reduction by a Flowing Liquid Desiccant Film Over a Roof", ASME Journal Solar Energy Engineering, vol. 105, August 1983, pp. 336-338.

33. Howell, John R., "Hybrid Active Solar Cooling Systems," in Proc. NATO Advanced Study Institute on Solar Energy Utilization: Fundamentals and Applications, Izmir, Turkey, June, 1986. Martinus Nijhoff, Publisher, Dordrecht, the Netherlands, 1986.

34. Kettleborough, C. F., "Solar Assisted Desiccant Dehumidification and Cooling with an Indirect Evaporative Cooler," ASME paper 80-WA/SOL-29, November 1980.

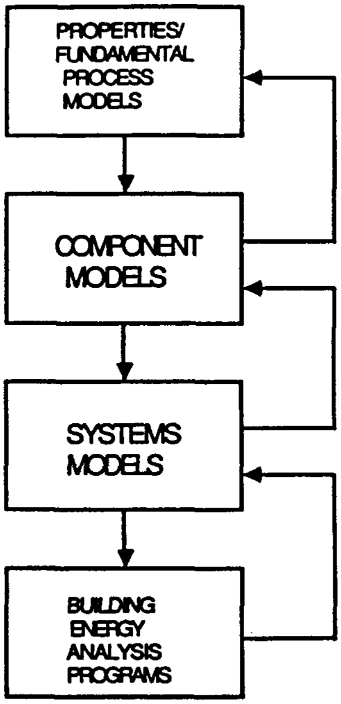


Fig. 1 Model Level Hierarchy

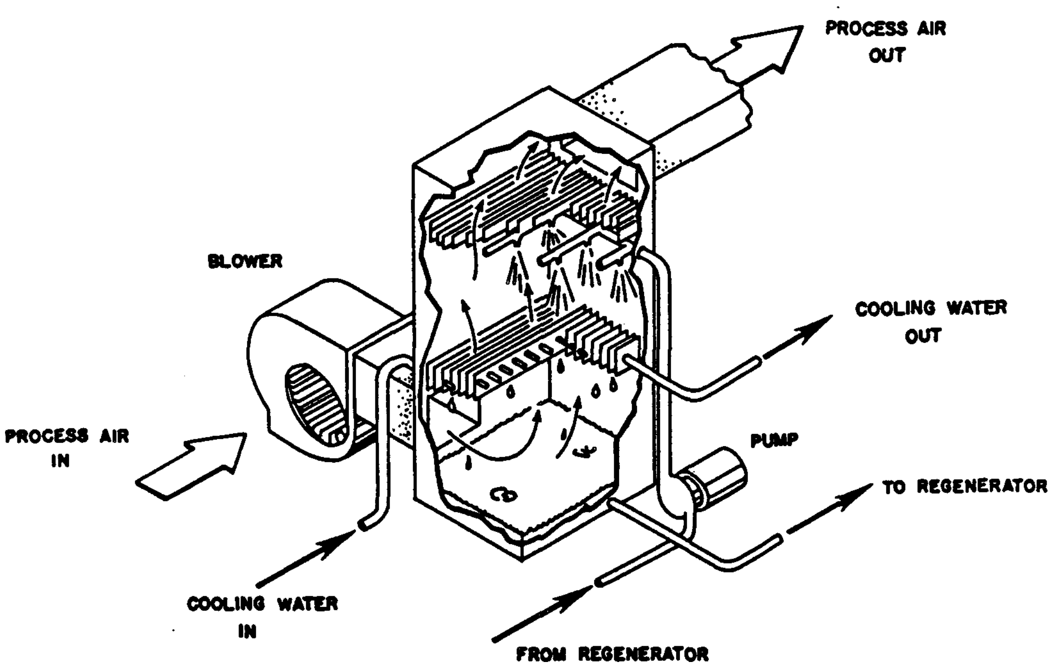


Fig. 2 Typical Liquid Desiccant Absorber Assembly

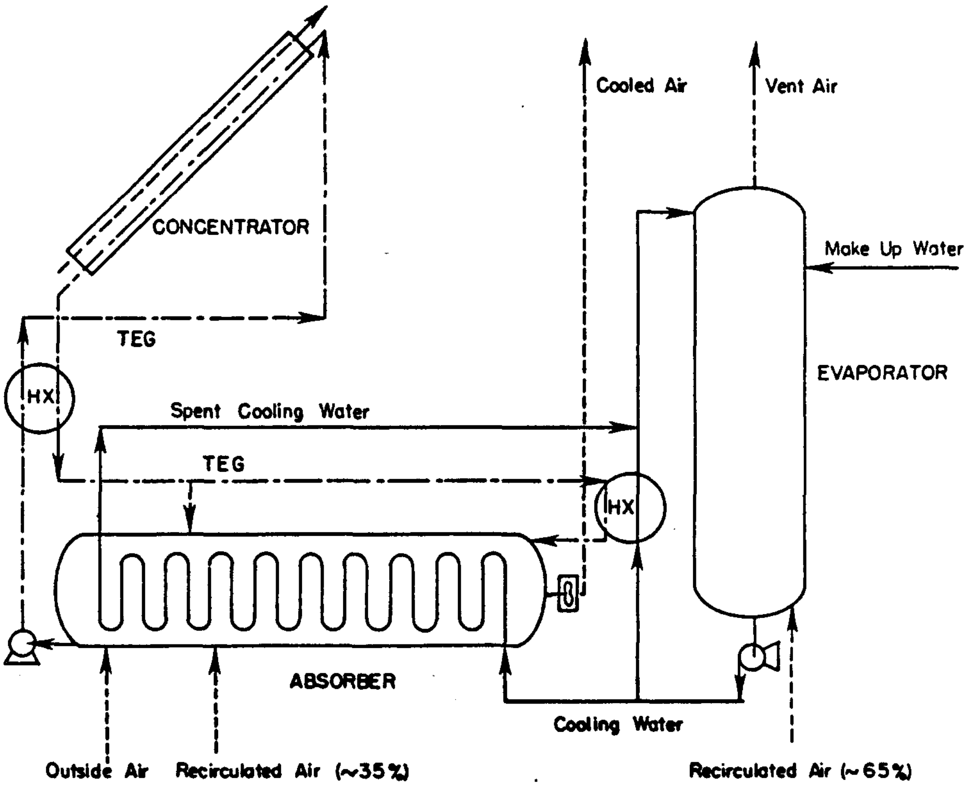


Fig. 3 Liquid Desiccant Cooling System

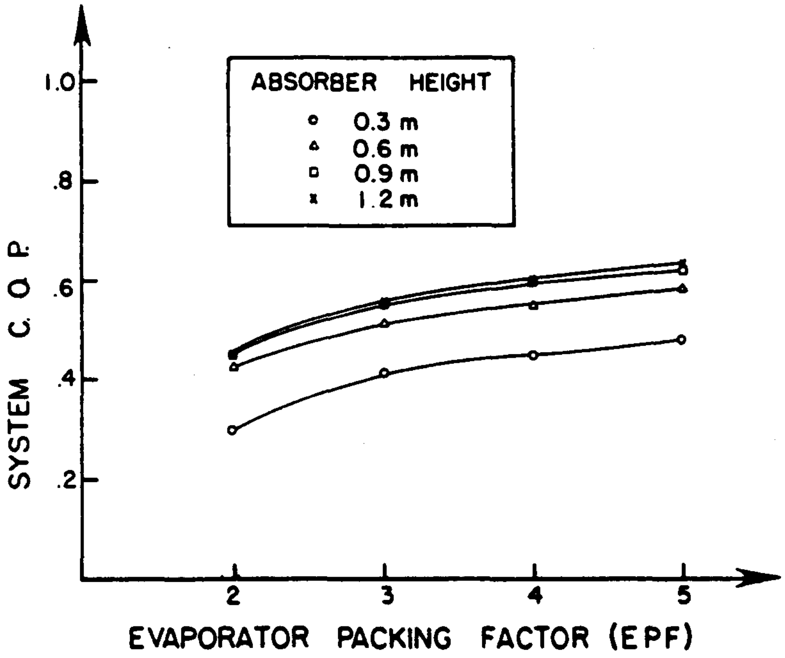


Fig. 4 Liquid Desiccant Cooling System COP vs. Evaporator Packing Factor

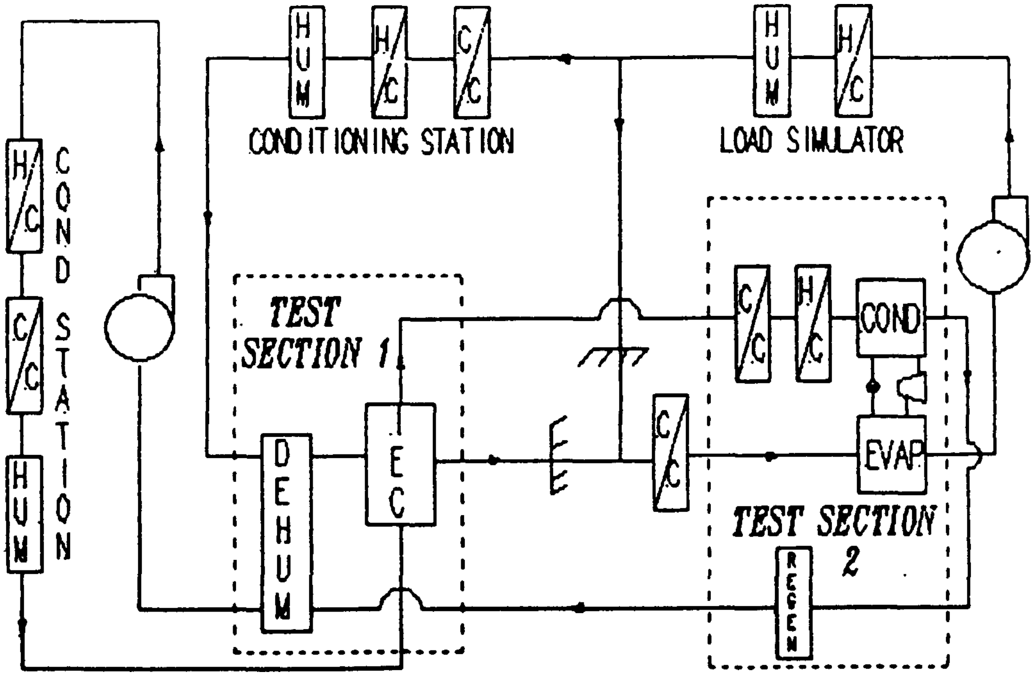


Fig. 5 Schematic of Dual-Air-Loop Cooling Test Facility at The University of Texas at Austin

ANALYTICAL TOOLS FOR SOLID DESICCANT COOLING RESEARCH

K. Schultz, J.W. Mitchell, and S.A. Klein
Solar Energy Laboratory
Univ. of Wisconsin - Madison
Madison, WI 53706

Abstract

Desiccant air conditioning systems have been investigated extensively as alternatives to vapor compression systems. Many component and system models have been developed to aid this investigation. Detailed models of the heat and mass transfer processes in desiccant dehumidifiers have been used to analyze dehumidifier design alternatives. Simplified dehumidifier models, coupled with other component models, have been used to study system performance and optimization. However, the accuracy with which many of these models describe real components and systems has not been adequately verified.

Usefulness of analytical tools in desiccant cooling research

Dehumidifier performance depends on many variables. The regeneration temperature available impacts the choice of desiccant. The thermal properties of the supporting matrix affect dehumidifying capacity. The heat and mass transfer coefficients and the friction factor are determined by the matrix geometry. Dehumidifier performance can be improved by optimizing flow rate ratios and wheel rotation speed and by using purge streams. The effects of these variables can be conveniently determined by using detailed models of desiccant dehumidifiers. Preliminary decisions based on model results can greatly improve the effectiveness of a development or design program.

System performance depends upon the interaction of the various components under the influence of changing ambient conditions and the cooling load. Design point calculations provide a preliminary estimate of performance on which the potential viability of a particular system concept can be judged. (Note that adequate component models are needed to do system design point calculations.) However, the competitiveness of a desiccant cooling system depends on its seasonal performance which is difficult to deduce from design point data alone. Computer simulations of seasonal performance provide a more accurate estimate of system viability and the potential impact to the consumer and utilities in terms of energy and costs. System simulations are also a tool for investigating control strategies which maximize the thermal performance and minimize the parasitic power requirements of desiccant cooling systems.

Detailed models for desiccant dehumidifiers

Much of the potential for further increases in desiccant cooling system performance lies in the dehumidifier. Detailed models of the heat and mass transfer processes that occur in a dehumidifier can be used to judge the potential benefits of various matrix geometries. Optimization of performance in terms of flow rate ratios and rotation speed as well as the desiccant/matrix thermodynamic properties is also easily accomplished with these models.

Finite difference techniques have been used by many researchers to obtain detailed models of dehumidifiers (e.g.: Pla-Barby, 1978; Holmberg, 1979; Barker and Kettleborough, 1980a; Mathiprakasam and Lavan, 1980; Pesaran and Mills, 1984). The finite difference program MOSHMX (Method Of Solving Heat and Mass Xfer), developed by Maclaine-cross (1974), has been used extensively at the Univ. of Wisconsin - Madison Solar Energy Laboratory to model dehumidifier operation, including transient performance (Brandemuehl, 1982), purging (Jurinak and Mitchell, 1984a), desiccant property effects (Jurinak and Mitchell, 1984b), and optimization (Jurinak, et. al., 1984). In MOSHMX, a dehumidifier is described by five nondimensional parameters (a "number of transfer units" and a "mass capacity rate ratio" for each air stream and a "Lewis number"). Two step sizes in the time (rotation angle) coordinate are used, one to follow the initial rapid transfer wave and a second larger step to efficiently cover the rest of the period. Both a matrix technique and successive substitution are used to provide fast convergence to the periodic steady-state solution. To increase precision, the results of three successively smaller grid sizes are extrapolated to zero size making use of the known second order accuracy of the difference equations.

The program DESSIM (Barlow, 1982) was developed at the Solar Energy Research Institute (SERI). In the original DESSIM procedure, the matrix is discretized and each node is treated as a counterflow heat and mass exchanger in which the mass transfer and heat transfer are assumed to be uncoupled. The procedure is not fundamentally correct and recent work (Schultz, 1986a) has shown that unless sufficiently small step sizes are used, inaccurate results may be obtained. This problem can be easily corrected by substituting a set of finite difference equations derived from the partial differential equations into the numerical procedure.

DESSIM is a very well documented program which is easy to follow and modify as needed. It is also a very flexible program providing a wide range of investigative options (e.g.: Barlow and Collier, 1981; Collier, 1985; Schultz, et. al., 1985). In addition, the dehumidifier description is in terms of simple geometrical design parameters. It is these features which make DESSIM an attractive analysis and design tool. To date however, the results from DESSIM have not been compared with those from more fundamental models.

While MOSHMX and (potentially) DESSIM represent solutions to a set of differential equations, the programs have been compared with only limited experimental data. This limitation applies to many of the dehumidifier models available in the open literature. The question therefore remains - does the set of partial differential equations from which the finite difference equations are derived properly describe the desiccant dehumidifier? Of recent interest has been the question of solid-side moisture resistance. Both MOSHMX and DESSIM use an effective gas-side resistance to model the total resistance. High performance dehumidifiers are expected to have small solid-side moisture transfer resistances, but this has yet to be demonstrated experimentally. Recent experimental work at SERI (Schultz, 1986b; Bharathan, 1986) is directed at checking the validity of MOSHMX and DESSIM. Several attempts have been made to model solid-side resistance directly (Pesaran and Mills, 1984; Majumdar, et. al., 1985). However, further experimental efforts are needed.

Several models have been developed for "ideal" dehumidifiers, that is, dehumidifiers in which the air stream and the matrix are in equilibrium at all times and locations (e.g.: Epstein, et. al., 1985; Grolmes and Epstein, 1985). These types of models are particularly useful for investigating desiccant/matrix property effects. Van den Bulck, et. al. (1985a), have solved the conservation equations for an equilibrium dehumidifier using a wave analysis which includes the effects of "shocks" (irreversible sharpening of the wave fronts). An analytical expression for dehumidifier performance results for the case of either complete regeneration or complete saturation during adsorption. Intermediate cases are found by interpolation. The model currently employs property correlations for a nominal silica gel dehumidifier. Future work involves incorporating a general expression for adsorption properties into the model such that a wide range of desiccant/matrix combinations can be investigated with minimal computational effort.

Simplified models for desiccant dehumidifiers

As noted above, detailed models of dehumidifiers are valuable tools in desiccant cooling research. However, their computational complexity limits their application in systems studies to parametric and design point calculations. Efficient hour-by-hour simulation of desiccant cooling systems to estimate seasonal performance and investigate optimal control strategies requires simple, quickly computed models of the dehumidifier.

One method of obtaining a simple dehumidifier model is to correlate the results of a detailed model. Correlations of the outlet states of the dehumidifier can be made (Barker and Kettleborough, 1980b) or the performance of a complete system which incorporates a detailed dehumidifier model can be correlated (Majumdar, el. al., 1982). Investigations using this technique are generally restricted to a particular dehumidifier and to a limited number of describing parameters which are limited in range.

A simplified version of the so-called "analogy method" has been used extensively in system simulations. In the analogy method, the normal dependent variables of temperature and humidity ratio are transformed to two variables termed the F_1 and F_2 potentials (Banks, 1972). The outlet state is then determined by superposing F_1 and F_2 effectivenesses which are directly analogous to the heat exchanger effectiveness. The F_i potentials are functions of the desiccant/matrix properties. In the simplified version, the F_i potentials are represented as nonlinear analytic functions of temperature and humidity ratio for a particular desiccant; to date, this has only been done for a silica gel dehumidifier.

Van den Bulck, et. al. (1985b), have defined an enthalpy and a humidity ratio effectiveness based on their ideal model. They have correlated these effectivenesses in terms of inlet conditions and operating parameters (Ntu's and capacity rate ratios) for a silica gel dehumidifier. While the functional form of the effectiveness correlations is more concise than the direct correlation of outlet states, this method is again currently applicable only to a matrix composed entirely of silica gel.

Confidence in system simulation results requires confidence in the models used. The analogy method has been shown to agree sufficiently well with MOSHMX for simulation purposes (Jurinak, 1982). Van den Bulck's effectiveness model was generated using MOSHMX results. The level of confidence of these simplified models, as well as the correlation models, therefore depends on the confidence level of the detailed models. This provides further justification for experimental validation efforts.

Other component models

Simulation of desiccant cooling systems requires quickly computed models of the other components that make up the system such as heat exchangers, evaporative coolers, and vapor compression equipment. Heat exchangers are conveniently modeled using the ε-Ntu concept (Kays and London, 1984). Typically, the use of a constant effectiveness is satisfactory for system simulations. However, for systems in which flow rate modulation is used to control cooling capacity it may be desirable to include the variation of effectiveness with Ntu.

The ε-Ntu concept can also be used to adequately model both direct and indirect evaporative coolers; again, constant values of effectiveness are usually satisfactory. Maclaine-cross and Banks (1981) have analyzed wet surface exchangers and have proposed a heat exchanger-cooling tower combination in which part of the air conditioned in the heat exchanger by the chilled water from the tower is used as the inlet cooling tower air stream. The advantage of this device is that, unlike evaporative coolers, the heat and moisture transfer processes become reversible for an ideal component. A model of this device based on the analysis of Maclaine-cross and Banks is computationally too extensive for direct use in system simulations. Crum (1986) has developed a simplified model in which the performance at the desired conditions is interpolated from a table of values generated by the more detailed model.

Models of vapor compression units are needed to simulate the performance of hybrid systems. In addition, simulation of vapor compression systems is important to provide information against which to judge the competitiveness of desiccant cooling systems. Jurinak (1982) used an interpolative routine based on manufacturer's data to simulate the performance of a residential heat pump. Howe (1983) used a nonlinear regression fit of manufacturer's data to model a commercial scale vapor compression system. In hybrid systems, nonstandard vapor compression unit operating conditions are encountered which may be beyond the range of the available data. Burns (1985) extrapolated vapor compression unit performance to high condensing temperatures using the functional dependence of the Carnot COP on condensing temperature.

Systems models

Desiccant cooling system models are easily developed by appropriately connecting the various component models together. Simulations can then be run by driving the system with time dependent forcing functions such as weather data. The program TRNSYS (TRaNsient SYstem Simulation, Klein, et. al., 1983) provides a very flexible structure for doing this. TRNSYS contains models for solar collection/storage systems and building cooling loads along with routines for reading weather data and summarizing results. The modular nature of TRNSYS makes it convenient to consider a variety of component arrangements or replacement of one component model with another. The hour-by-hour nature of the simulation allows investigation of control-strategies for responding to changing conditions and of the potential impact on the power utilities.

Jurinak, et. al. (1984) have simulated the performance of the solar-assisted ventilation cycle for residential applications using TRNSYS. Several levels of the analogy method, along with MOSHMX, were used as dehumdifier models. Howe (1983) has simulated a desiccant/vapor compression hybrid system for a small office building. The simplified analogy method was used to model the dehumifier. Standard TRNSYS models were used to model the building cooling loads. Burns (1985) has simulated the hybrid system at the Jewel supermarket in Chicago. The dehumidifier model used is combination of the simplified analogy method and van den Bulck's effectiveness correlations. Crum (1986) has simulated a desiccant cooling system proposed by Maclaine-cross (1985) which incorporates a heat exchanger/cooling tower in place of an evaporative cooler. The objective of the work is to optimize system operation and to determine the impact of such systems on an electric utility.

Conclusions

A variety of tools exist for modeling desiccant cooling systems, both detailed and simplified. These tools have significantly aided the analysis and development of competitive desiccant cooling systems. However, further efforts in the areas of experimental verification and model development are needed. Further experimental work is needed to ensure that the detailed dehumidifier models properly describe real dehumidifiers. More data on actual systems is needed to ensure that component interactions are being properly modeled and that simulation results are accurate. Currently, there appears to be a lack of simplified models for dehumidifiers with desiccants other than silica gel. While system design point studies can be used to choose an appropriate desiccant/matrix combination, simulation of the system requires a quickly computed model that incorporates the chosen properties. There is also much modeling work to be done in optimizing specific system operation and control.

References

Banks, P.J., 1972. "Coupled Equilibrium Heat and Single Adsorbate Transfer in Fluid Flow Through a Porous Medium: 1. Characteristic Potentials and Specific Capacity Ratios," Chem. Engr. Sci., vol. 27, pp. 1143-1155.

Barker, J.M., and C.F. Kettleborough, 1980a. "The Adiabatic Adsorption-Desorption Characteristics of Silica Gel Beds - Part 1," 80-WA/Sol-30, ASME Winter Annual Meeting, Nov.

Barker, J.M., and C.F. Kettleborough, 1980b. "The Adiabatic Adsorption-Desorption Characteristics of Silica Gel Beds - Part 2," 80-WA/Sol-31, ASME Winter Annual Meeting, Nov.

Barlow, R.S., 1982. "Analysis of Adsorption Process and of Desiccant Cooling Systems - A Pseudo-Steady-State Model for Coupled Heat and Mass Transfer," SERI/TR-631-1330, Solar Energy Research Institute, Golden, CO, Dec.

Barlow, R., and R.K. Collier, 1981. "Optimizing the Performance of Desicccant Beds for Solar Regenerated Cooling," Proc. 1981 Annual Meeting of AS/ISES, Philadelphia.

Bharathan, D., 1986. draft report, Solar Energy Research Institute, Golden, CO.

Brandemuehl, M.J., 1982. "Analysis of Heat and Mass Transfer Regenerators with Time Varying or Spatially Nonuniform Inlet Conditions," Ph.D. thesis, Solar Energy Lab, Dept. Mech. Engr., Univ. of Wisconsin - Madison.

Burns, P.R., 1985. "An Analysis of Hybrid Desiccant Cooling Systems in Supermarket Applications," M.S. thesis, Solar Energy Laboratory, Univ. of Wisconsin - Madison.

Collier, R.K., 1985. "Principles of Open-Cycle Desiccant Cooling System Operation and Design," presented at GRI Desiccant Materials Workshop, Chicago, IL, Oct. 3, 1985, TE4315-169-85, Gas Research Institute, Chicago, IL.

Crum, D., 1986. unpublished work to be completed in June at the Solar Energy Laboratory, Univ. of Wisconsin - Madison.

Epstein, M, et. al., 1985. "Desiccant Cooling System Performance: A Simple Approach," ASME J. Solar Energy Eng., vol 107, pp. 21-28, Feb.

Grolmes, M.A., and M. Epstein, 1985. "Property and Composition Effects on the Cyclic Performance of Fixed Bed Desiccant Air Dehumidification," presented at ASME 23rd National Heat Transfer Conf., Denver, CO, Aug.

Holmberg, R.B., 1979. "Combined Heat and Mass Transfer in Regenerators with Hygroscopic Materials," ASME J. Heat Transfer, vol. 101, p. 205, May.

Howe, R.R., 1983. "Model and Performance Characteristics of a Commercially-Sized Hybrid Air Conditioning System Which Utilizes a Rotary Desiccant Dehumidifier," M.S. thesis, Solar Energy Laboratory, Univ. of Wisconsin - Madison.

Jurinak, J.J., 1982. "Open cycle Desiccant Cooling: Component Models and System Simulations," Ph.D. thesis, Solar Energy Lab, Dept. of Mech. Eng., Univ. of Wisconsin - Madison.

Jurinak, J.J., and J.W. Mitchell, 1984a. "Recirculation of Purged Flow in an Adiabatic Counterlfow Rotary Dehumidifier," ASME J. Heat Transfer, vol. 106, pp. 369-375, May.

Jurinak, J.J., and J.W. Mitchell, 1984b. "Effect of Matrix Properties on the Performance of a Counterflow Rotary Dehumidifier," J. Heat Transfer, vol. 106, pp. 638-645, Aug.

Jurinak, J.J., J.W. Mitchell, and W.A. Beckman, 1984. "Open-Cycle Desiccant Air Conditioning as an Alternative to Vapor Compression Cooling in Residential Applications," J. Solar Energy Eng., vol. 106, pp. 252-260, Aug.

Kays, W.M., and A.L. London, 1984. Compact Heat Exchangers, McGraw-Hill, New York.

Klein, S.A., et. al., 1983. "TRNSYS - A Transient Simulation Program," Univ. of Wisconsin - Madison, Engineering Experiment Station Report 38-12, version 12.1.

Maclaine-cross, I.L., 1974. "A Theory of Combined Heat and Mass Transfer in Regenerators," Ph.D. thesis, Dept. of Mech. Engr., Monash Univ., Clayton, Victoria, Australia.

Maclaine-cross, I.L., and P.J. Banks, 1981. "A General Theory of Wet Surface Heat Exchangers and its Application fo Regenerative Evaporative Cooling," J. Heat Transfer, vol. 103, pp 579-585.

Maclaine-cross, I.L., 1985. "High Performance Adiabatic Desiccant Open Cooling Cycles," J. Solar Energy Eng., vol. 104, pp. 229-236, Feb.

Majumdar, P., W.M. Worek, and Z. Lavan, 1982. "Simulation of a Cooled Bed Solar Powered Desiccant Air-Conditioning System," Proc. 1982 Annual Meeting of AS/ISES, p 627.

Majumdar, P., W.M. Worek, and Z. Lavan, 1985. "Heat and Mass Transfer in a Composite Desiccant Material," presented at ASME 23rd National Heat Transfer Conf., Denver, CO, Aug.

Mathiprakasam, B., and Z. Lavan, 1980. "Performance Predictions of Silica Gel Desiccant Dehumdifiers," DOE/CS/31589-T10 (DE82003367), U.S. Dept. of Energy, Jan.

Pesaran, A.A., and A.F. Mills, 1984. "Modeling of Solid-Side Mass Transfer in Desiccant Particle Beds," presented at ASME 6th Solar Energy Divisional Conf., Las Vegas, Nev., Apr.

Pla-Barby, F.E., G.C. Vliet, and I.L. Panton, 1978. "Performance of Rotary Bed Silica Gel Solid Desiccant Dryers," 78-HT-36, AIAA-ASME Thermophysical and Heat Transfer Conf., May.

Schultz, K., R. Barlow, A. Pesaran, and F. Kreith, 1985. "An Analysis of a Direct Radiation Solar Dehumidification System," ASME 23rd National Heat Transfer Conf., Denver, CO, Aug.

Schultz, K., 1986a. unpublished ongoing work at Solar Energy Laboratory, Univ. of Wisconsin - Madison.

Schultz, K., 1986b. "SERI Desiccant Cooling Test Facility: Status Report," SERI/TR-252-2718, Solar Energy Research Institute, Golden, CO, Mar.

van den Bulck, E., J.W. Mitchell, and S.A. Klein, 1985a. "Design Theory for Rotary Heat and Mass Exchangers - I. Wave Analysis of Rotary Heat and Mass Exchangers with Infinite Transfer Coefficients." Int'l J. Heat Mass Transfer, vol. 28, no. 8, pp. 1575-1586.

van den Bulck, E., J.W. Mitchell, and S.A. Klein, 1985b. "Design Theory for Rotary Heat and Mass Exchangers - II. Effectiveness - Number-of-transfer-units Method for Rotary Heat and Mass Exchangers," Int'l J. Heat Mass Transfer, vol. 28, no. 8, pp. 1587-1595.

ENERGYMASTER® - DESICCANT COOLING IN THE MARKETPLACE

James A. Coellner, Vice President/Research & Engineering
American Solar King Corporation
Waco, Texas

In 1985 American Solar King Corporation of Waco, Texas began retail sales of its Energymaster® Cooling, Heating and Ventilating System. The open cycle desiccant unit had been under development by American Solar King Corporation (ASK) since 1982. After starting with technology obtained from Exxon Corporation, ASK moved to its own proprietary concepts in 1983. Field tests were conducted with pre-production prototypes in 1984. Production equipment was installed in the first half of 1985 and manufacturing began in the summer of 1985. As of January 1, 1986, approximately 100 units are in operation.

Energymaster offers the user many significant features not found in conventional heating and cooling systems.

1. Ventilation - All air supplied to the home is brought in from outdoors in both heating and cooling modes.
2. Effective Filtration - 40% efficiency, two inch pleated supplied as standard.
3. Low Speed Mechanical Components - Low noise, low vibration and long life.
4. Long Product Life - Electrostatic applied baked finish on stainless steel.
5. Low Maintenance - Belt adjustments and filters.

The Energymaster supplies 1250 CFM of heated or cooled air with electrical power consumption of just 1200 watts. It can be used in conjunction with almost any thermal source needing 33,000 BTU per hour in the cooling mode and thermal input equal to the space load for winter operation. The thermal input to the basic unit is in the form of hot water at a temperature of 140 to 160°F.

As shown in the preceding table, with regard to cooling capacity, Energymaster is a more than adequate replacement for a 4-ton unit in the Southwest, or a 3-ton unit in the Southeast. In a self-contained, gas fired configuration Energymaster is first cost competitive with replacement high efficiency heating and cooling systems. The operating cost comparison rests solely on residential electric and gas rates. In its gas fired configuration 40 cubic feet per hour of natural gas are consumed in addition to 1.2 KW of electric power. Based on operation at mean summer conditions Table II presents operating cost comparisons.

	Miami	Houston	Phoenix
Electric Rate - $/KWH	.101	.088	.092
Gas Rate - $/CCF	.927	.513	.520
Operating Cost - Conv.	$.333/HR	$.290/HR	$.396/HR
Operating Cost - EM	$.492/HR	$.311/HR	$.318/HR
Electric/Gas Cost Ratio	3.19	5.02	5.18

TABLE II

It is clear that, although climate is a variable, the key to operating cost competitiveness is the cost of gas relative to the cost of electricity. If a heat pump were substituted for the high efficiency air conditioner used in Table II, its operating cost would be $.407/HR in Miami, $.354/HR in Houston and $.484/HR in Phoenix. Energymaster would show significant savings in Houston but still have an operating cost disadvantage in Miami. This is solely due to the low electric/gas price ratio.

MARKET ACCESSMENT

Although Energymaster offers features unavailable in any other HVAC product, ASK believes that market penetration will be largely dependent on operating cost advantages. Functionally the unit is competitive in all climatic conditions in the Continental U.S., therefore the geographic markets are defined only by substantial cooling needs and higher electric/gas price ratios.

Based on current residential utility pricing the Southeast is not an attractive market. Texas and the Southwest provide utility pricing which gives Energymaster strong economic advantage over even the highest rated all-electric units. American Solar King Corporation is now concentrating on those markets with success.

This success will continue to grow with consumer and trade understanding of the product as long as gas remains inexpensive compared to electricity. Entry into other markets is totally dependent on the pricing of utilities.

Based on the features offered and the low level of energy consumption it would seem to be readily marketable. There is a great misconception that an innovative, cost-effective product will be welcomed by consumers. The difficulties have become very apparent to us. Much of the problem is due to the confused state of the marketplace. In the real world of heating and air conditioning, perceptions and beliefs, not facts or needs, govern. The confusing situation has been generated and maintained by the existing infrastructure of manufacturers and dealers as well as utility companies. The major impediments to easy acceptance center on equipment ratings and load determination for equipment selection.

EQUIPMENT RATINGS

Vapor compression refrigeration equipment is understood by the HVAC industry and the public to be synonymous with cooling or air conditioning. It is difficult for most people to perceive that there is any other means of creating cooled air. The air conditioning industry through the American Refrigeration Institute has set up a methodology and mechanism to rate one vapor compression machine relative to another in terms of capacity, power consumption and efficiency. Although this system is fair for vapor compression equipment because of common performance characteristics, it puts alternative systems at a distinct disadvantage.

Since the performance characteristics are significantly different, a single rating point comparison is not valid. This is particularly true when the rating point bears no relationship to actual operating conditions. There is no established, "sanctioned", method for regionalized operating conditions. For any cooling unit it is absurd to state that capacity or efficiency (EER, COP or SEER) will be the same for Miami and Phoenix. With regard to Energymaster, the apparently simple question "What's Your EER?" must be answered "It All Depends." This is the same answer that should be given for any system which will be operational in varying climates.

LOAD DETERMINATION

Although the literature abounds with discussions of "Comfort", "Comfort Indices" and "Comfort Envelopes", I have been unable to find any definitive analysis of the conditions of temperature and humidity in actual residences.

The most telling piece of information is probably the fact that although vapor compression units are rated on the sum of sensible and latent cooling, control is by temperature sensing only. Since machine operation is governed by sensible load only, indoor humidity is uncontrolled.

Latent and sensible space loads are independent of one another. Sensible loads are imposed by heat sources within the building and heat transfer dependent on temperature difference. Moisture loads are imposed by internal sources and mass transfer dependent on partial pressure difference. The lack of interdependence is rarely recognized in the real world.

Random indoor measured humidity levels indicate that indoor dew point tends to approach outdoor dew point. This means that in humid climates indoor relative humidity levels are well above assumed values. In spite of scholarly discussions and comfort zone rating points, it appears that consumers are concerned primarily with temperature control and that humidity control has little real value.

This hypothesis is consistent with the methods employed by HVAC contractors in selecting equipment for residences. Generally rules of thumb are employed, presented in unit of CFM per square foot or square feet per ton. The most common value employed is 600 square feet/ton throughout most of the United States. A major exception is found in the hot Southwest where the magic numbers are 400 square feet/ton for conventional construction and 500 square feet/ton for well insulated buildings. Although this variation reflects both higher summer design dry-bulb temperatures and reduced air conditioner performance at higher ambients, it apparently disregards latent loads. ASK has encountered some acceptance difficulty with the industry and the public by terming Energymaster a "3-ton" unit. The following table illustrates the confusion of nomenclature. It compares two identical residences of 1,800 square feet. The conventional units had SEER ratings of 10.6.

	Southwest		Southeast	
Design temperature	107°F		92°F	
Mean temperature	90°F		82°F	
Unit	4-ton	EM	3-ton	EM
Sensible cooling at design (BTUH)	35,900	38,000	18,900	20,000
Sensible cooling at mean (BTUH)	40,000	41,000	19,300	22,000
Electric consumption at design	4.6 KW	1.2 KW	3.5 KW	1.2 KW
Electric consumption at mean	4.3 KW	1.2 KW	3.3 KW	1.2 KW

TABLE I

It is clear that nominal ratings are nearly meaningless when output under operating conditions is the only real measure of consequence.

A Desiccant/Refrigeration Hybrid System for Air Conditioning Buildings

By
Kennard L. Bowlen
Cargocaire Engineering Corporation

Presented at Tennessee Valley Authority's
Solar & Energy Conservation Training Institute
Chattanooga, Tennessee, June 10-12, 1986
Sponsored by EPRI, GRI and TVA
Coordindated by DOE/SERI

Desiccants, particularly calcium chloride, were used during the 19th century for drying air and other gases for a few industrial applications including blast furnaces. Broader applications for desiccants were employed after World War I when researchers at John Hopkins University developed silica gel while investigating adsorbent materials for gas mask canisters. This was a break-through as silica gel could be conveniently reactivated by heating, whereas desiccant salts previously used, deliquesced and had to be replaced.

The John Hopkins development spawned a company that produced silica gel and multiple desiccant bed dehumidifiers. These dehumidfiers were used for drying industrial gases; transformer breathers; numerous applications of space drying; even including accommodating the latent heat load of an office building in Baltimore.

Effective performance of desiccant dehumidifiers led to further air drying applications where the process, product or storage required low humidity and/or condensation control. These applications included inactive ships; warehouse and cave storage; pharmaceuticals; chemical and food processing; missile silos and submarines; rubber and plastic moulding; production of electronic components, film, automotive safety glass and numerous other applications where humidity would lead to degradation or where moisture could not be tolerated.

Before the use of desiccant dehumidifiers, many of these industries would shut-down for maintenance and vacation during the humid summer season. Some companies moved to naturally dry climates to achieve year-round production, but sometimes they found that they had moved out of their area of marketing and into a short labor area. The use of desiccant dehumidfiers for humidity control allowed full year-round quality production in the most advantageous areas for the company.

During the 1950's, Neil Pennington[1] recognized the potential use of desiccants to dry air sufficiently so that the air could be evaporatively cooled to a comfortable level for air conditioning. Concurrently, Carl Munters[2] developed a desiccant-treated, compact-extended surface for heat and mass transfer, as well as a similar principle of air conditioning by dehumidification and sensible heat transfer followed by evaporative cooling.

Munters' development led to an efficient rotary air-to-air heat exchanger, a compact, efficient desiccant dehumidifier and various studies of the Munters Environmental Controller (MEC). MEC progressed to the demonstration stage, but MEC was not developed beyond demonstration during the following twenty years, probably because relatively inexpensive electricity did not provide motivation for the development of alternatives to conventional electrically-powered vapor compression refrigeration.

Rotary heat exchangers were quickly accepted in the market where a large volume of outside air was conditioned. Applications of desiccant dehumidifiers increased, but these were still mainly in the segments of space drying, condensation control and industrial processing.

[1] Neil Pennington, "Patent No. 2807258, Air-Conditioner, Including Furnace", U. S. Patent Office September 24, 1957.

[2] Carl Munters & Per Norback, "Patent No. 2926502, Air Conditioning System", U. S. Patent Office March 1, 1960. (Filed July 7, 1955)

Increased energy costs during the late 1970's and early 1980's resulted in further investigation of the MEC principle and analysis of desiccants to accommodate the latent heat portion of air conditioning loads.

Conventional vapor compression air conditioners handle the latent heat portion of the load by condensing water from the air by cooling the air below its dew point temperature. This process is limited to temperatures above freezing, therefore the volume of air treated may be large to meet the requirements of the latent heat load. The process is inefficient; as the low dew point temperatures needed to remove humidity require low evaporator temperatures, thereby reducing the system's coefficient of performance (COP). There is further inefficiency when reheating is necessary to raise the dry bulb temperature to a level of comfort.

The use of a desiccant for the latent heat portion of the load results in just a sensible heat load for the vapor compression system, allowing operation with dry evaporator coils at a higher temperature, resulting in improvement of the vapor compression system's COP.

A reactivated desiccant absorbs or adsorbs humidity , depending on the type of desiccant, directly from the air to its surface where the water vapor is condensed, converting the latent heat of sorption to sensible heat. This causes the temperature of the air to rise. The air leaving the desiccant is warm and dry. When the application requires cool dry air the system's efficiency will be enhanced by the use of low energy cooling such as an indirect evaporative cooler or an air-to-air heat exchanger to remove a portion of the sensible heat load.

A hybrid desiccant/refrigeration air conditioning system which intergrates gas reactivated desiccant dehumidification, indirect evaporative cooling and electric vapor compression cooling has been developed by Cargocaire Engineering Corporation under a share-funded contract with the Gas Research Institute (GRI) (Figure 1).

This system employs several features which contribute to reducing the connected load and energy consummed. These features follow:

- heat for reactivation of desiccant
    - lowest practical temperature for effective reactivation
    - heat reclaimed from refrigeration condensing circuits
    - highly efficient gas-fired air heater
- post-cooling by low energy means
- improvement in COP of vapor-compression refrigeration
- reduced electricity for supply air blower.

Industrial dehumidifiers may be required to dry to -40°F dew point temperature or even lower. This requires a deep desiccant bed or a wheel, moderate or low air velocities and high reactivation air temperatures, 250°F to 300°F. Air conditioning applications do not require this degree of drying. This allows thinner beds or wheels, higher velocities and lower reactivation air temperatures. This system takes advantage of these and its reactivation temperature is 190°F to 220°F with velocities higher for compactness and at reduced static pressure loss due to the thinness of the desiccant bed or wheel.

Desuperheating coils that are in the condensing circuit of the final post-cooling stage, provide further economy of reactivation by raising the reactivation air temperature from ambient to typically 110°F.

An advanced gas-fired heat pipe reactivation air heater that has an efficiency of 85% is employed. This compares with usual indirect air heaters that are 60% to 70% efficient.

The first of two stages of post-cooling is by an indirect evaporative cooler that cools the air that has been dried and heated by the dehumidifier. This air is generally 100°F to 110°F, and it is cooled to a 75% to 85% approach between these values and the outside wet bulb temperature. The indirect evaporative cooler handles 30% to 40% of the total sensible heat load when at design conditions. At less than design conditions, the indirect evaporative cooler handles a greater portion of the total sensible heat load as it is controlled to be the first stage of cooling. Energy efficiency ratios (EER's), when considering all debits, of 45 have been recorded.

The second stage of post-cooling is by vapor compressor refrigeration. The load is sensible heat only, therefore a moderately high suction temperature for an improved COP is possible. The COP is further improved by integrating an evaporatively-cooled condenser that is in series with the secondary or wet circuit of the indirect evaporative cooler (Figure 2). This lowers the condensing temperature to an approach of the outside wet bulb temperature. Operating cost of an evaporatively-cooled condensing unit are 25% to 30% less than an air-cooled condensing unit, as the COP is typically increased from 3.0 to 5.0. COPs of greater than 6.0 have been recorded during field testing (Figure 3).

A desiccant system is not limited by the frost point, therefore the latent heat load can be accommodated with less air volume than a vapor compression system that maintains its suction temperature above the frost point to avoid freeze-up. This results in reduced supply air when the latent heat load establishes the air volume. In applications with a high ratio of latent heat, this can be a 50% reduction of supply air volume. The size of the supply blower and motor, duct size and operating cost are proportionately reduced.

Field testing in supermarkets is revealing that desiccant-based hybrid systems with capacity equivalent to 64 tons of refrigeration requiring 99 kWh of electricity and 240 mBh of reclaimed after heat, are operating with 54 kWh of electricity and 279 cfh of gas. In addition, the lowered humidity reduces the operating cost of the refrigerated display cases, improves the appearance of frozen products by keeping them frost-free and it allows higher, more comfortable temperatures in the frozen food aisles.

Other building applications, where the latent heat load affects the activity, and the integrated desiccant/refrigeration hybrid air conditioning system provides definite benefits include, but are not limited to the following:

- hospitals and other health care facilities
- research laboratories
- pharmaceuticals
- semi-conductors
- dehydrated food.

Conclusions

The use of desiccant/refrigeration hybrid systems for space air conditioning where there is a latent heat load is presently viable. Projected energy and demand charges will make the system more attractive in the future. Development of more effective desiccant materials and further improvement of components will provide a competitive alternative to vapor compression refrigeration for air conditioning buildings.

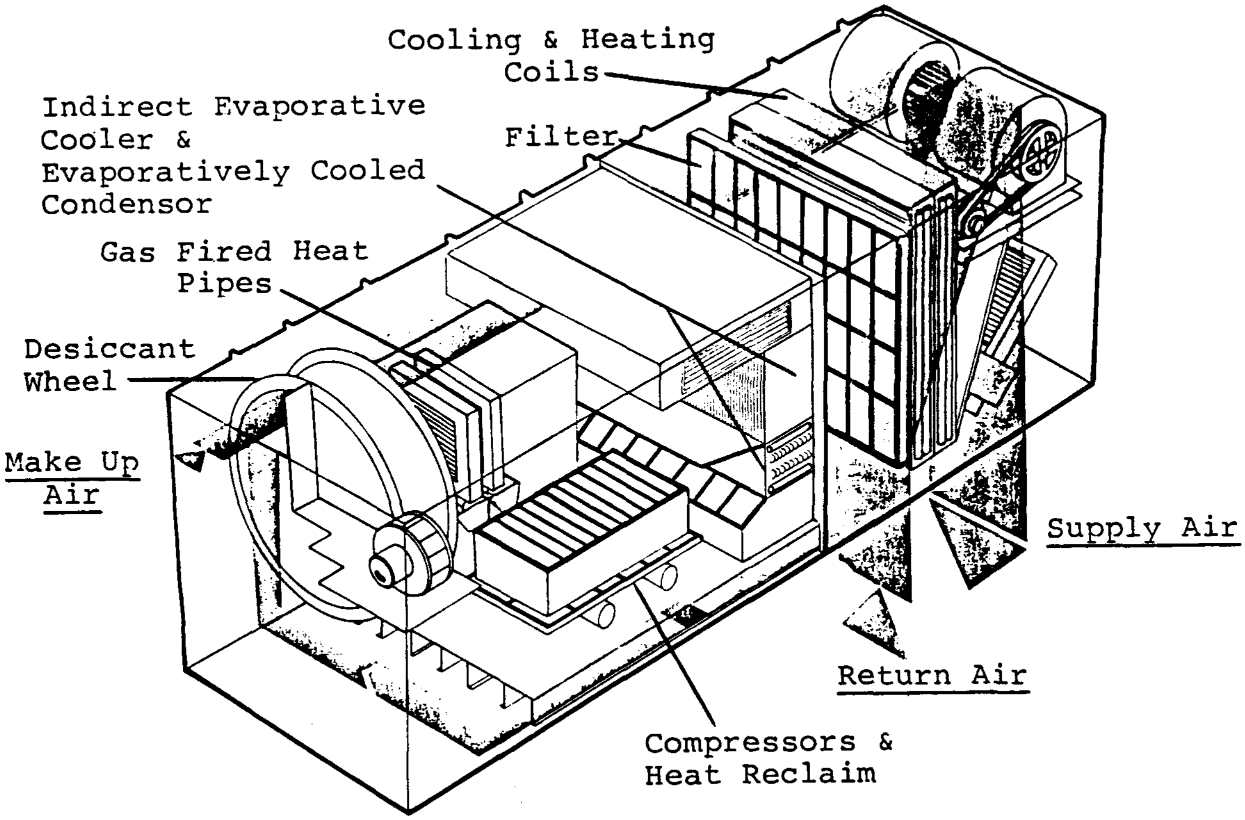


Figure 1
Desiccant/Refrigeration Hybrid
System for Air Conditioning Buildings

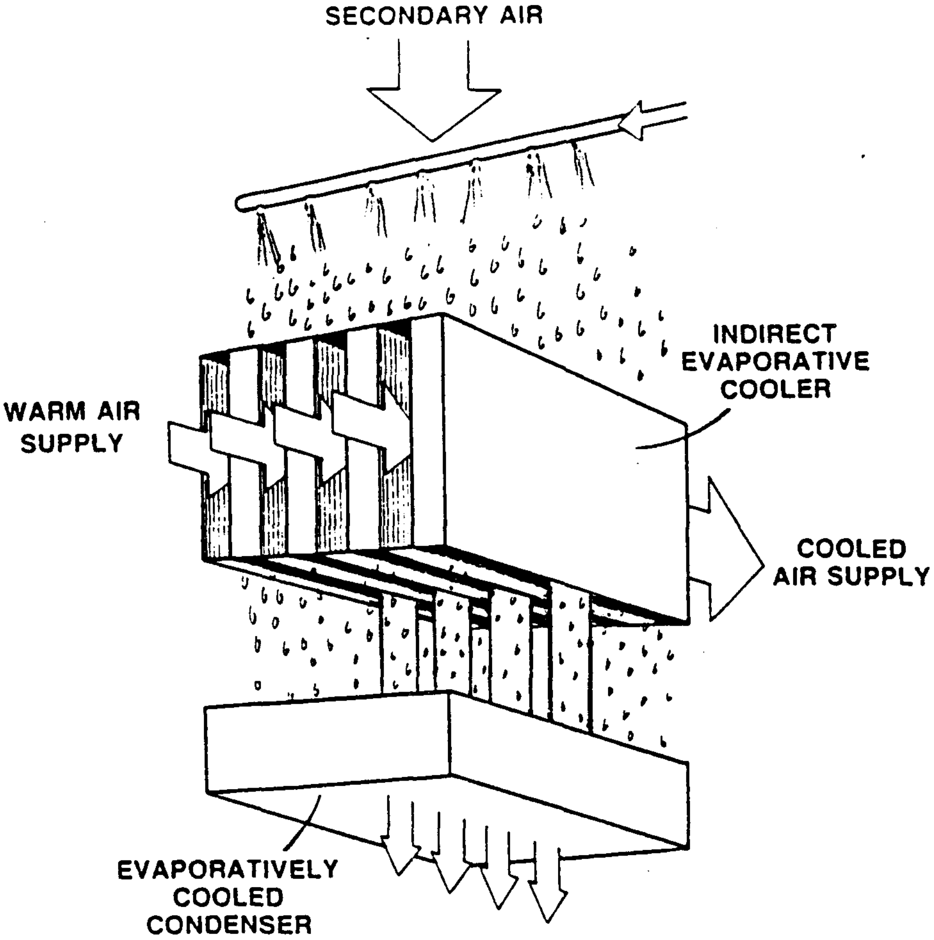


Figure 2
Intergration of Indirect Evaporative Cooler and Evaporatively Cooled Condensor

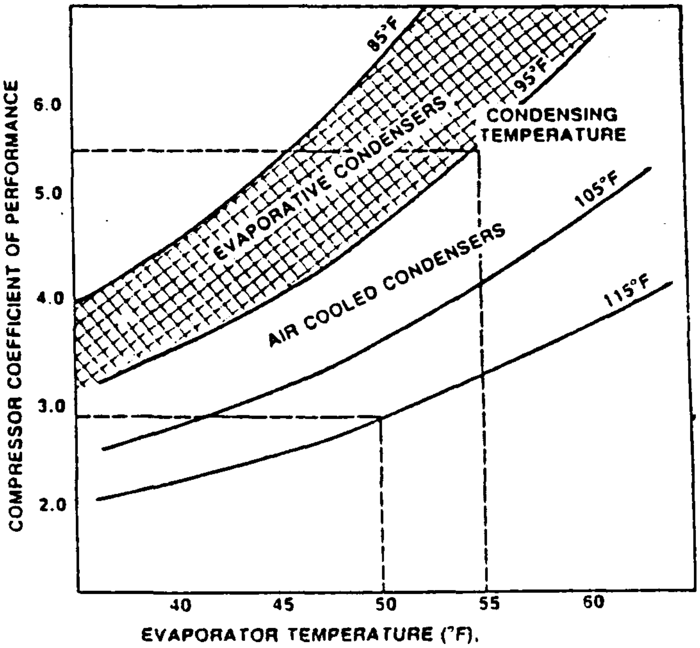


Figure 3
Compressor Performance Characteristics

RESEARCH SUMMARY

TITLE

FIELD DEVELOPMENT OF A
DESICCANT-BASED SPACE-CONDITIONING
SYSTEM FOR SUPERMARKET APPLICATIONS

CONTRACTOR

Thermo Electron Corporation

PRINCIPAL INVESTIGATORS

B.M. Cohen, D.L. Manley,
R. Arora and A.H. Levine

GRI PROJECT MANAGER

Doug Kosar
Project Manager/Cooling Systems

REPORT PERIOD

February 1982 to June 1984
Final Report

INTRODUCTION TO THE EXTRACT AND SUMMARY

This material is an extract from the Gas Research Institute (GRI) Project Report 84/0111. It was edited and produced by Cargocaire Engineering Corporation, 79 Monroe Street, Amesbury, MA 01913. It is intended as a relatively brief summary to acquaint the Design Engineer with the key differences between supermarket air conditioning and standard commercial HVAC designs and load patterns.

This extract does not include all of the conclusions of the original report, nor does it contain the detailed equipment and installation descriptions of the original report. Copies of the complete report can be obtained by writing directly to the National Technical Information Service (NTIS) at the U.S. Department of Commerce, 5285 Port Royal Road, Springfield, VA 22161 (703/487-4650) and requesting document PB85-101921. Microfiche copies are $4.50 and paper copy prices vary.

OBJECTIVE

This program was initiated to provide an assessment of the potential benefits of using a hybrid system that combines a gas-regenerated desiccant dehumidifier with electric sensible cooling for supermarket space conditioning.

TECHNICAL PERSPECTIVE

ligh latent to sensible
oad ratio favors use
f desiccants.

Supermarkets are recognized as having relatively high latent-to-sensible cooling loads because of the presence of food refrigeration equipment which performs significantly more sensible cooling than latent. The inefficient process by which commonly used electric vapor-compression air conditioners dehumidify suggest that the use of desiccant dehumidifiers to perform latent cooling in supermarkets may be advantageous.

An electric vapor-compression air conditioner must sensibly cool air to a temperature below the air's dewpoint before moisture can be removed. This typically means overcooling the air to condense the moisture and then reheating it to maintain a comfortable condition.

A hybrid space-conditioning system which couples a gas-regenerated desiccant dehumidifier to an electric vapor-compression air conditioner handles the latent and sensible components of the cooling load separately. This system offers several advantages. Because the electric air conditioner performs sensible cooling only, no overcooling is done and the requirement for reheat is eliminated. Also, the electric air conditioner's evaporator temperature may be raised, resulting in an improved efficiency, because temperatures below the air's dewpoint no longer have to be achieved. Lower circulation flow rates are also permitted. The use of an air-to-ambient air heat exchanger to post cool between the desiccant dehumidifier and the electric air conditioner actually reduces the load being handled by the hybrid system by rejecting sensible heat to the ambient. Hybrid systems may also be used to maintain store humidities below conventional levels, thus reducing the load on the store's food refrigeration system.

RESULTS

educed store
umidity lowers
verall operating
osts.

Two field installations of gas-fired desiccant dehumidifiers coupled to conventional electric air-conditioning systems were studied. Savings in food refrigeration operating costs were observed with reduced store humidities at both sites. It was also demonstrated that circulation flow rates lower than those of conventional vapor-compression systems could be used.

A computer program to simulate a supermarket's loads and equipment performances was developed and validated with field data. It was then used to evaluate the hybrid space-conditioning concept relative to a conventional system in different climates and at conventional and reduced store humidity level operation. At conventional humidity levels, the concept was viable with annual operating cost savings up to $8000 with local 1983 energy prices in regions of moderate to high humidity. Immediate savings or less than two-year payback periods were shown to be attainable in many of these areas, with the primary savings being reduced air-conditioning and fan operating costs and peak electric demand reductions.

Refrigerant-desiccant hybrid system reduce other installation costs.

The installed cost of a hybrid system in a new application is less than that of a conventional electric vapor-compression system because of the downsizing of the vapor compression component and the reduction of circulation airflows allowing for savings in fan and ductwork costs. Thus, a desiccant hybrid system in a new application offers immediate savings. For a replacement application, the installed costs are comparable, and acceptable payback periods of less than two years can be attained. Retrofit applications of a dehumidifier alone are not practical in most cases because the installed cost of the dehumidifier must be amortized by operating cost savings which, given current equipment pricing, make acceptable payback periods difficult to achieve. The remaining value of any displaced vapor compression capacity must also be factored into the retrofit economics.

Desiccant reactivatio condensor heat reclaim offers additional savings.

As a consequence of these results, the continuation of support for the development of improved efficiency desiccant dehumidifiers and integrated cooling systems is recommended to increase the economic viability of the hybrid system concept and thus increase the potential market size. Improvement can be made in the desiccant dehumidifier's COP, the method and effectiveness of postcooling the process air stream before it is cooled by the electric air conditioner, and the parasitic power requirements. Particular consideration should be given to the ideas of using reclaimed refrigeration condenser heat in conjunction with gas heating or heat produced by gas-fired congeneration for desiccant reactivation.

PROJECT IMPLICATIONS

Success of evaluation has lead to follow on projects.

Based on the results of the completed evaluation of the industrial dehumidification equipment as adapted to the commercial space-conditioning application in supermarkets, a separate project, also under GRI sponsorship, was initiated in November of 1983 to increase the economic viability of the integrated cooling concept. Cargocaire Engineering Corporation, a leading dehumidification equipment manufacturer, is GRI's cost-sharing partner in the follow-on project. Targeted component and system developments are projected to yield a higher efficiency gas-fired dehumidifier with improved post cooling and electric vapor compression sensible cooling in an integrated package for commercialization in 1985. Other commercial applications besides the supermarket will be assessed for economic application of the cooling concept during the course of the follow-on effort.

SUMMARY

INTRODUCTION

This report describes a program conducted by Thermo Electron to evaluate the potential for the use of hybrid systems that combine gas-regenerated desiccant dehumidifiers with electric vapor compression sensible cooling for supermarket space conditioning. The relatively high latent-to-sensible cooling load typical of supermarkets, because of their food refrigeration equipment, and the inefficient process by which commonly used electric vapor-compression air conditioners dehumidify suggested that the use of desiccant dehumidifiers might be advantageous. Thus, the primary objective of the program was to evaluate desiccant hybrid space-conditioning systems and identify all of their potential benefits.

An electric vapor-compression air conditioner must sensibly cool air to a temperature below the air's dewpoint before moisture can be removed. This typically means overcooling the air to condense the moisture and then reheating it to a temperature suitable for maintaining comfort. This inherently inefficient situation is common in supermarkets because of the presence of refrigerated food cases which perform significantly more sensible than latent cooling. Trends in store design which include more food refrigeration and more efficient lighting will reduce the sensible component of the load even further. Hybrid space-conditioning systems which use a desiccant dehumidifier, handle the latent and sensible loads separately. A desiccant dehumidifier converts the latent load to a sensible load, of which most may be rejected to the environment through air-to-air heat exchangers and the remainder of which can be removed with electric air conditioners. Indirect evaporative cooling can also be used to meet the sensible load. The excessive sensible cooling and requirement for reheat of a vapor-compression system are eliminated.

Desiccant-refrigerant hybrid design is optimum for supermarket load profile.

Two approaches which have potential benefits to supermarkets may be taken. One is to operate a hybrid system at conventional humidity levels and realize savings through reduced air-conditioning and reheat energy costs. The second approach is to operate at lower humidities to reduce the load on the food refrigeration system and thus its energy consumption. Because of the additional latent load being met with the second approach, little if any, savings are possible in air conditioning costs with currently available equipment. Both approaches allow for reduced circulation flow rates and higher air conditioner evaporator temperatures. Thus, savings can be attained through a reduction in fan operating costs and air conditioner efficiency improvements.

Lower humidity is key to operating economies.

Electric and gas utilities could also benefit from the use of desiccant dehumidifier hybrid systems. Electricity sales would be replaced by gas sales during the summer. This is a period when gas sales are low and electric utilities are having difficulties satisfying their peak demands.

Peak electrical loads reduced.

Thermo Electron has studied both approaches and identified the conditions necessary for economic feasibility of the hybrid system concept. Recommendations have also been made for the continuation of work to further develop and improve hybrid systems.

PROGRAM DESCRIPTION

The program to assess the benefits associated with the use of gas-fired desiccant dehumidifiers in supermarket space-conditioning systems was conducted over a 2-year period. The first year involved the evaluation of a field installation of a hybrid desiccant dehumidifier space-conditioning system in West Chicago, IL. In the second year, another installation in San Antonio, TX was included. During the course of the program, time was spent developing in an evolutionary manner, and finally using for parametric analysis, a computer simulation of the loads and equipment performances characteristic of a supermarket.

Two years of operating test data.

Field tests confirm practical operating considerations.

The primary objectives of the field work were to acquire a better understanding of how desiccant dehumidifiers interact with the different systems in a supermarket, to confirm the theories of reduced space-conditioning and food refrigeration energy consumptions, and to develop criteria for the design and installation of desiccant hybrid systems. Although both field sites involved gas-fired desiccant dehumidifiers coupled to existing air conditioning systems, the two were different in many respects. The West Chicago installation in a Jewel/Osco store used a gas-fired desiccant dehumidifier, in a ventilation mode, directly firing into a silica gel packed bed as the desiccant. The climate was moderately humid in the summer and the store was not open 24 hours a day. A complete description of this installation may be found in "Field Development of a Desiccant-Based Space-Conditioning System for Supermarket Applications," Annual Report, GRI Report No. 83/0013 (NTIS document No. PB83-251017). On the other hand, San Antonio's climate is very hot and humid for several months each year and the HEB store in which the hybrid system was installed is open 24 hours a day. The HEB system operated in a recirculation mode with no ventilation air and the dehumidifier used indirectly fired lithium chloride held in a honeycomb flow channel structure as the desiccant. These differences allowed for a broader assessment of the hybrid space-conditioning system concept.

Computer model developed for comparison with observed data.

A computer program was developed to provide a means for a complete evaluation. The simulation allowed for the evaluation of hybrid and conventional space-conditioning systems at the varying operating conditions experienced during an entire year. Moreover, the effects of such variables as climate, ventilation requirement, sensible/latent load ratio, building characteristics, and refrigeration case capacity could be studied efficiently. The program performs an hour-by-hour quasi-steady-state analysis using sequential hourly weather data. It calculates steady-state loads and equipment performances using equations developed from field experience and manufacturer's specifications. A certain number of assumptions had to be made. Their appropriateness was validated with data collected from the Jewel and HEB stores.

The two sources, field data and computer simulations, provided the information necessary to thoroughly identify the benefits associated with the use of gas-fired desiccant dehumidifiers in supermarket space-conditioning systems. Areas where improvements were possible could also be outlined and thus recommendations for future development work could be made.

WORK COMPLETED

1st field test in Chicago at Jewel/Osco.

Upon selection of a Jewel/Osco store in West Chicago, IL as the site for the first field evaluation, work began with the preparation of plans and specifications for the installation of a desiccant dehumidification system. A design point analysis was performed to determine the size of the desiccant dehumidifier to be coupled to the store's conventional air conditioning system. Related controls, instrumentation, air-handling equipment, and data acquisition equipment were also specified. Assistance was provided by Jewel engineers and E.J. Flynn Engineers, Inc., an A/E division of Thermo Electron.

Upon completion of the installation and checkout of the system and instrumentation in mid-June of 1982, a test program involving alternate periods of desiccant hybrid system and conventional system operation was conducted until mid-October. Energy consumption data were manually recorded daily, and indoor and ambient temperature and humidity conditions were recorded continuously on hygrothermographs. An event recorder was also used to monitor the cycles of various space-conditioning system components.

equired installation
nprovements
dentified.

During this field evaluation period, several problems and areas where improvements could be made were identified. Ventilation airflow rates were higher than intended because of leaky dampers and improper enthalpy control. Infiltration rates were also high because compressor room fans drew air from the store, not allowing it to be pressurized. A significant amount of stratification was present which increases the apparent load on the system. Because these loads had not been properly estimated, the desiccant hybrid system was unable to hold the desired conditions. The operation of the automated energy management control system was often erratic, turning equipment on and off improperly. The ductwork was insufficiently sealed and thus allowed moisture to leak into the dry air stream. Manual data recording was inconsistent and the instrumentation was not always accurate.

Because of the problems encountered and the relatively short test period, the amount of reliable data collected was limited. For this reason, the evaluation program was extended into 1983 to include modifications to the store and hybrid space-conditioning system and testing for another cooling season. The changes and modifications made included improved ventilation air dampers, under-the-case returns, additional data acquisition equipment, and the hiring of a local engineering student for reliable data recording.

econd test
nstallation at
IEB store in
an Antonio, TX.

Also included in the program extension was the evaluation of another installation of a desiccant dehumidifier hybrid space-conditioning system. This system had been installed in an HEB store in San Antonio, TX by Cargocaire Engineering Corporation. The store was already well instrumented because a DOE field test program to evaluate an unequal parallel compressor concept for the food refrigeration system was being conducted concurrently. This provided Thermo Electron with the opportunity to evaluate another hybrid system in a very different climate and perform a broader assessment. Some additional instrumentation was installed at HEB to monitor the actual load on four medium-temperature cases (low temperature was already installed) and the performances of the space-conditioning equipment. A 4-month test program was then conducted at HEB during the 1983 cooling season. As at Jewel, the program included alternate periods of conventional and hybrid system operation. The hybrid system was operated with various store humidity and temperature set points and with a reduced circulation air flow rate.

The data collected at the two field sites during the 1983 season indicated the importance of sealed ductwork, process air postcooling, proper control, and improved efficiency dehumidifiers. Anticipated reductions in refrigeration operating costs with reduced humidities were verified and the potential for substantial savings through system optimization was indicated. It also became evident during the field evaluations that stores are unique and climate has a significant effect on the economics. Although the results of the two field evaluations provided information which was important and helpful, it was not enough to completely assess the benefits of gas-fired desiccant hybrid systems.

Computer program
xtends utility of
bserved data.

A computer program was developed by Thermo Electron which simulated a store's loads and associated space-conditioning and refrigeration equipment performances on an hour-by-hour basis for an entire year. A conventional electric vapor-compression system was modeled as well as hybrid systems operating in ventilation and recirculation modes. Hourly average weather data for nine different cities representing a range of climates were used. Comparisons were made between the store's annual operating costs with conventional and hybrid systems. The sensitivity of the costs to such variables as store humidity set point, ventilation requirement, and refrigeration capacity was also identified.

To enable the estimation of payback periods for gas-fired desiccant dehumidifier hybrid space-conditioning systems, installed costs were estimated for the hybrid and conventional systems. Industry standards and the experience of E.J.

Flynn Engineers and Cargocaire Engineering Corporation were used. All comparisons were made on the basis of new or replacement applications as opposed to a retrofit installation.

To complete the evaluation program a number of conclusions were derived from the results of the computer simulations and the information obtained during the field studies. Several recommendations for future work were also outlined. The more important of these will be presented in the following section. Finally, a set of guidelines was developed to assist the supermarket engineer who is considering the use of a gas-fired desiccant dehumidifier hybrid space-conditioning system. These guidelines have been included as an appendix to this report.

MAJOR CONCLUSIONS AND RECOMMENDATIONS

Many conclusions were arrived at during the course of the evaluation program. These are discussed in more detail in Chapter 11, but the most significant conclusions will be presented briefly here.

Results confirm beneficial effect of lower humidity level made possible with desiccants.

As had been expected, reduced store humidities reduce the load on the food refrigeration systems and thus the operating costs. This was seen at both field sites. It was concluded from a test at the HEB store that desiccant hybrid systems can hold store conditions including humidities below those of conventional systems with lower circulation air flow rates than used with electric air conditioning. This can produce substantial savings in fan operating costs. Both field installations also indicated the importance of sealed ductwork to prevent moisture leakage and of post-cooling the hot, dry process air to reduce the load on the electric air conditioner.

Both moderate and high humidity climates offer potential economies for desiccant-refrigerant hybrid.

The primary conclusion of the numerous computer simulations performed is that the concept of using gas-fired desiccant dehumidifiers in hybrid space-conditioning systems in supermarkets is currently viable in areas of moderate and high humidities. Immediate savings or simple payback periods of less than two years are possible when operating the hybrid system to maintain humidity levels comparable to those of conventional systems in all new and most replacement applications. However, an even more important conclusion is that the improved desiccant dehumidifier efficiencies and summer gas price reductions would significantly increase the potential savings of a hybrid system, consequently improving its marketability. Also, modern, energy conserving store designs have reduced sensible cooling loads, which further increases the savings potential of a hybrid system.

Optimized dehumidifier design required for greatest economies.

Currently, at low humidities, the savings in refrigeration operating costs are less than the additional electric air-conditioning and dehumidifier gas and fan energy costs. In the future, with more efficient dehumidifiers and post coolers and advanced gas-fired desiccant cooling systems that can operate in the ventilation mode with very high COP's, it would become more advantageous to operate at reduced store humidities. Moreover, if the summer gas cost is reduced as an incentive in the future, overall savings will further increase. The trend towards larger frozen food departments and thus more refrigeration cases will also strengthen the argument for reduced humidities.

These conclusions led to the recommendation that development work presently being sponsored by GRI be continued to improve dehumidifier efficiencies and optimize integrated hybrid systems. It is also suggested that gas utilities promote incentive programs to make it more advantageous for consumers to use gas in place of electricity during the summer.

GENERAL DISCUSSION

This chapter is intended to provide some of the necessary background information that will be useful to a person unfamiliar with desiccant-based space conditioning. The distinguishing characteristics of the space-conditioning loads in supermarkets and methods of space-conditioning that take advantage of these characteristics are outlined here. Some discussion of the refrigeration and space-conditioning equipment in supermarkets is also included. Approaches to reduce operating costs of the equipment are indicated.

CHARACTERISTICS OF SUPERMARKET SPACE CONDITIONING

The space-conditioning loads on a supermarket are unique in that the latent-to-total load ratio is higher than for most other commercial buildings. This is because the refrigerated cases act as open air coolers, absorbing predominantly sensible energy from the store and rejecting it to the outside of the building through the condensers. Figure 1. presents the typical load profile for a supermarket in Miami which is open 24 hours a day. The latent-to-total load ratio is at least 0.4 most of the time.

High ratio of latent to sensible heat in supermarkets.

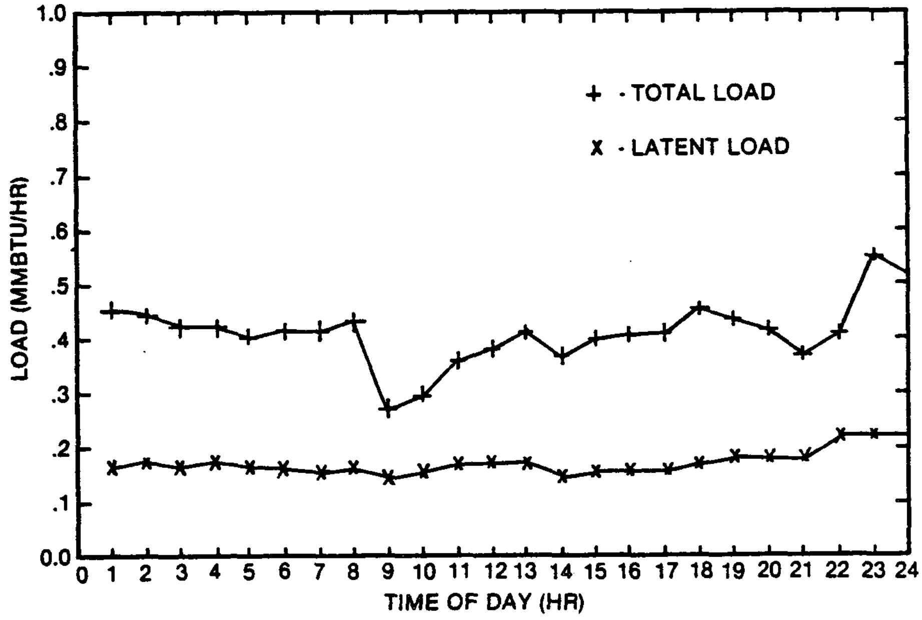


Figure 1 Hourly Latent and Total Load Variations on Space-Conditioning System, Miami-June

Engineers involved with the design of HVAC and food refrigeration systems for supermarkets are well aware of the unusual load patterns resulting from the interaction of these systems. The space-cooling effect of the refrigerated cases can provide all of the sensible cooling required on the hottest days, and may result in a need for heating during periods when other conditioned spaces may require cooling.

Because the cases produce more sensible cooling than latent, a sizable latent load can remain. Handling such loads using a conventional air conditioning system alone can be inefficient. To remove moisture from the air, the air must be cooled to below its dewpoint. This is typically lower than is required to meet the sensible load alone, necessitating re-heating of the air before it can be distributed to the store. This process, shown in Figure 2. becomes increasingly inefficient when handling higher latent-to-total load ratios.

Lower humidity means operating inefficiency for vapor-compression refrigeration.

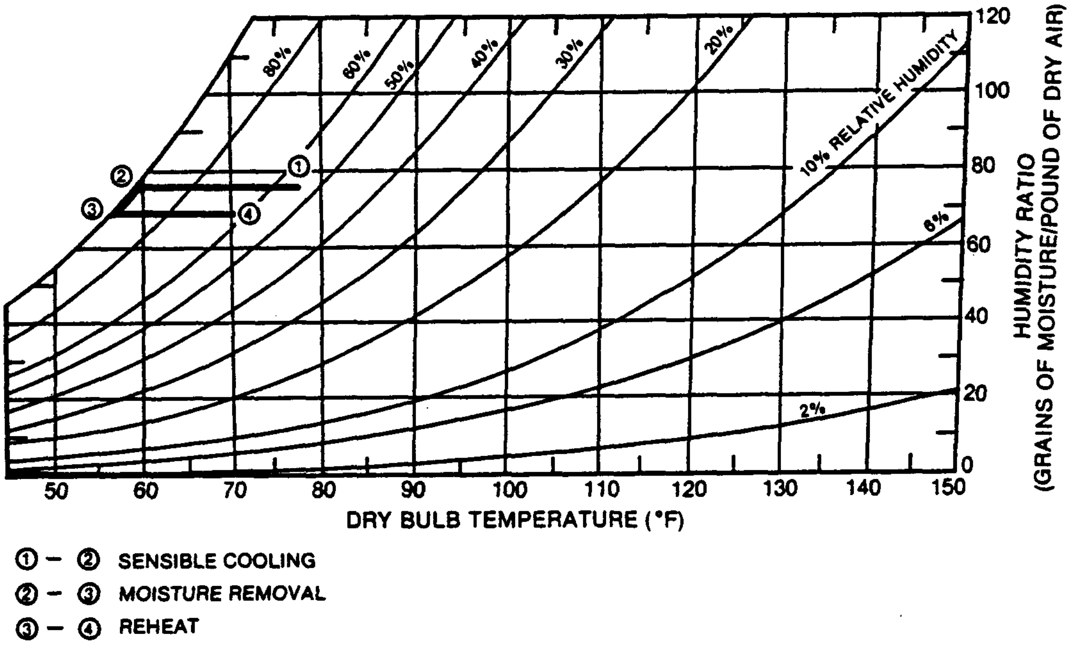


Figure 2 Typical Vapor Compression Air-Conditioning Process

Such space-conditioning systems operate more efficiently when they perform sensible cooling only, eliminating the reheating requirement. Also, since the air would not have to be cooled to below its dewpoint, higher evaporator temperatures can be allowed. This would result in higher operating coefficients of performance (COP's). These factors suggest a potential advantage in decoupling the sensible and latent cooling, using an alternative method for latent cooling.

One method for achieving predominantly latent cooling is through the use of specially deisgned low-temperature air conditioning coils, such as those developed by Trane Corporation. According to the manufacturer's specifications, these coils are designed to have higher latent-to-total cooling ratios than standard coils. A drawback of such an approach appears to be the inability to attain very low humidities, if such are desired. However, at this time insufficient information is available on this method to discuss it in greater detail.

Another method of performing latent cooling alone is through the use of sorbents, materials capable of removing moisture from their surroundings. Essentially, a sorbent causes some of the water vapor in an air-water vapor mixture to condense, releasing the latent heat of sorption which is typically 10 to 15 percent greater than the latent heat of condensation. This causes the air temperature to rise. This process may also be viewed as converting the latent heat of the air-water vapor mixture to sensible heat. In terms of loads on the space-conditioning systems, this is similar to converting latent load to sensible. The hot dry air can now be sensibly cooled, using conventional air conditioners, air-to-air heat exchangers, indirect evaporative coolers, etc. The process is shown on a psychrometric chart in Figure 3.

Desiccants remove water vapor in the vapor phase by sorption, not condensation.

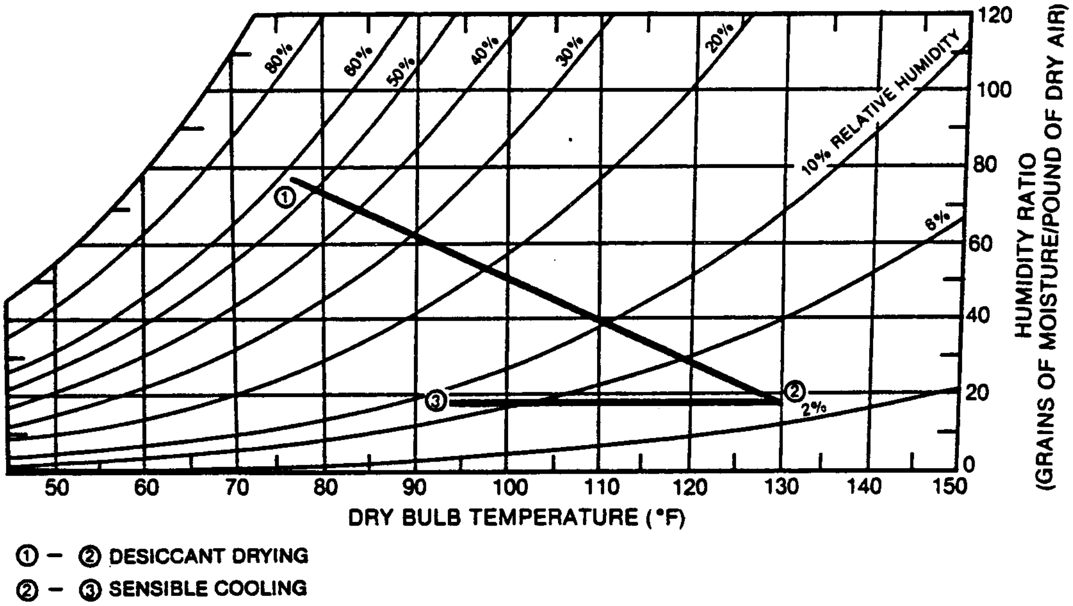


Figure 3 Desiccant Cooling Process

SORPTION DEHUMIDIFICATION

Dehumidification of air refers to the reduction of its moisture content. Sorbent materials used in dehumidification equipment may be either the liquid or solid type. The performance of different types of sorption dehumidification machines is, to some extent, a function of the sorbent used. These materials must be capable of removing the sorbate (such as water) from the fluid stream being treated.

Absorbents undergo physical or chemical change during sorption process.

Sorbents used in dehumidification equipment can be divided into two classifications; absorbents and adsorbents. A sorbent which changes either physically, chemically, or both during the sorption process is known as an absorbent. Lithium chloride is an example of a solid absorbent. When water is absorbed on this material, the lithium chloride is converted into hydrates, eventually reaching a saturation point after which additional moisture causes the material to lose its crystalline structure and dissolve in the water that was absorbed. A sorbent which does not change physically or chemically during the sorption process is known as an adsorbent. Some common adsorbents are silica gel, activated alumina, and activated charcoal. A solid adsorbent, also known as a desiccant, has the property of adsorbing moisture from, or adding moisture to, a gas such as an air-water vapor mixture. Actually all gases and vapors are adsorbed to some extent on an adsorbent surface. In the case of an air-water vapor mixture, the amount of air adsorbed is very small compared to the amount of water adsorbed, and consequently, dehumidification of the air occurs.

Adsorbents do not change during sorption process.

Adsorption takes place at the surface of the adsorbent, where the gas and solid come in contact with each other. Common solid adsorbents have a porous structure of submicroscopic dimensions, which gives them an extensive internal surface area. The amount of material adsorbed under equilibrium conditions is proportional to the surface area of the desiccant, if the pores are large enough to allow the adsorbate to penetrate into the interior of the desiccant.

Both processes are reversible through the addition of heat.

For both solid adsorbents and absorbents, the sorption process is completely reversible by application of heat and dry gas, such as air. In the case of solid adsorbents, reactivation temperatures are generally within the range of 150 F to 500 F.

Because no physical or chemical change occurs to the adsorbent in this process, it is again ready to extract water vapor from a wet gas after reactivation.

Both liquid and solid sorbents may be used in dehumidification equipment. Because liquid sorption equipment is generally bulky and expensive, this method was judged to be impractical for supermarket applications.

Dehumidification by use of a solid desiccant may be performed under static or dynamic conditions. In the static method, there is no forced circulation of air through the desiccant. Instead, the air immediately surrounding the desiccant is dried, and through convection and vapor diffusion, water vapor from more remote areas reaches the desiccant, where it is adsorbed. This process can take considerable time to reach equilibrium.

Dynamic dehumidification indicates that the air being dried is flowing through a desiccant bed. For space-conditioning applications, therefore, an air-moving device would be an essential part of the dehumidifier.

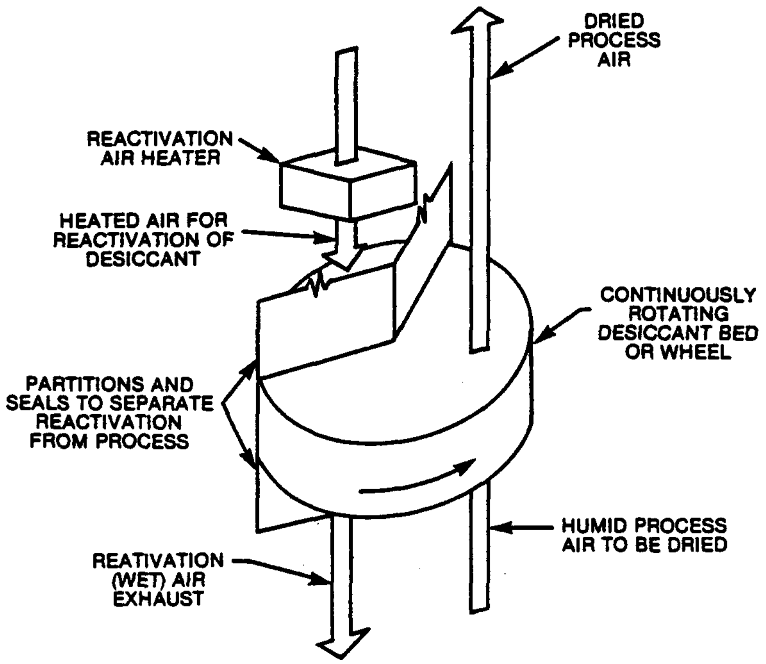


Figure 4 Typical Rotary Dehumidification Unit

A common type of solid adsorption dehumidifier is the rotary-bed dehumidifier. Rotary-bed dehumidifiers, illustrated in Figure 4. have one or more beds of solid desiccant. The beds are usually in counter air flow and each bed handles air to be dried and regeneration air simultaneously. The two air streams are separated by seals to prevent mixing, and the beds are physically rotated within the casing to expose one portion of the desiccant to the process stream and the remainder to the regeneration stream. Since a rotary dehumidifier has a continuously rotating bed, outlet conditions will be constant when the inlet conditions are held constant.

Figure 5. shows the performance of a typical rotary-bed dehumidifier. As would be expected, as the inlet temperature and humidity rise, so do the outlet temperature and humidity. Such devices can be used to obtain fairly low dew-points in a closed space. The actual performance depends on the type of desiccant used, method of regeneration, inlet air conditions, and other factors.

Rise in dry bulb temperature of the air being dried is a natural consequence of sorption.

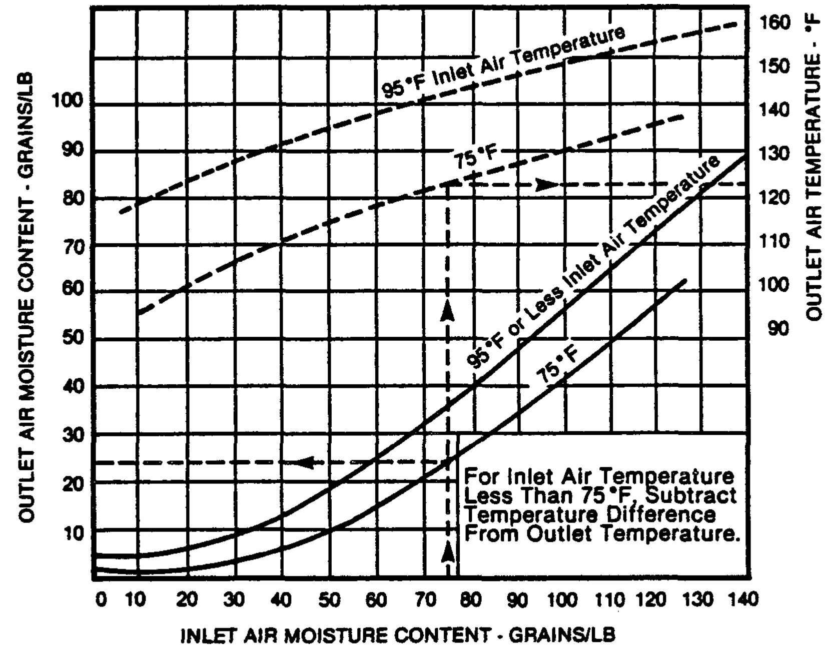


Figure 5 Typical Performance of a Rotary Bed Absorbent Dehumidifier*

SUPERMARKET APPLICATIONS

Both desiccant and refrigerant technology required for maximum system efficiency.

As discussed in the Section on Characteristics of Supermarket Space Conditioning, the unusually high latent-to-total load ratios often encountered in supermarket space conditioning can make desiccant-based space-conditioning systems an economically attractive option. A hybrid system, consisting of a desiccant dehumidifier and a conventional air conditioner, can be efficiently used to separately handle the latent and sensible components of the total space-conditioning load. Such a system can be configured in several ways; the best configuration for any application would depend on the characteristics of the space conditioning and refrigeration systems, loads on the store, and building codes.

One possible configuration is illustrated in Figure 6. some of the recirculation air is mixed with the ventilation air and the mixture is passed through the dehumidifier and the heat exchanger in series. This warm, dry air is then mixed with the remainder of the recirculation air and is passed through the air conditioner which cools the air sufficiently to meet the sensible load on the space.

The configuration can be operated in three different modes. *Ventilation mode* refers to no recirculation air being mixed with the ventilation air prior to being passed through the dehumidifier. *Recirculation mode* refers to the case as illustrated, with the mixture of ventilation and recirculation air being passed through the

---

*"The Dehumidification Handbook," The Cargocaire Engineering Company, 1982.

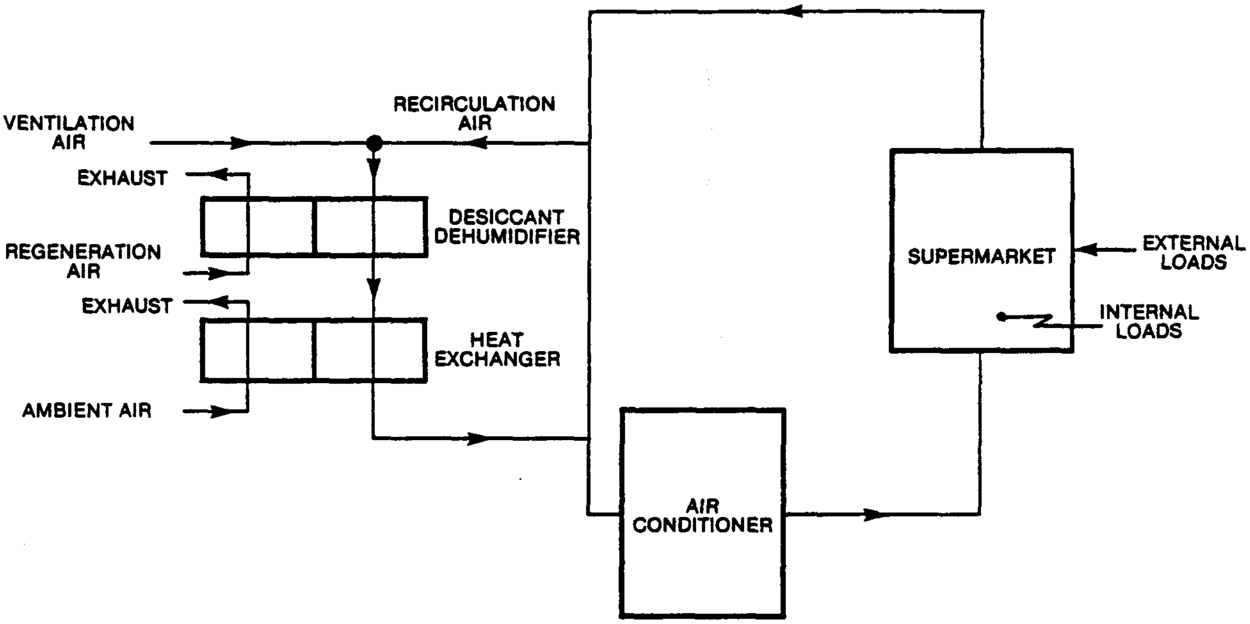


Figure 6 Hybrid Space-Conditioning System

dehumidifiers. The third mode of operation is *pure recirculation* with only recirculation air being dried by the dehumidifier. In the recirculation mode, the optimum mix of ventilation and recirculation air will depend on many factors, including ambient conditions, desired indoor conditions, local building codes, and the performance characteristics of the equipment.

In general, the energy consumption per unit of water removed decreases as the inlet humidity rises. This is intuitively apparent since it would be easier to remove moisture from a more humid stream. However, as was shown in Figure 5. for very high inlet humidities the outlet humidity may not be low enough to fully meet the latent load. This must be considered in the system design. In most cases, however, the most economical mode of operation would be the one using the minimum amount of ventilation air required by the local building code and desired indoor comfort levels.

Vapor compression refrigeration has low latent capacity requiring large airflow to remove latent load.

Another important characteristic of the hybrid system is that the total air-circulation rates required are lower than those for a conventional space-conditioning system. Because a conventional air conditioner has a low latent-to-total capacity ratio, high air circulation rates are required in order to meet the latent load. Since for a hybrid system the latent and sensible loads are met separately, the necessary air circulation rate is lower. In new installations this allows the use of smaller air-distribution ducts and circulating fans. These savings can make the installed cost of the hybrid system lower than that of a conventional system.

Low-cost heat can assist desiccant regeneration.

Virtually any source of heat at a sufficiently high temperature can be used for regeneration. Lower regeneration temperatures will allow more efficient use of the thermal input. However, for the same capacity the required size of the dehumidifier will be significantly larger and the parasitic electric power consumption by blowers handling more air will be greater. Some potential regeneration energy sources are refrigerant condensers, cogenerators, solar collectors, process waste heat, and natural gas heaters. Reclaim heat from the food refrigeration or the conventional air conditioning system condensers can be an economical means of preheating the regeneration air, but will generally be at too low a temperature to be the only heat source. The exiting regeneration air may also be at a high enough temperature to permit some preheating of the incoming regeneration air.

In addition to the ability to perform space-conditioning in an efficient manner, the use of desiccants can have another very important advantage. Because the desiccant can dry the air to very low humidities, the minimum space humidity attainable can be much lower than that possible with a conventional system.

To understand the significance of this, it is necessary to examine the effects of lowering the indoor humidity. The primary effect of lowering the indoor humidity is on the performance of the food refrigeration system. This is discussed in the next section.

REFRIGERATION EQUIPMENT

Product-cooling equipment in supermarkets is responsible for unique excess sensible space-cooling capacity.

It is the presence of a disproportionate amount of food refrigeration equipment that gives supermarkets their unique space-conditioning characteristics. The refrigeration equipment can account for 50 percent of the total energy consumption of the store, so a reduction in the load carried can result in significant savings.

The energy gains that make up the load on the refrigerated cases may be divided into four main components: (a) conduction heat gains through the walls, (b) heat gains associated with the products brought into the refrigerated space, (c) heat gains associated with lights, electric motors, and other internal heat sources, and (d) heat gains due to convective and diffusive flow into the refrigerated space. Of these four components, (b) and (c) are relatively independent of store conditions, and (a) depends on store temperature only. Component (d), however, depends on the sensible and latent energy content of the air entering the refrigerated space. Figure 7. shows the probable air movements in and around typical refrigerated cases. The recirculation currents within the case are maintained by fans, and form an air-curtain between the refrigerated space and the store air. This air-curtain significantly reduces the amount of store air entering the refrigerated space. Still, a substantial amount of store air does enter to make the load associated with it on the order of 85 percent of the total load that the compressors must handle.

Store air accounts for 85% of case refrigeration system load.

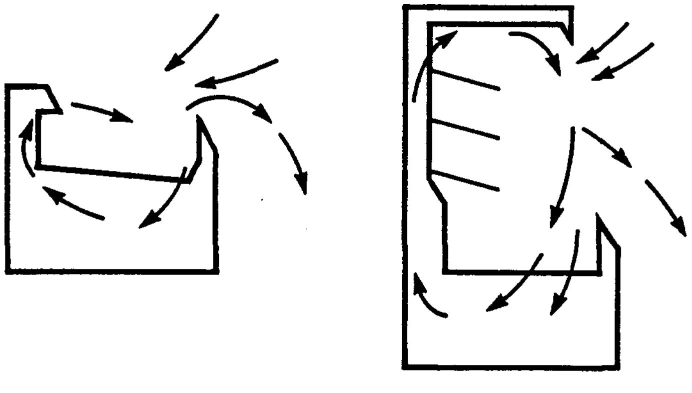


Figure 7a Typical Medium-Temperature Cases

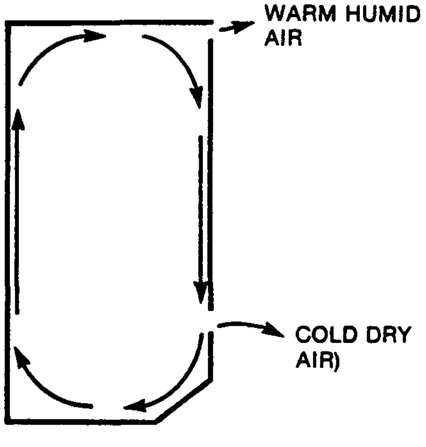


Figure 7b Low-Temperature, Glass Door Case Air Patterns (When Door is Open)

For the closed door cases, air exchange with store air occurs only when the case door is opened. Therefore, the proportion of the compressor load associated with the exchange may be expected to be smaller than for the open cases.

Results obtained by the Tyler Company indicate that a reduction in the relative humidity of the store air can reduce the loads on the refrigerated cases. A +75°F reduction from 55% RH to 42% RH was reported to have reduced the load by 10 percent and 7 percent for the medium and low temperature cases respectively. A +75°F reduction from 55% RH to 25% RH reduced the load by 30 percent and 20 percent respectively. These results provide a useful estimate of the refrigeration savings associated with reduced store humidities.

Product display case manufacturer confirms case load reduction at lower store humidities.

Reduction in store humidity can further reduce refrigeration system energy consumption through slower frost buildup on the refrigeration evaporator coils. Slower frost buildup allows operation of the cases with fewer defrosts, reducing the cooling required to recool the refrigerated space. The frequency of defrosts can be adjusted either manually or automatically, e.g., by using frost sensors. Also, fewer defrosts will allow better control of the space temperature, improving the shelf life of the products. Alternately, if the defrost frequency is maintained the same, then the average amount of frost of the coils will be lower. Since the frost covering the coil essentially functions as an insulator, less frost buildup will allow higher evaporator temperatures and hence higher COP's for the refrigeration system. Finally, lower store humidities will allow reduced operation or possibly elimination of the anticondensate heaters in the glass doors of the closed refrigerated cases.

Substantial additional benefits to lower humidity.

STORE AIR STRATIFICATION

The refrigerated cases in a supermarket remove moisture and heat from the air in the store and deposit the cooler air on the floor where a layer of cold, dry air builds up and spreads around the store. Many supermarket HVAC systems are designed to use this air to cool the store by placing return-air ducts under the cases. These ducts pull in cooled air from the aisles and recirculate it, thereby getting some "free" cooling.

This stratification can have several deleterious effects on the performance of any space-conditioning system operating on the store air. Due to stratification, the average air temperature and humidity are often maintained at lower levels than they would have been had the air been fully mixed. This typically has the effect of making the space-conditioning equipment perform more cooling than it would with fully mixed air. This is the source of many customer complaints about cold freezer aisles, which leads to reduced product sales in the profitable frozen food lines. The effect on the refrigeration equipment is a little more difficult to estimate. If destratification decreases the temperature and humidity of the air flowing into the cases, then its effect is to reduce the load on the cases. The converse is, of course, also true. The actual situation would depend on the level of stratification and the patterns of the air flow into the cases.

Customer complaints arise from excess of sensible cooling in frozen food aisles.

In a stratified store, the layers of cold, dry air on the floor would probably migrate towards openings and exfiltrate at the lower levels while unconditioned ambient air would infiltrate above it. In a destratified store, this exfiltration loss will also be lower.

SAVINGS

Use of low-cost energy (gas) is a major economic benefit to desiccant systems.

Separation of the latent and sensible space-conditioning loads using a desiccant dehumidifier coupled to a sensible air conditioner offers several potential sources of savings. Operating savings can be obtained by displacing some of the electrical consumption of conventional air conditioners by a less expensive energy source. Further operating savings are possible by displacing some of the electrical consumption of the food refrigeration system in the same manner. The actual amount of savings from each source will depend on the humidity maintained in the store. If the hybrid system is used to maintain normal indoor humidities, all the savings will be realized from the space-conditioning system. If lower-than-normal indoor humidities are maintained, the operating savings from the space-conditioning system will be lower, possibly even negative, but additional savings are available through reduction in the food refrigeration system operating costs.

Optimum humidity level varies according to store operation and load profile.

For any given application, the indoor humidity maintained should be the one for which the sum of the space-conditioning and food refrigeration operating costs is a minimum. Figure 8. illustrates how the variation of these costs can be used to determine the optimum indoor humidity. The actual optimum operating point will, of course, depend on the particular store and the loads imposed on it.

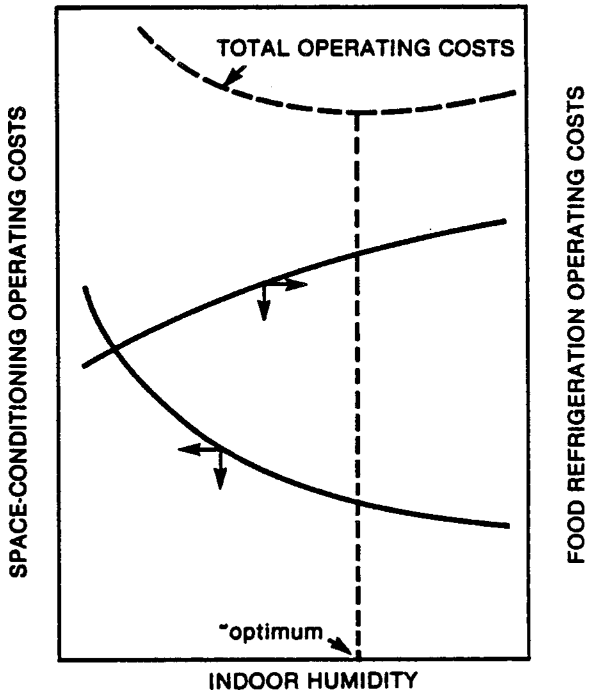


Figure 8 General Variation of Operating Costs with Indoor Humidity

Peak electrical load reduction possible through use of desiccants.

Large commercial users, such as supermarkets, usually have to pay peak electrical demand charges based on the maximum electrical demand at any time during the given month. During the cooling season, the peak demand usually occurs when the cooling loads on the store are the highest, typically at mid-afternoon. Use of the hybrid system to substitute an alternative energy source for electricity will result in a lowering of the peak electrical demand, with its associated demand savings.

Payback periods are under two years.

Because the installed cost of a hybrid system can be comparable to that of a conventional space-conditioning system, all of these sources of operating savings can result in payback periods suitable to the industry of under two years.

Use Of Liquid Desiccant Dehumidification For Cooling Buildings

- by -
William C. Griffiths
Kathabar Systems
P.O. Box 791
New Brunswick, NJ 08903
Telephone Number: (201) 356-6000

Abstract

For over 50 years, liquid desiccant humidity conditioning equipment has been used to provide air at controlled temperature and humidity for a wide variety of industrial and institutional process air-conditioning applications. Performance features of liquid desiccant equipment utilized in these applications include the following:

- Ability to simultaneously cool and dehumidify air.
- Ability to dry air to dewpoint humidities about 40°F below the conditioner coolant temperature.
- Ability to fully modulate coolant and heat use and fully control air cooling and dehumidification to match the process load requirements.
- Ability to deliver bacteriologically decontaminated air.

In the past, the relatively low cost of electric energy created little economic incentive to incorporate desiccant dehumidification equipment in comfort conditioning systems. More recently, increases in the cost of electricity and the increased impact of electrical demand charges make desiccant dehumidification considerably more attractive for these applications. Use of liquid desiccant dehumidification to handle building latent loads can reduce maximum refrigeration loads by 25 to 40%, and minimize refrigeration load variations.

The effectiveness of liquid desiccant dehumidification in reducing comfort-conditioning refrigeration loads has been demonstrated in a number of buildings. The cost-effectiveness of integrated liquid desiccant/refrigeration comfort conditioning systems can be further enhanced by improving the thermal efficiency of the regeneration process and by developing lower-cost liquid desiccant equipment designed for the commercial market. GRI-sponsored research currently planned will significantly advance toward these objectives and also provide valuable tools for modeling the performance of integrated systems in a variety of building types and climates.

INTRODUCTION TO LIQUID DESICCANT DEHUMIDIFICATION

Figure 1 is a schematic depiction of a liquid desiccant air conditioning system. The system is similar to an absorption refrigeration system in that it operates on the principle of chemical absorption of water vapor by a desiccant solution, typically a water solution of lithium chloride. In operation, air passes through the conditioner unit and is cooled and dehumidified by contact with the desiccant solution. The heat absorbed from the air by the desiccant solution is rejected to a coolant through a suitable heat transfer surface. Coolants typically used are cooling tower water, ground water, chilled water, refrigerants, or air. The conditioner unit is analogous to the absorber section of an absorption refrigeration system.

Water absorbed by the desiccant in the conditioner unit is stripped from the desiccant in the regenerator unit by heating the desiccant to elevate its water vapor pressure and contacting the heated desiccant with a scavenger airstream. The desiccant solution is heated through a suitable heat transfer surface by steam, hot water, gas or oil firing, or refrigeration condenser heat. The regenerator unit is analogous to the generator section of an absorption refrigeration system.

Figure 2 shows the vapor pressure characteristics of lithium chloride solution. At a given concentration of salt by weight, the solution is in equilibrium with air at a practically constant relative humidity, regardless of temperature. The air humidity leaving the conditioner unit can be adjusted by adjusting the concentration of the solution. At practical operating concentrations, the conditioner unit can dehumidify air to relative humidities as low as 18%, and can humidify air to relative humidities as high as 90%.

COMPARISON OF DEHUMIDIFICATION METHODS

Figure 3 illustrates the three common methods of dehumidifying air for process and comfort air-conditioning applications. In the refrigeration approach, the air is dehumidified by condensing the water on the surface of a cooling coil. The dew point humidity of the air leaving the coil is roughly equal to its dry-bulb temperature, so the coolant must be several degrees colder than the dew point humidity of the air. Substantial reheating of the air may be required to maintain temperature control in the conditioned space. In comfort conditioning applications, if the air cooling is limited to that required to maintain space temperature control, the dew point humidity of the delivered air will increase and high building humidities and discomfort may result.

In the dry desiccant approach, the air is roughly adiabatically dehumidified. The hot, dry air leaving the dehumidifier is subsequently cooled to maintain space temperature control.

In the liquid desiccant approach, the air is simultaneously cooled and dehumidified to the desired temperature and humidity. The air can be dehumidified to a dew point humidity about 40°F below the coolant temperature, for example, 45°F dew point with 85°F cooling tower water. In an integrated comfort system, the liquid desiccant dehumidifier can thus provide building humidity control independent of building temperature control. In addition, the building latent loads can be rejected to cooling tower water, reducing the refrigeration cooling load.

Figure 4 illustrates the desiccant regeneration processes. In the dry desiccant approach, the scavenger air is heated and subsequently heats and dries the desiccant. In the liquid desiccant approach, the desiccant solution is directly heated, and in turn is regenerated by heating and humidifying the scavenger air. The temperature level required for the regeneration process is basically a function of the relative humidity of the air leaving the dehumidifier. The high air temperatures leaving the dry desiccant dehumidifier result in a lower relative humidity than the same dewpoint delivered from the liquid desiccant dehumidifier; hence, substantially higher heat source temperatures are required to drive the dry desiccant regeneration process.

APPLICATION OF LIQUID DESICCANT DEHUMIDIFICATION TO COMFORT CONDITIONING SYSTEM

Figure 5 schematically represents a typical refrigeration air conditioning system. In operation, building return air and ventilation air are mixed. The mixed air is cooled and dehumidified by condensation by one or more cooling coils. All the heat extracted from the air is rejected to the refrigeration system. This process is psychrometrically depicted in Figure 8. It should be noted that the portion of the cooling load attributed to the ventilation air is typically 25 to 40% of the total cooling load at summer design conditions. As shown in Figure 8, if the ventilation air is pre-conditioned in a liquid desiccant dehumidifier, the ventilation air cooling load can be removed from the refrigeration system, and the dehumidifier can deliver the air dry enough to absorb the building latent load.

The removal of the ventilation air load from the refrigeration system not only reduces the size and design power requirements of the refrigeration plant, but also minimizes the change in refrigeration load with changing weather conditions. This can significantly reduce the impact of high demand charges by allowing the smaller refrigeration plant to operate closer to full load, since it handles only building sensible loads, which are relatively insensitive to weather conditions.

The reduction in refrigeration equipment size and operating cost via use of the liquid desiccant dehumidifier is at the expense of heat energy required to operate the regenerator. State of the art regenerators have thermal COP's in the range of 0.4 to 0.5 (BTU Latent Work per BTU Heat Input). At this efficiency level, the use of liquid desiccant dehumidification for comfort conditioning is economically attractive only in areas with very high combined electric demand and usage charges. Reductions in prime (fossil-fueled) heat input would make the use of liquid desiccant dehumidification for comfort conditioning economically attractive in a much broader range of applications and rate structures. The relatively low operating temperatures of the liquid desiccant regeneration process make it possible to effectively use waste heat from the refrigeration cycle to supply a portion of the heat required for regeneration. GRI-sponsored research currently planned will investigate two such integrated liquid desiccant-refrigeration systems.

The first, schematically shown in Figure 6, will utilize condenser heat from the vapor compression refrigeration system to provide a portion of the regeneration heat by preheating the regenerator inlet air and/or preheating the desiccant solution. The remaining heat energy required will be supplied by a gas-fired solution heater. Targeted thermal efficiency for the regeneration process is 0.9 to 0.95 BTU of cooling per BTU of gas input, on a seasonal basis.

The second integrated system, schematically shown in Figure 7, will utilize condenser heat from a double-effect gas-fired absorption chiller as the sole source of heat for regeneration. This system basically adds an open-cycle third effect to the closed-cycle double effect chiller.

Using state-of-the-art technology, liquid desiccant dehumidification has been successfully applied in a number of comfort-conditioning systems, typified by the following examples.

Figure 9 schematically represents the HVAC system installed at the Social Security Administration Metro West Complex in Baltimore, MD. In this system serving approximately 790,000 FT^2 of office space, 0.1 CFM/FT^2 of ventilation air is dehumidified to 31 GR/# moisture content in a two-stage liquid desiccant conditioner. The dehumidified air is distributed to fan coil terminal units utilizing air cells in the steel deck flooring. Each fan coil recirculates the air locally and provides only sensible cooling. Since the building latent load is handled by the dehumidified primary air, the building chiller water temperature was raised to 55°F with no loss in occupant comfort. Use of the liquid desiccant dehumidification system reduced the chiller size by 600 tons and improved its COP by about 8%. In addition, the low ventilation air volume and the use of the decking for ventilation air distribution (made possible by dehumidification of the ventilation air) reduced the floor-ceiling sandwich by 30%, substantially reducing building costs.

Figure 10 schematically shows an integrated system installed at Monroe County Courthouse in Stroudsburg, PA. In this 75,000 FT^2 facility, 0.2 CFM/FT^2 of ventilation air is dehumidified to 45 GR/# by a liquid desiccant conditioner using cooling tower water as a coolant. The dehumidified ventilation air is distributed to unitary terminal heat pumps which perform sensible cooling only. Regeneration heat is supplied by evacuated-tube solar collectors supplemented by jacket heat from an engine-generator. Electricity from the engine-generator is fed into the HVAC power distribution system.

FIGURE 1

KATHABAR SYSTEMS DEHUMIDIFIER

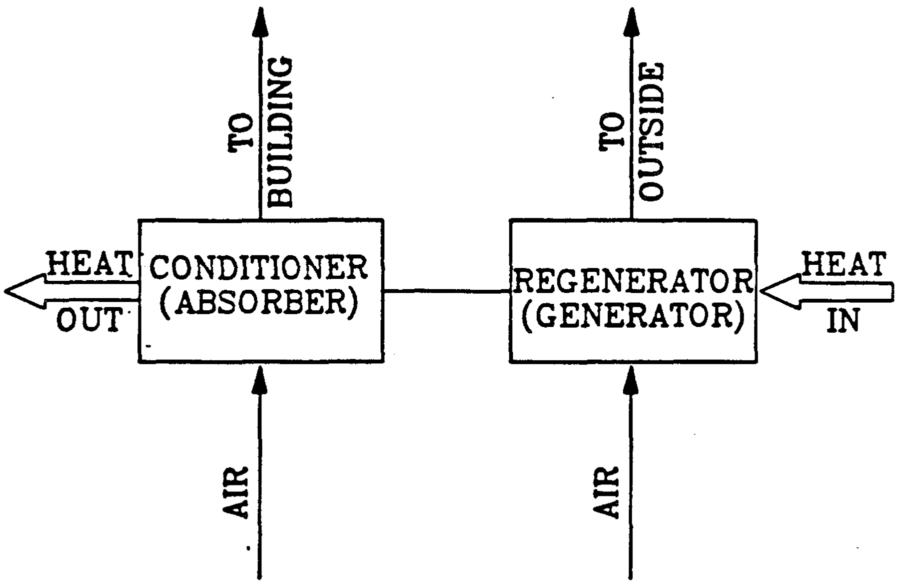


FIGURE 2

VAPOR PRESSURE PROPERTIES OF KATHENE SOLUTION

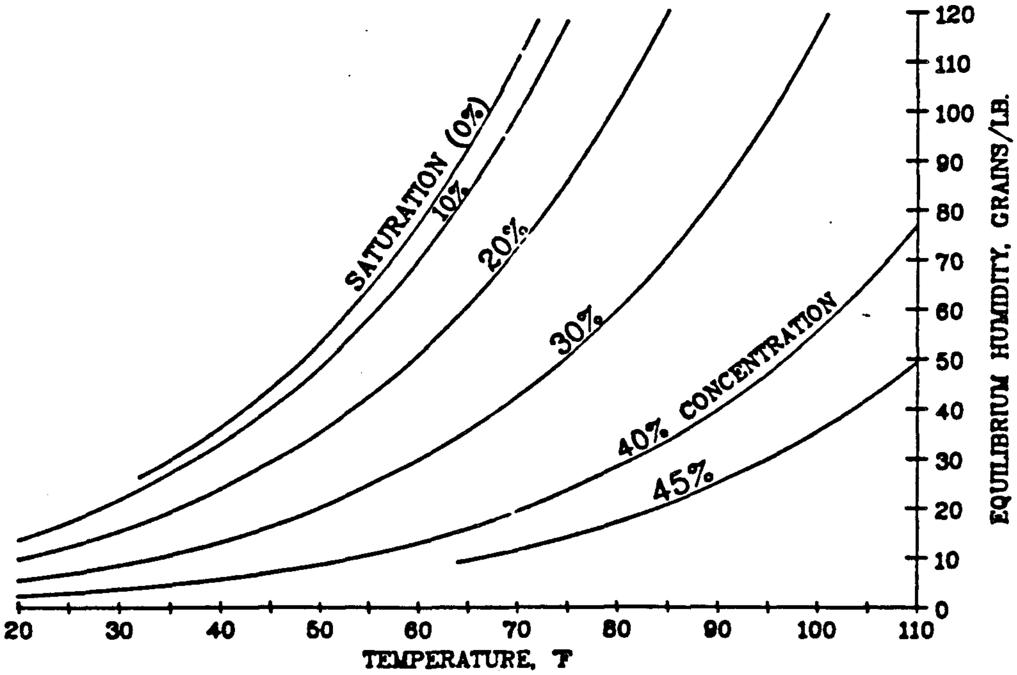


FIGURE 3

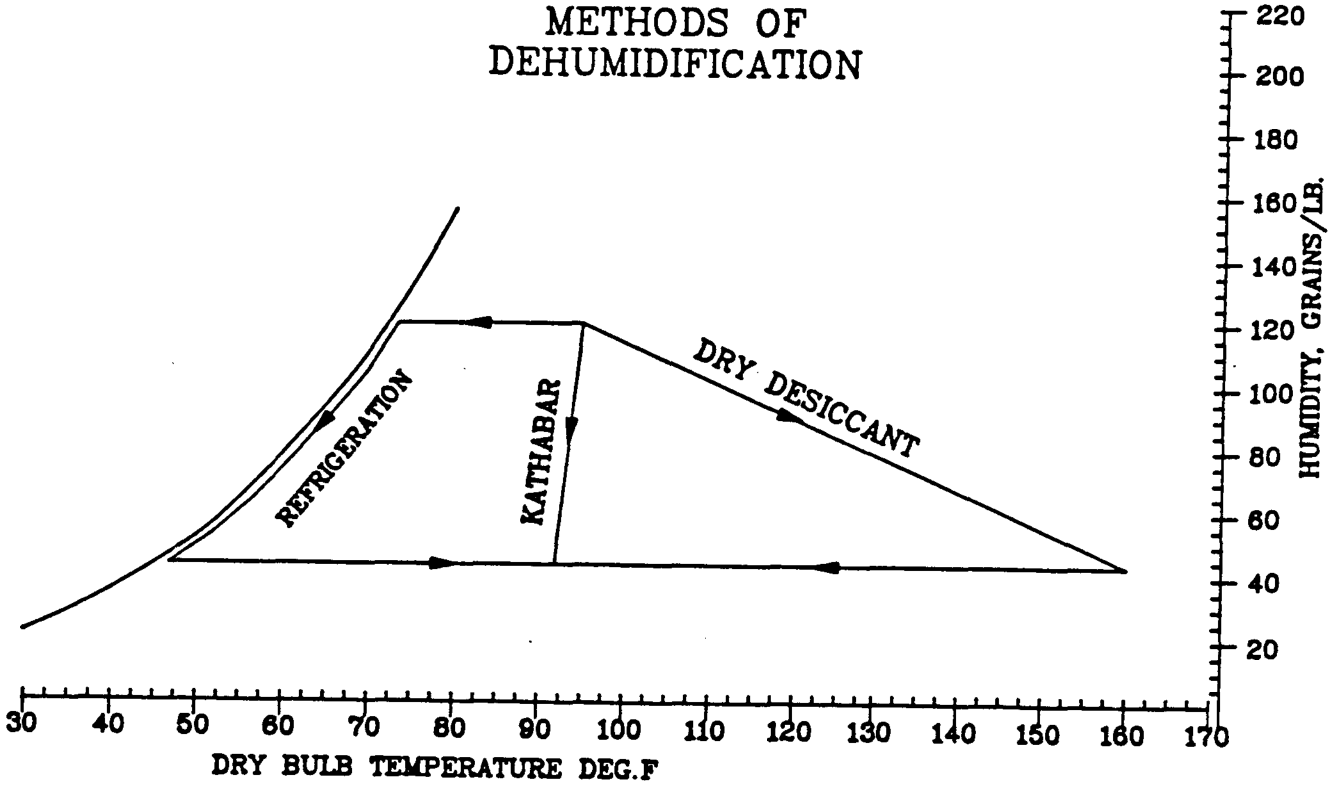


FIGURE 4

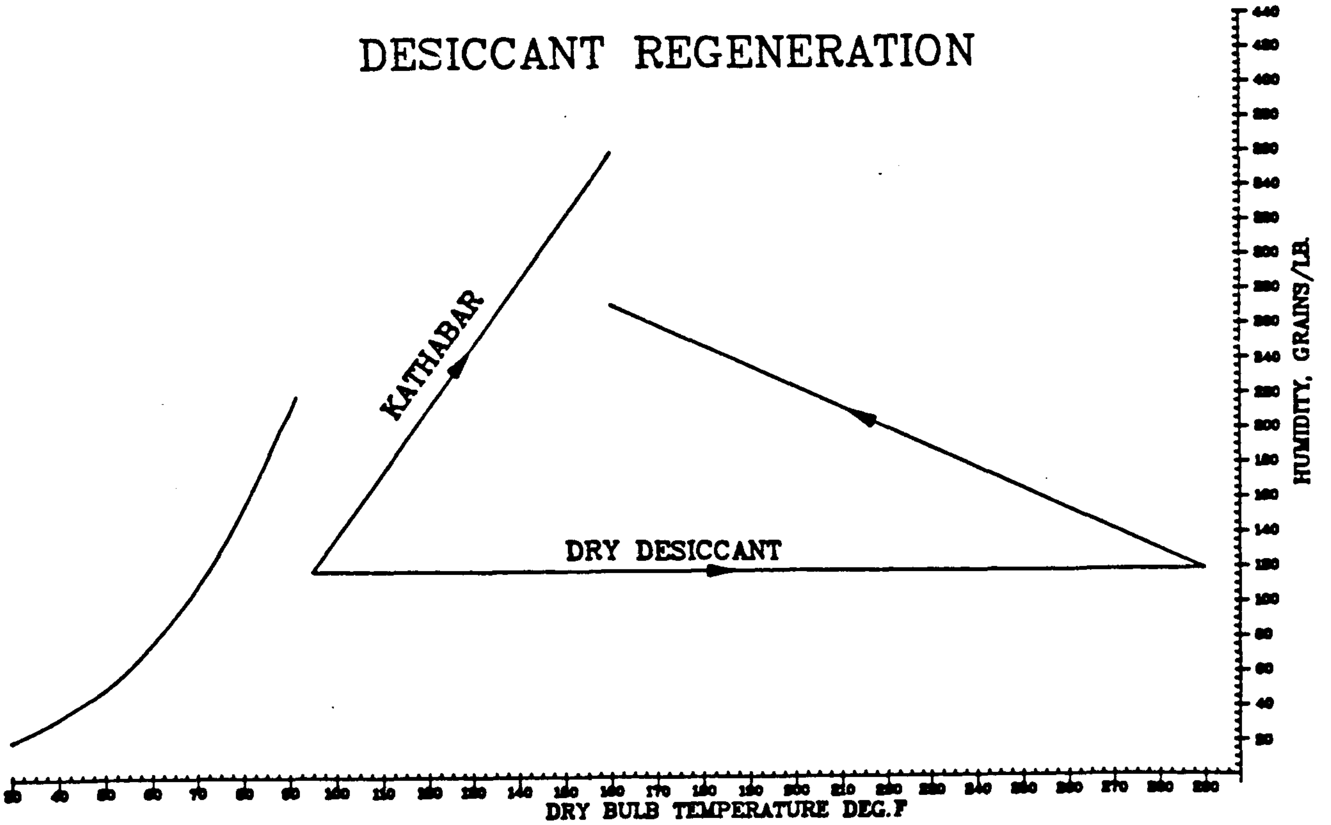


FIGURE 5

TYPICAL
VCR HVAC SYSTEM

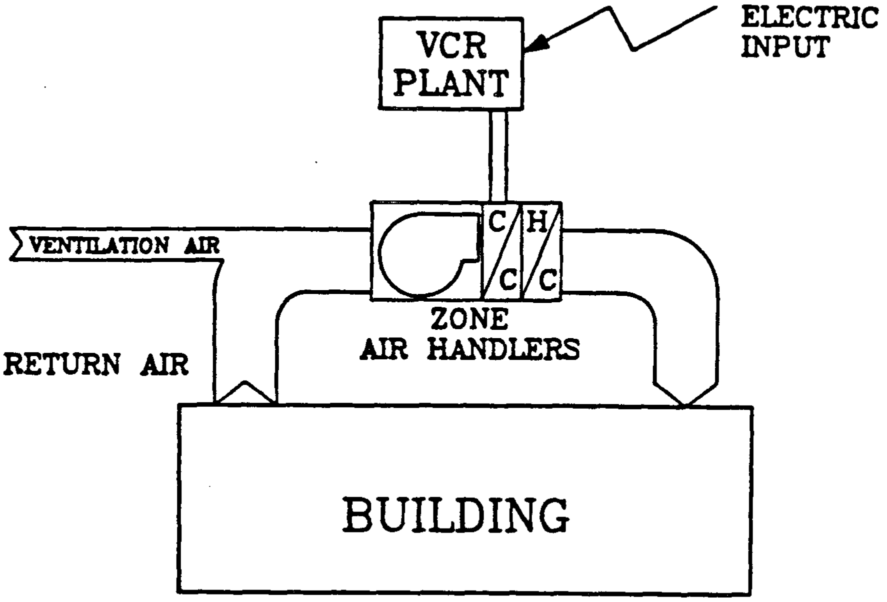
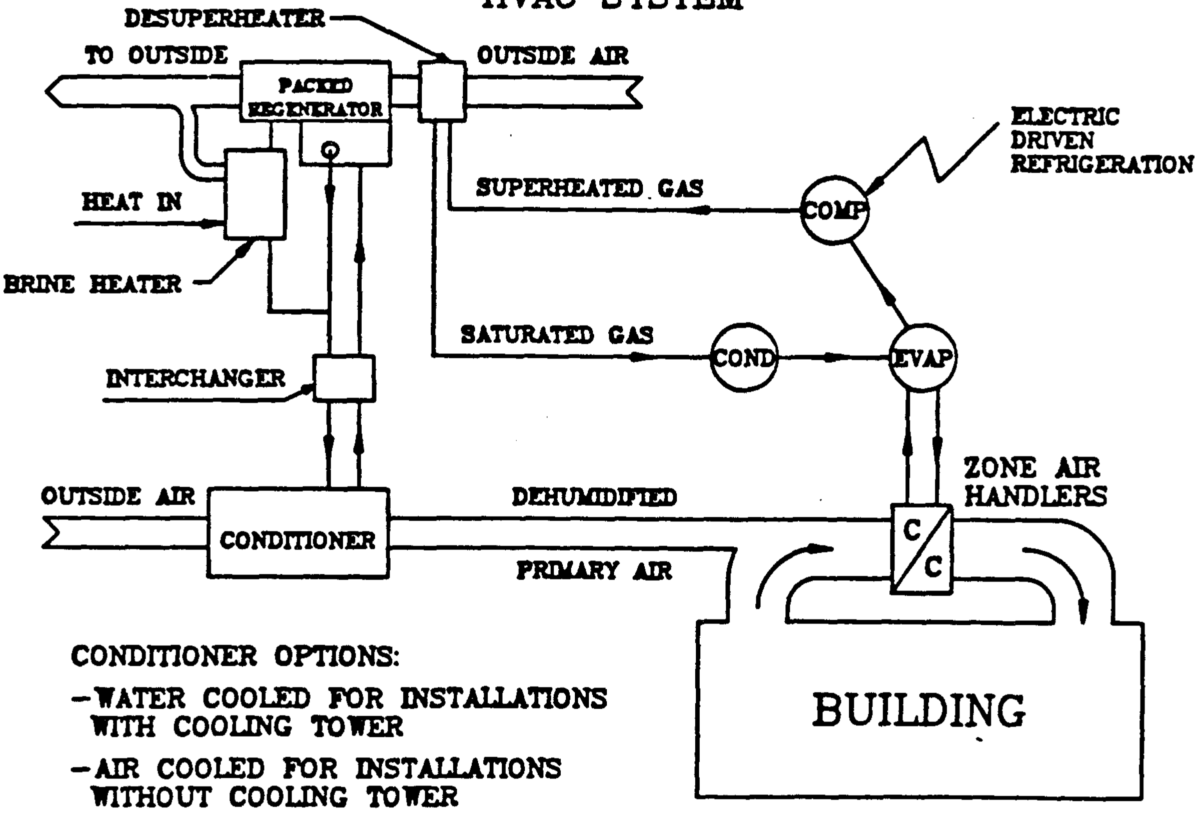


FIGURE 6

INTEGRATED
KATHABAR/VCR
HVAC SYSTEM



CONDITIONER OPTIONS:

-WATER COOLED FOR INSTALLATIONS
WITH COOLING TOWER

-AIR COOLED FOR INSTALLATIONS
WITHOUT COOLING TOWER

FIGURE 7

INTEGRATED
KATHABAR/ABSORPTION REFRIGERATION
HVAC SYSTEM.

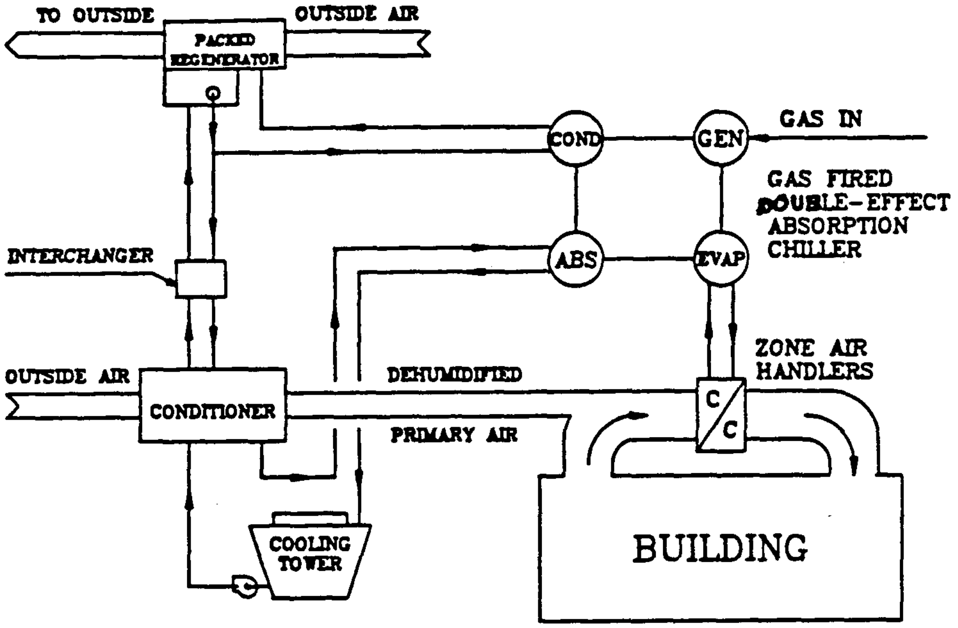


FIGURE 8

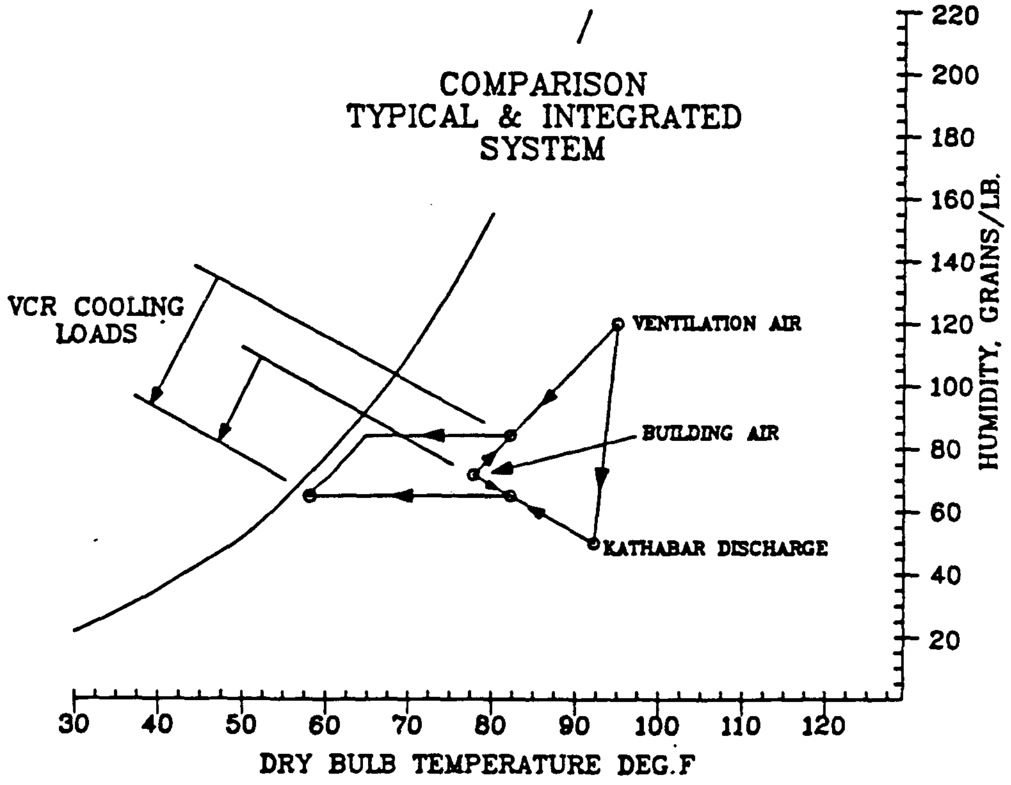
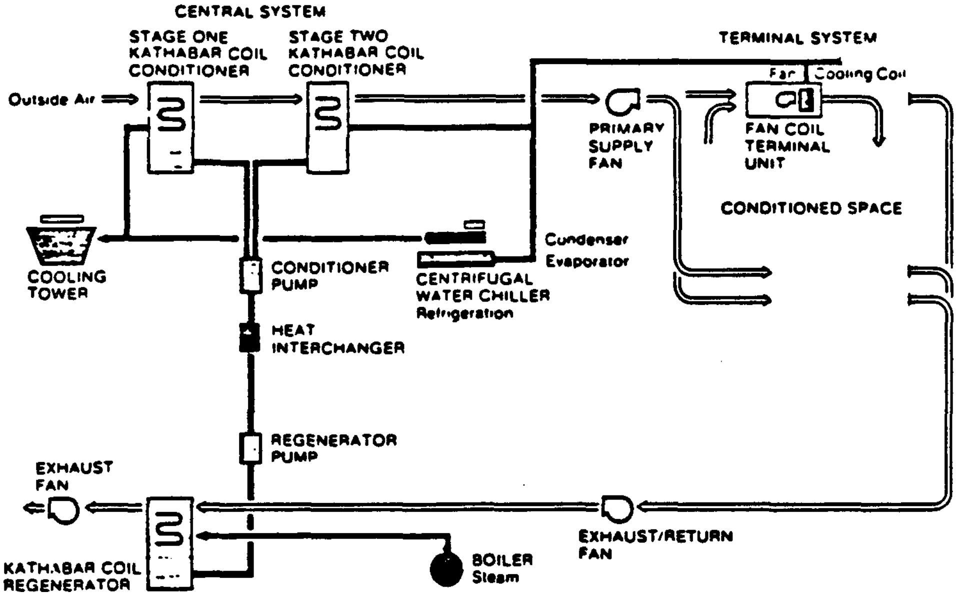

COMPARISON
TYPICAL & INTEGRATED
SYSTEM

FIGURE 9

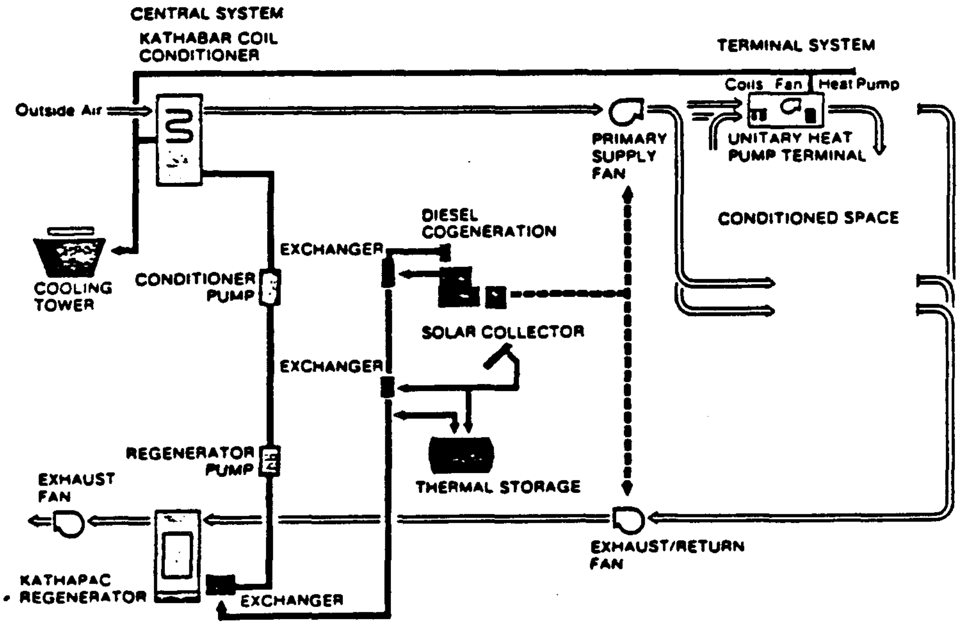
FEDERAL OFFICE BUILDING



TWO STAGE KATHABAR® COIL CONDITIONER,
DIRECT STEAM REGENERATION,
REFRIGERATION AND FAN COIL TERMINALS.

FIGURE 10

COUNTY GOVERNMENT BUILDING

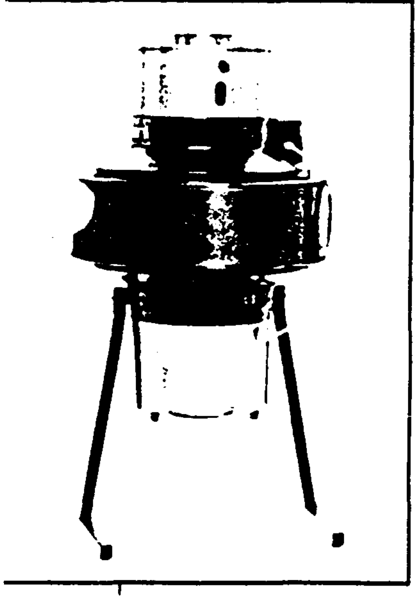


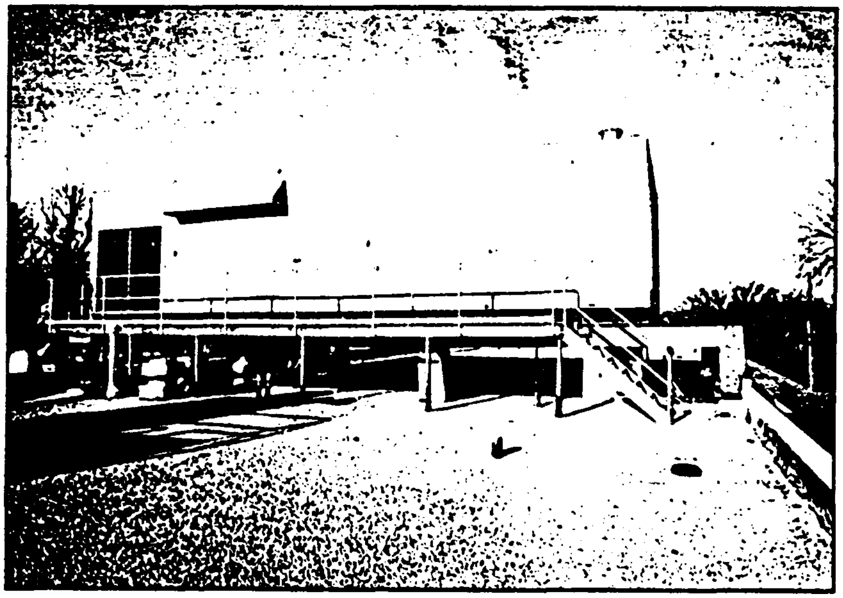
ONE STAGE KATHABAR®,
SOLAR REGENERATION, DIESEL COGENERATION,
AND UNITARY PUMP TERMINALS.

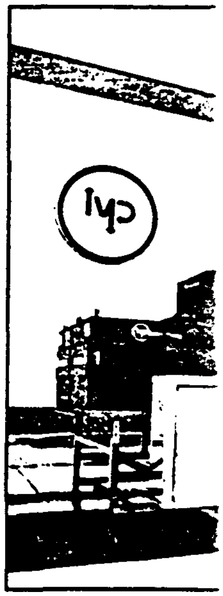
Dehumidification Systems

BY MILLER-PICKING CORPORATION

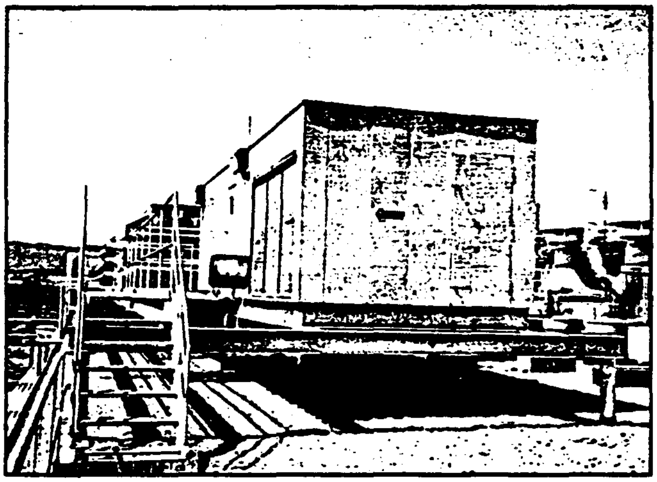
BRIAN BLACKBURN - MANAGER OF ENGINEERING







PACKAGED UNITIZED CONDITIONING SYSTEMS



Many applications requiring dehumidification involve a requirement not only for control of moisture but also temperature in an enclosed space or process. Miller-Picking dehumidifiers can be integrated into a complete unitized conditioning system for total environmental control. The experience of twenty-five years in building custom, industrial air conditioning systems is reflected in the marriage of the Miller-Picking dehumidifiers into total operating systems. These systems reflect the same quality, design flexibility, accessibility, attention to detail, and maintainability that have been a hallmark of Miller-Picking industrial systems in continuous service after years of operation. All components are selected to match and compliment the dehumidifiers which are incorporated in the system to produce the performance and efficiency required. Whether operating on a recirculating air system or a completely once through system, these units are designed with the highest quality components and engineering expertise to provide the lowest energy usage on a continuous basis. All the functions of filtration, air moving, cooling, heating, dehumidifying, humidifying, heat recovery, and controls are completely integrated into a total design package. This total package is designed with a rigid steel base with insulated weather tight enclosures, and provision for simplified job site handling and connection to services. There is no application too big or too small for a Miller-Picking dehumidification system.

Miller-Picking can design the specific dehumidification system for your application. Let us discuss your requirements, loads, duty cycles, etc., in order that we may have an understanding of your need. We will then apply our unique capacity sizing computer program to design the specific dehumidifier for your application. We will give you a specification report outlining the performance details of the dehumidifier selected for you.

SOME TYPICAL APPLICATIONS

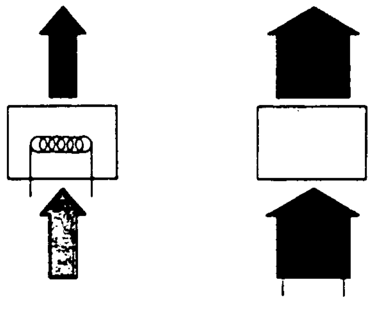
- Industrial processing of hygroscopic materials.
- Storage of hygroscopic materials to prevent unwanted microbial activity.
- Drying processes in food and pharmaceutical industries.
- Laminated glass manufacturing.
- Archieval storage.
- Corrosion protection of steel and other metals.
- Conditioning of resin for injection molding.
- Mothballing of equipment for rapid start-up.
- Protection of out-of-service boilers and turbines.
- Cargo protection.
- Cold storage.
- Corrosion protection in pumping stations.
- Protection of sand blasting operations.
- Drying of water damaged property.

THE COST EFFECTIVE ANSWER FOR ATMOSPHERIC DEHUMIDIFICATION

Three reasons you SAVE ENERGY DOLLARS with a Miller-Picking Dehumidifier

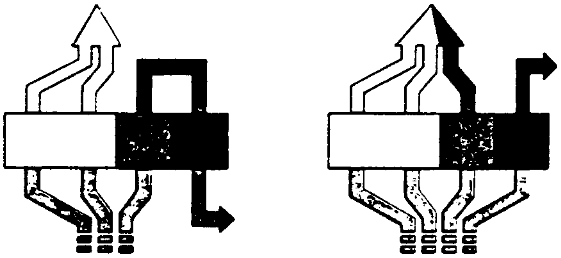
LOWER REGENERATION ENERGY LOSSES

The Miller-Picking dehumidifier applies the regeneration energy directly to the desiccant. The air is used only to transport water. This results in a lower regeneration airflow and entering air temperature than competitive systems using an air stream to transfer all the regeneration energy. This becomes even more important at lower ambient temperatures. Much less energy goes out the exhaust with the Miller-Picking dehumidifier.



ENERGY RECOVERY IS STANDARD

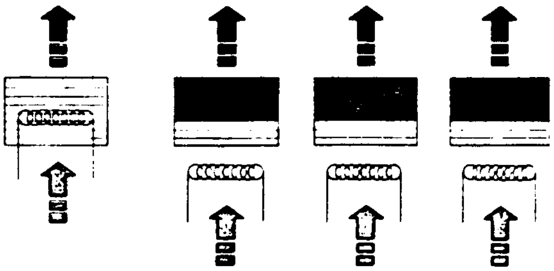
After regeneration of the Miller-Picking dehumidifier the hot desiccant bed is cooled and the energy totally transferred to the next bed which is being regenerated. Unlike Miller-Picking, continuously rotating systems move the hot bed into the process stream with the result that the energy is lost from the regenerated loop and added to the process stream where additional cooling will be required.



PULSED REGENERATION

The Miller-Picking dehumidifier always regenerates a bed which is close to being saturated with water. This means you heat the bed only once to remove a full measure of water.

Competitive continuously rotating systems under varying loads regenerate a partially saturated bed. What results is energy required to heat the unused portion of the bed several times to get one full measure of water.



Gas-Fueled Desiccant-Based Air-Conditioning

Barry M. Cohen
Tecogen, Inc.

Introduction

Tecogen, Inc., a subsidiary of Thermo Electron, is currently under contract to the Gas Research Institute to develop a gas-fueled air-conditioning system based upon the principle of desiccant dehumidification. The system operates in a well-known cycle originally devised at AB Carl Munters of Sweden and often referred to as the MEC (Munters Environmental Control) cycle. The development program involved basic studies of materials and concepts, system optimization, feasibility assessment and cost analysis. At the present time a product has been defined which can satisfy the general air-conditioning market, but is especially attractive when one can take advantage of its unique features.

The cycle involves four basic components: a desiccant dehumidifier consisting of a rotating desiccant wheel and a reactivation air heater, a ventilation heat recovery heat exchanger, a humidifier component processing two air streams, and a blower component to drive two air streams. In the common ventilation mode the two air streams are: one which is ambient air (process air) which is cooled and dehumidified and introduced to the space at an enthalpy below that of the room; and a second which is room air (regeneration air) passed in counterflow to the ambient air and rejected to the external surroundings. The room air is humidified to lower its temperature, heated by the incoming air stream to both preheat it for regeneration and cool the air to be introduced to the space, further heated by a gas heater to bring it to a regeneration temperature, and passed through the desiccant wheel to dry the desiccant. The ambient air is passed through the desiccant wheel to be adiabatically dried, cooled by the outgoing air to reject energy to the ambient, and humidified to lower its dry bulb temperature to create sensible cooling.

The system has a number of attributes which can, in the right applications, make it very attractive when compared to electric vapor compression systems. It uses natural gas as its prime energy source which is an advantage when electric rates are high. It affords a high degree of ventilation which can be beneficial if internal air quality is poor. It can provide a high degree of dehumidification, which is useful if internal latent loads are high. It can operate in energy conservation modes such as regenerative evaporative cooling or heat recovery heat exchange which can at times avoid the need to consume energy to provide space-conditioning, especially in dry climates like the Southwest United States.

Performance

The performance of the unit is measured by two key parameters, the fuel-based coefficient of performance (COP_f) and the electric-based energy efficiency ratio (EER), where:

$$COP_f = \frac{\text{Cooling Capacity}}{\text{Gas Consumption Rate x Heating Value}}$$

$$EER = \frac{\text{Cooling Capacity}}{\text{Electric Power Draw x } 3412 \, \frac{\text{Btu}}{\text{kWh}}}$$

The objective in designing such a system as a "gas-fueled" device is to exceed a reasonably high value of EER, say 35, and to achieve an acceptable COP_f which results in economical operation. A COP_f of about 1.0 is usually found to be acceptable if the first cost is reasonable. In fact, lower values of COP_f can be tolerated if the first cost is low.

We have built a prototype unit which has been tested in our laboratory. The results of prototype testing and correlation with computer analysis have led to the specification of a system which meets the required performance thresholds. It was necessary to utilize the test data to refine the computer simulation.

In the lab, testing produced cooling capacity results ranging typically from 320 cfm/ton to 440 cfm/ton, while the desired level was about 350 cfm/ton. All data is at ARI test conditions of 80°F/67°F indoors and 95°F/75°F outdoors (T_{db}/T_{wb}). At the same time thermal COP's (not adjusted for gas heater inefficiencies) ranged from .80 to .95. The heat exchanger effectiveness was confirmed from measurements to be about 93% as expected.

Review of the data led to some modifications to the computer simulation and some redesign of the system. The redesign came principally in the form of a revised desiccant core design. As a result, specifications for the current unit, if sized for 3 tons of cooling capacity inside the conditioned space are at ARI conditions:

Air Flow = 350 cfm/ton
E_{hx} = 93%
COP_f = 1.0
EER = 45
Package size = 4' x 4' x 8'

This design reflects the first in a planned series of development steps which, by use of more sophisticated design options, are expected to lead to progressively better performance. However, the performance as quoted is considered to be well in excess of that required for early niche market penetration and sufficient for a significant penetration into the commercial air-conditioning market in general in many areas of the U.S.

Market Perspective and System Cost Goals

It is important to understand the complex air-conditioning market in order to best determine the ideal path to commercialization. Certain applications, i.e., those which take full advantage of the unique characteristics of the system, are best for early market penetration. Later, as the public is more educated to the features of the system and as the manufacturer is able to progressively reduce the selling price when development and setup costs are repaid and when manufacturing methods improve, the market will broaden considerably.

We have done seasonal performance simulations for various applications in various cities in the U.S. One should remember also that since local weather was coupled with local utility rates in the evaluations, many other combinations are possible. A favorable weather pattern would result in the most cost-competitive operation where utility rates are also favorable.

We typically looked at two types of applications in cities representing extreme climates, but in relatively southerly areas. The longer cooling season in southerly climates results in larger annual savings and thus faster payback of any cost premiums. Moreover, while the seasonal COP's and EER's are generally considerably larger than the test point values, dry southerly climates like in Arizona enhance the seasonal values significantly, especially if intelligent control is applied.

The extremes of application types were an office which represents a highly sensible load for a relatively limited number of hours per week and a health club which represents a more latent load for a greater number of hours per week. The extremes of climates were Miami which represents a hot and very humid cooling season and Phoenix which represents a hotter, but very dry cooling season. Buildings were sized to require between 5 and 15 tons of cooling.

Seasonal COP_f's ranged from as low as 1.0 for very humid ambients to as high as 3.0 for dry ambients, while seasonal EER's were typically in the range of 60 to 70 with no consistent pattern. As a result of these seasonal performances, the allowable delivered costs for a 3-year payback ranged to as high as $2100/ton in the favorable cases, but down to the point where no premium above electric air-conditioning could be tolerated in the worst cases.

Currently, we are in the process of design evolution. We have already been able to project costs below the $1400/ton level without having to sacrifice attractive markups in order to keep the price low. However, many of the methods and materials we are now considering should allow us to realize even lower prices.

Conclusion

It has been shown that a gas-fueled desiccant based air-conditioning system is an attractive product. Early marketing efforts should concentrate on users who can take advantage of the unique features, but ultimately a significant market can be expected.

UNIVERSITY OF NEW SOUTH WALES

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

Brief Notes and Sample Cost Calculations
on
Gas Engine Dessicant Hybrid Cooling
for
Australian Conditions

Dr. I. MacLaine-Cross
School of Mechanical and Industrial Engineering
University of New South Wales
P.O. Box 1
Kensington: N.S.W. 2033
Australia

The essential modifications are a gas carburettor and automatic oil feed. Desireable modifications are larger air cleaners, higher compression ratio pistons, an auxiliary oil pump and tuning of the pollution control system. Annual maintenance should be sufficient. Automatic monitoring of oil pressures and jacket inlet and outlet temperature will prevent engine damage. Automatic logging of oil and gas comsumption will give the engine condition. Design of the package for fast engine change over will reduce down-time. Capital cost will be higher and reliability will be lower than an electric motor initially, but there is no reason why experience and quantity production should not reduce these differences substantially.

4. THE DESICCANT DEHUMIDIFIER PACKAGE.

The desiccant dehumidifier package handles all the fresh air introduced at the plant room and its size and cost is dependent on the fresh air volume. It consists of a heating coil, an adiabatic solid desiccant dehumidifier and heat exchanger (Figure 1). The dehumidifier takes the ambient air (1 in Figure 2) down a path just above an adiabatic saturation line to a relative humidity (2) close to that of the regeneration air (6). The heat exchanger then cools it while heating the regeneration air from (10) to (5). Gas engine waste heat is added from (5) to (6) to allow the dehumidifier to be regenerated from (6) to (7).

Compact low cost, high performance, heat exchangers have been developed in Australia [4] and the principles used in a desiccant dehumidifier tested in the U.S.A. [3]. The life of the desiccant dehumidifier is a topic of current research. A desiccant dehumidifier package based on this technology is being considered for a demonstration project planned by PRI Energy Systems of Hawaii.

5. ECONOMICS

Data for and Sample cost calculations are attached (Appendix A and B). The payback time has been calculated for 10 units sold per year and 100 units sold per year. The data have been chosen to be representative of the medium to large commercial market. For Brisbane and Sydney, substantial penetration of the potential cooling market is clearly possible.

For Melbourne, improved performance over that assumed for gas engine desiccant hybrids is necessary, but GEDH is still very attractive for high heating loads, latent loads, fresh air or operating hours. Cooling market penetration may be close to Brisbane and Sydney when these are considered.

6. REFERENCES.

1. T.R. PENNEY and I.L. MACLAINE-CROSS.
"Promising advances in desiccant cooling",
Solar Buildings Conference, Washington DC, March 1985,
SERI/TP-252-2683.

2. A.A. PESARAN, I.L. MACLAINE-CROSS and E. VAN DEN BULCK.
"Measurements on Promising Dehumidifier Materials and Geometries",
Solar Energy Research Institute, Golden, Co.,
SERI/TR-252-2898, February 1986.

3. D. BHARATHAN, J.M. PARSONS and I. MACLAINE-CROSS,
"Experimental Studies of Heat and Mass Exchange in Parallel-Passage Rotary Desiccant Dehumidifiers for Solar Cooling Applications",
Solar Energy Research Institute, Golden, Co.,
SERI/TR-252-2897, February 1986.

4. American Society of Heating Refrigerating and Air Conditioning Engineers Inc., "ASHRAE Guide and Data Book, Systems volume, Chapter 33, Engine and Turbine Drives", 1983.

5. C.W. AMBROSE, I.L. MACLAINE-CROSS and E.B. ROBSON,
"The Use of Rotary Regenerative Heat Exchangers for the Conservation of Energy in Buildings", Final Report Project 109, National Energy Research Development and Demonstration Council, Australian Government Canberra, 1983.

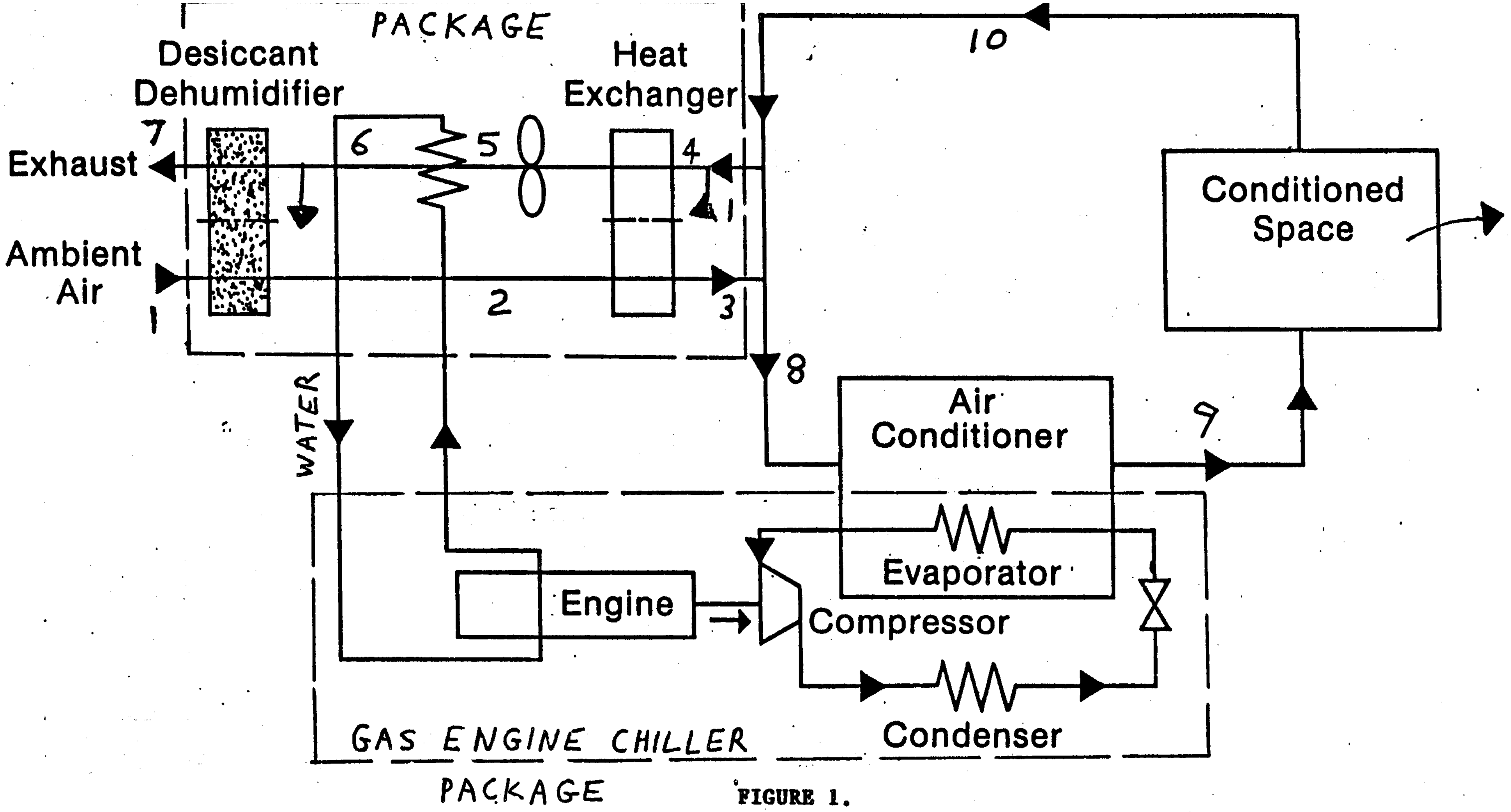
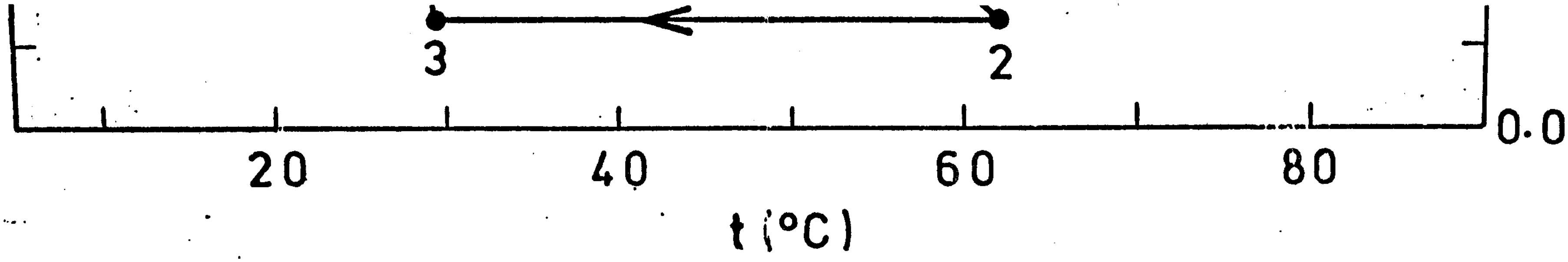


FIGURE 1.

Schematic of Gas Engine Desiccant Hybrid Cooling System. Numbers correspond to typical air states in Figure 2.



cant hybrid
re 1.

APPENDIX A

DATA FOR SAMPLE COST CALCULATIONS

GAS ENGINE DESICCANT HYBRID COOLING SYSTEM

MEDIUM TO LARGE COMMERCIAL INSTALLATION

1. OCCUPATION

3,000 hours per year.

2. OUTDOOR DESIGN AND LOAD CONDITIONS

Location (Airport)	Dry bulb (°C)	Wet bulb (°C)	Full load operation (Hrs)
Brisbane	31.0	25.0	2000
Melbourne	34.5	21.0	1000
Sydney	30.5	23.0	1500

Space load 20% latent

Return air 26.7°C DB, Supply air 10°C DB, 10°C WB

Minimum fresh air 25%.

3. ENERGY PRICES (1st March, 1986)

Location	Gas	Electricity	
	Energy (c/MJ)	Demand($/kW month)	Energy(c/kW hr)
Brisbane	0.801	16.34	3.20
Melbourne	0.800	13.79	2.68
Sydney	0.6481	15.27	3.78

4. VAPOUR COMPRESSION PLANT

Capital cost = 200$/kW Cooling COP = 3.5

5. GAS ENGINE DRIVEN COMPRESSOR PACKAGE.

Engine capital cost = 400$/kW shaft
Engine efficiency = 0.30
Compressor COP = 4.0
Heat recovery efficiency = 0.80

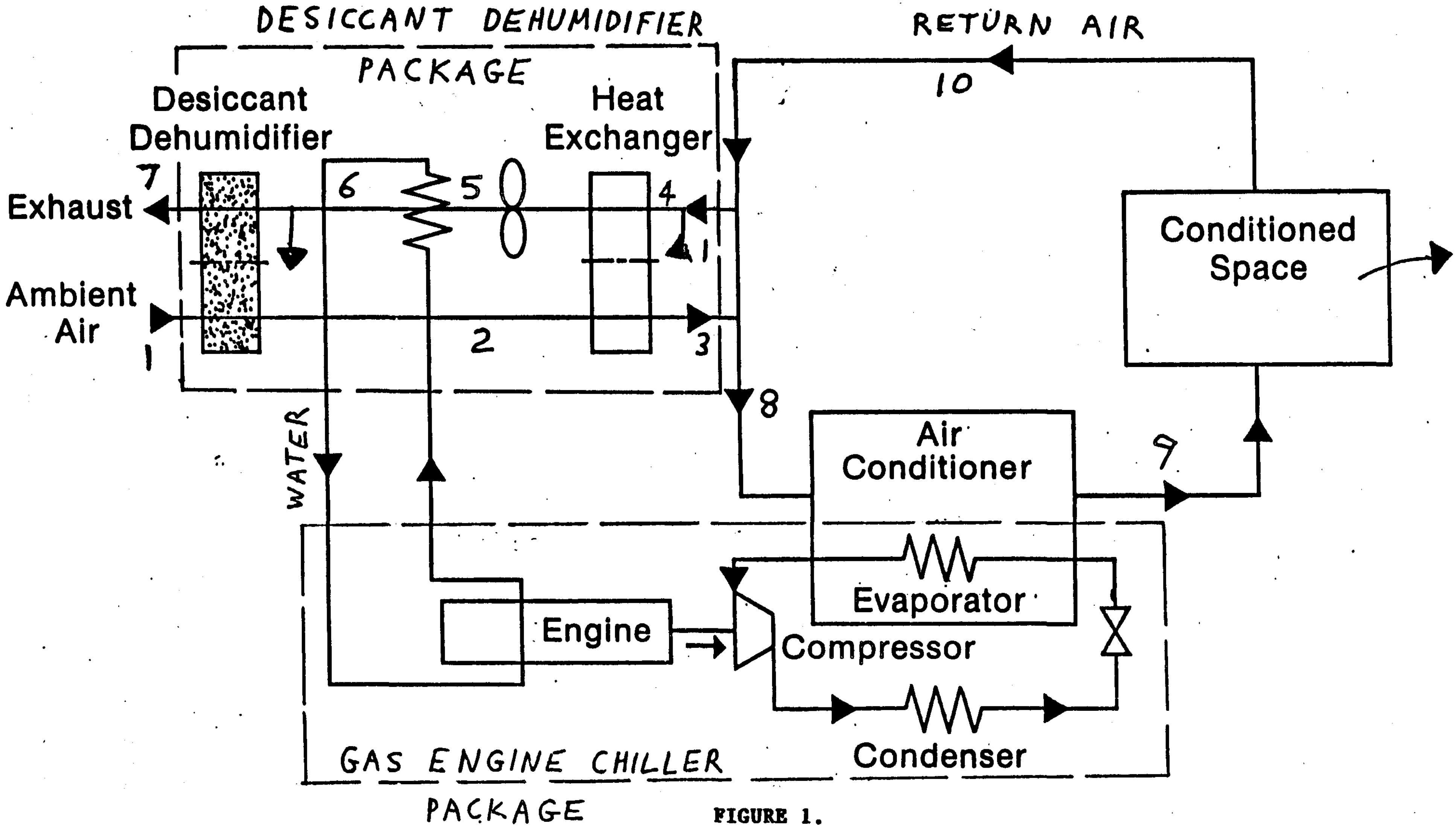


FIGURE 1.

Schematic of Gas Engine Desiccant Hybrid Cooling System. Numbers correspond to typical air states in Figure 2.

6. DESICCANT DEHUMIDIFIER PACKAGE

Installed capital cost = 8$ s/L fresh air
Thermal COP space latent load = 1.0
Thermal COP fresh air load = ∞
Parasitic electric consumption = 1 J/L fresh air

7. ADVANCED COMPONENTS.

The above engine and dehumidifier costs assume 10 units sold per year. Research into package design and manufacture to reduce material costs and promotional activity to increase sales to 100 units sold per year could halve capital costs.

Advanced engine capital cost = 200$/kW shaft
Advanced dehumidifier capital cost = 4$ s/L fresh air

APPENDIX B

SAMPLE COST CALCULATIONS - PAYBACK TIME

GAS ENGINE DESICCANT HYBRID COOLING SYSTEM

MEDIUM TO LARGE COMMERCIAL INSTALLATION

1. PLANT DESIGN LOADS

Supply specific enthalpy h_s = 29.3 J/g dry air
Return specific enthalpy h_R = 51.4 J/g dry air

Hybrid compressor cooling load = 0.8x(51.4-29.3)=17.68 J/g dry air
engine power = 17.68/4.0 = 4.42 J/g dry air
gas consumption = 4.42/0.3 = 14.73 J/g dry air
engine waste heat = 0.8x(10.3)x14.73 = 8.251 J/g dry air.

Space latent load = 0.2x(51.429.3) = 4.42 J/g dry air.

	BRISBANE	MELBOURNE	SYDNEY
Ambient specific enthalpy (J/g dry air)	76.3	61.0	68.4
Conventional plant load (J/g dry air)	28.33	24.50	26.35
Hybrid Thermal Coefficient of Performance	1.92	1.66	1.79

2. PLANT CAPITAL COSTS.

Hybrid compressor = 200 x 17.68 = 3536$s/kg dry air
Gas engine = 400 x 4.42 = 1768$s/kg dry air

Desiccant dehumidifier package = 8x10^3 x 0.25/1.18 = 1695$s/kg dry air.

Total hybrid plant cost = 6999$s/kg dry air
Total advanced hybrid plant cost. = 5268$s/kg dry air

	BRISBANE	MELBOURNE	SYDNEY
Conventional Plant ($s/kg dry air)	5666	4900	5270
Incremental cost hybrid ($s/kg dry air)	1333	2099	1729
($/kW load)	47	86	66
Incremental cost advanced hybrid			
($s/kg dry air)	-398	368	-2.5
($/kW load)	- 14	15	0

3. ANNUAL ENERGY COSTS

	BRISBANE	MELBOURNE	SYDNEY
Conventional Electric Demand ($s/year kg dry air)	1058	386.1	689.8
Conventional Electric Energy ($s/year kg dry air)	518.0	187.6	426.9

	BRISBANE	MELBOURNE	SYDNEY
Hybrid gas consumption (\$s/year kg dry air)	849.7	424.3	515.6
Hybrid electric demand (\$s/year kg dry air)	41.5	35.1	38.8
Hybrid electric energy (\$s/year kg dry air)	20.3	17.0	24.0
Nett energy saving (\$s/year kg dry air)	664.5	97.3	538.3

4. PAYBACK TIME FOR INCREMENTAL HYBRID CAPITAL COST.

	BRISBANE	MELBOURNE	SYDNEY
Available components (years)	2.0	22	3.2
Advanced components (years)	0	3.8	0

INTEGRATING DESICCANT BASED HVAC SYSTEMS

By: Milton Meckler, P.E., FASHRAE
President/CEO, The Meckler Group
Encino, CA 91316-3871

Abstract:

Thermally regenerated liquid or solid desiccant systems in combination with either electric vapor compression, gas fired absorption or heater-chiller refrigeration equipment with TES, and evaporatively chilled (i.e. non-refrigerated) water systems will be explored for a variety of hybrid desiccant cycles ranging from conventional air distribution supply to integrated three "tier" air handling unit systems incorporating desiccant section and employing fan powered terminals, to reduce overall ductwork, etc. Desiccant system integration allows de-coupling sensible and latent cooling coil control, permitting higher chilled water temperatures and a significantly lower energy use requirement for installed mechanical refrigeration serving as peaking machines only. Emphasis will be placed on the special role of the consulting engineer or design build firm as the HVAC systems integrator essential in maintaining continuity of the concept/design/performance commitment required from early schematics through construction to start-up. Need for advanced software capable of demonstrating hybrid desiccant cycles will also be examined employing an illustrative example. Also, use of a novel liquid desiccant storage and distribution concept will be presented to demonstrate how one can significantly reduce desiccant regenerator size, and improve system performance with a coupled cogeneration system either of conventional design or augmented by cost effective waste heat source driven Rankine cycle heat engine auxiliary.

Introduction

When climatic conditions permit, cooling loads can be shifted from mechanical refrigeration to evaporative chilling and/or desiccant dehumidification HVAC components at far less energy for the same thermal load requirement. The term evaporative chilling denotes both an enhancement to and a narrowing of well known evaporative cooling principles. Evaporative chilling places emphasis on substitution of evaporatively chilled for mechanically refrigerated water and should not be confused with replacing mechanically refrigerated with evaporatively cooled air as with direct or indirect evaporative coolers.

Off-seasonal use of conventional, direct cooling towers has expanded in recent years to include year-round use, employing novel chemical treatment methods combined with high efficiency filtration of the open cooling tower circuit. A major concern exists among most HVAC designers of potential cross-contamination and fouling of the normally closed circuit chilled water piping and coils from direct inter-connection of the open cooling tower loop. Yet, there are great number of buildings where cooling is required at relatively low outdoor air temperature and for extended periods of time.

Year-round utilization of cooling towers can result in considerable energy savings from displacement of mechanical refrigeration capacity (and associated auxiliaries, etc.) and should be considered wherever possible. Evaporative chilling can bring about significant positive cash flow potential even after accounting for additional costs associated with freeze-up protection, plate and frame heat exchanger controls, piping and chemical treatment, etc. Additional operating costs associated with extended operation of the tower fans (including related additional maintenance, etc.) must be considered, however, in any systematic evaluation of overall cooling system performance. Use of reliable co-incident weather data permitting more accurate determination of co-incident outdoor wet and dry bulb temperatures for each of three nominal 8-hour daily segments has provided the catalyst for HVAC designers to explore year-round cooling tower use. When latent heat loads are shifted to evaporatively chilled media mechanical refrigeration, energy demands fall significantly depending upon the proportion of outdoor air in total mixture and space sensible to latent heat ratio. In this way, higher supply water temperatures to air handling (or terminal) cooling coils are possible at efficient sensible load removal rates.

Interfacing Evaporative Chilling and Desiccant Dehumidification

Recommended comfort standard (ASHRAE 55-74), suggests that the comfort zone lies between 73°F (23 C) and 77°F (25 C) dry bulb temperature and within the 20 to 60 percent relatively humidity range. ASHRAE also has established a standard effective temperature scale which allows for direct comparison of comfort conditions, since both temperature and humidity are considered. It is this standard not a fixed maintained wet/dry bulb design condition that should dictate desiccant system design performance and response. Also new, higher ventilation rates brought on by higher occupant density, synthetic materials and higher concentrations of disease bearing microorganisms require a more thorough evaluation of ventilation space pattern than available from, say ADPI considerations. Option # 1 System (DOS #1) one of several proprietary systems now marketed by our affiliate, Design Build Systems) is illustrated in Figure 1. It employs a indirect contact evaporative chiller of type shown in Figure 2 to take advantage of the ambient wet bulb driving force. Evaporatively chilled water from this unit is mixed directly into (closed circuit) chilled water distribution network as shown. The proportion of mechanically chilled to evaporatively water required to maintain space comfort will fluctuate during any hour of any given day. All desiccant components of DOS #1 are available commercially and operate to remove latent load from the chilled water supply as follows: The dehumidifier contactor section (see Figure 3) reduces the dew-point of entering mixed outdoor, relief air and (the balance of) untreated outside air before it also enters the indirect (i.e. evaporative) chiller to maximize contact area driving force.

The desiccant is concentrated in regenerator section as shown and uses waste heat recovered from the "high side" split-condenser attached to the two stage electric heat recovery chiller as shown. Also notice that the desiccant contactor section is supplied with evaporatively chilled water generated in a high efficiency indirect evaporative chiller as its principal coolant. The desiccant contactor section contains extended surface cooling coils and the regenerator section contains heating coils. Air to be dehumidified passes over the contactor (cooling) coils and comes into intimate contact with a desiccant solution such as tri-ethylene glycol, lithium chloride, or calcium chloride. The degree of dehumidification is fixed by temperature, concentration, and the physical characteristics of the hygroscopic solution selected which establishes the vapor pressure differential between the inlet air and hygroscopic liquid flow.

By varying the coolant flow to the contactor, the solution contact temperature is controlled in accordance with moisture loading. This assures a fixed outlet moisture content. Water coolant removes both the sensible and latent heat of condensation of water vapor from the air as well as the residual heat load from the regenerator (and it accompanying heat of solution). The regenerator section normally operates on 10 to 20 percent of the total contactor desiccant stream flow. The desiccant solution is sprayed over the heating coils and heated so water can be absorbed from solution by air resulting from a reverse vapor pressure differential (i.e., outdoor scavenger air). The hot, concentrated desiccant solution flows to a common or seperate (i.e. economizer) sump where it mixes with dilute solution from the contactor section before it returns to the contactor spray header for another cycle, etc.

Heat flow (by whatever means) supplied to the regenerator section heating coil can be controlled by either a level, specific gravity, or boiling point type control that responds to a pre-set concentration level for anticipated (i.e. design moisture) loading. Whereas most conventionally designed liquid desiccant systems utilize steam or heated hot water, DOS #1, utilizes waste heat normally rejected directly from the two stage refrigeration chiller condenser bundle. Other potential (i.e. thermal sources of regeneration energy (i.e. solar heated fluids, exhaust gas engine jacket water, etc.) or other potentially usable high temperature waste heat streams will also be discussed subsequently. Employing chiller condenser bundle waste heat is especially beneficial during cold, dry periods for heating and at reduced space dehumidifying requirements. As outdoor temperatures rise, co-incident wet bulb and humidity levels rise also causing a shift in demand to lower space heating but a higher desiccant requirement and vice versus. This inherent self leveling mechanism maximizes DOS #1 System utilization of year-round (i.e. available) waste heat.

Optimum unit sizes for indicated DOS #1 are normally determined by computer simulation of the proposed desiccant system configuration and micro-climate. Figure 4 illustrates a graphical representation of hourly co-incident weather data for Los Angeles, CA. A representation computer simulation program run developed for comparing desiccant system performance as against conventional HVAC system is illustrated in Figure 5. Although not shown in Figure 1, use of solid desiccant systems such as manufactured by Cargocare and others can also be employed and should also be evaluated on a site specific basis.

Combined Peaking and Desiccant Regeneration

Single-stage electric water chillers have a wide range of performance and efficiency, depending upon capacity and compressor type (illustrated in Table 1); namely: reciprocating, centrifugal, and screw. The screw has the highest available discharge temperature potential for heat recovery followed by reciprocating, and centrifugal.

Useful heat recovery implies simultaneous heating and cooling demands for extended operating periods, yet operating at un-necessarily high condenser discharge temperatures requires a correspondingly higher condenser head pressure (and energy input) than that necessary to discharge the same heat to ambient. The two stage water chiller illustrated in Figure 1 permits simultaneous (yet independent) operation of separate "waste heat" and "heat recovery" condenser sections at properly matched lower and higher temperature (or pressure) levels, respectively. Single-stage chiller operational economics are still maintained for excess waste heat rejected to the cooling tower at roughly 105 to 115°F. Second stage compressor operation is reserved only for the balance of rejected heat needed for space heating or desiccant regeneration at higher system (available) temperatures (i.e., 160°F). Heat recovery by selective temperature discharge permits cascading energy flows while recognizing that the quality (or temperature) level of a given energy system decreases as it is processed.

Peaking with Gas Refrigeration

Equally attractive performance advantages can be obtained when employing, two stage absorption (i.e. with a COP of approximately 1.04) or gas fired chiller-heater refrigeration equipment in lieu of the above referenced two stage electric chiller (i.e. similar to Carrier 19HR series chiller, for example). The net effect of permitting the chilled water supply temperature of a nominal 100 ton two-stage absorption chiller rise only 8°F (i.e. from say 42 to 50°F) will increase chiller available refrigeration capacity from 19 to 31% depending upon entering the condenser water temperatures ranging from 80 to 90°F as illustrated in Figure 6.

Low chilled water temperatures are not essential for maintaining comfort. Conventional refrigeration systems generally require excessive refrigeration capacity by forcing sensible cooling and dehumidification to take place at the same cooling coil. Evaporative chilling can be used advantageously with both and electric motor or gas driven refrigeration equipment provided sensible cooling and dehumidification requirements are un-coupled. Therefore, a gas fired chiller-heater of the same capacity and type as produced by Yasaki of Hitachi could easily be substituted for the two stage electric illustrated chiller in Figure 1 and operated equally as well.

Figure 1 illustrates a typical DOS #1 operating cycle. For example, when thermostat C1 senses that the indoor temperature is too high, the system automatically operates in the cooling mode. In conjunction with the outdoor master control C2, the indoor thermostat activates the air handling unit and allows coolant water to flow from the refrigeration chiller and/or evaporative chiller. The proportion of evaporatively chilled to mechanically chilled water (controlled by valve V3) depends on both indoor and outdoor conditions as sense by C1 and C2, respectively. Coolant water flow to the desiccant contactor section is regulated by valve V4. The amount of chilled water bled into this flow is determined by action of a outdoor/indoor optimizing controller. Actual proportioning of return chilled water between the chiller and the evaporative liquid cooler, is as regulated by valve V5 under the control of C2. Positioning the inlet air dampers at the indirect evaporative chiller determines the mixture wet bulb and the leaving water temperature and also affects auxiliary loads. Outside, relief and dehumidified air supply dampers may be pre-set monthly/seasonably or continuously controlled by local sensors as needed.

The total amount of outside air flowing through the desiccant contactor section is determined by building ventilation requirements and coincident inlet damper positions on indirect evaporative chiller. The regenerator load requirement met by hot water supplied by the chiller heat recovery section is controlled by valve V6. Coolant water requirements are sensed by C2 and regulated by valve V4.

Indoor thermostat C1 also activates the flow of hot water from the "heat recovery" condenser bundle to the various air handling unit heating coils through valve V6. Since liquid desiccant regeneration requirements are reduced during the winter months, the condenser "heat recovery" section easily supply the combined space heating and regeneration loads with ease for most office building application. For conditioned zones that require cooling, operations remain as described. When heat is needed, thermostat C1 initiates the "heating mode" operation. Evaporative chiller operation can be reduced to the level required to provide coolant for the desiccant contactor section only. Should the two stage chiller be in-operative hot water must be made up from a redundant energy source.

Optimizing the DOS #1 System performance also involves close control of the mixed water temperature supplied to entering the air handler (cooling) unit coils. Generally speaking, coil performance criteria establishes the required mixted temperature. Based on the instant cooling load, as the (available) temperature of the evaporatively chilled water falls, its proportion of the total mix that can be supplied to the various air handling unit cooling coils increases, thus automatically reducing mechanically refrigerated chilled water demand and vice versus.

Option #2 System (DOS #2)

Refer to Figure 7 which illustrates proposed three tier air handling unit alternative to that illustrated in Figure 7. Although the desiccant portion of contactor in proposed DOS #2 remains the same as that shown for it as a seperate unit in Figure 1, the principal advantage allows a combination of single duct and fan powered terminal (i.e. drawing air from above ceiling space) directed to interior (or net cooling demand spaces) while warm return air is not mixed but directed to air handling hot air circulating fans serving exterior zones, thereby, allowing the full cooling effect of outdoor ventilation air (which) is preconditioned by desiccant contactor unit to be introduced directly to both spaces, thereby eliminating the need to re-cool warn return air in those zones demanding net cooling, etc.

By physically integrating the air handling and desiccant components into a compact a skid mounted, prefabricated assembly, one obtains the flexibility of a three deck system approach, incorporating ductwork savings and improved energy efficiency.

Option #3 System (DOS #3)

Rankine cycle engine consists of a working fluid, a boiler, a condenser, an expander and a feed pump. Heat is supplied at the boiler, which provides saturated (or super heated) vapor to the expander. Power is extracted in the expander after which is saturated liquid supplied to the feed pump where it is repressurized and returned to the boiler, thereby completing the cycle. Heat is normally rejected from the working fluid to ambient at the condenser. Expander shaft work transmitted through a speed reducing gearbox can be used to drive a refrigeration compressor or generator to produce electrical power.

If one adds to the (induction) motor, a prime mover and an over-speed protection device, the motor will be ready to generate and thereby, serve as a cogenerator. In this one device one can bring a load to speed and to drive it, to regulate the speed of another prime mover, or produce electricity. Such equipment is commercially available with a minimum of controls. System voltage and frequency are established at the induction motor generator terminals. The speed of the generator then determines the amount of generated current and its corresponding power factor. The generator has its design rated output in the same way the induction motor has its nameplate loading. Therefore, by adding a Rankine engine in tandem with single stage chiller we can arrive at an equivalent means of providing heat for regeneration but at a much higher temperature level about 350°F thereby permitting use of a smaller liquid desiccant unit or a solid desiccant unit.

The induction motor-generator automatically operates as a generator in the over-speed range, i.e., above its synchronous speed. It can perform several duties in a string of equipment to which it is interconnected by means of a common shaft. At times when the engine expander is unable to extract available energy or is down for repairs, it operates as a motor to direct drive equipment and can also serve as a speed controller. Figure 8 shows the pressure enthalpy diagram for the Rankine cycle, in which a regenerative heat exchanger is also illustrated. Initial cycle expansion is indicated as starting as state 3 on the saturated vapor line and proceeding at constant entropy to state 4 in the superheat region. The regenerator transfers the enthalpy quantity h_4h_5 from the vapor to the cooler pressurized liquid, raising the liquid enthalpy from h_2 to h_3. However, the primary factors governing the performance characteristics of a Rankine cycle are the working fluid and the operating temperatures and pressures. Of these, temperature has the strongest effect on cycle performance. In particular, for a given fluid, the choice of fluid state at the beginning of the expansion process has a strong influence on cycle efficiency. A simplified energy balance illustrates the cycle potential for a combination of conventional and Rankine cycle cogeneration equipment arranged as to that illustrated in Figure 9.

Option #4 System (DOS #4)

Referring to the Figure 10 DOS #4 integrates cogenerated electric power with available waste heat (i.e. as heat of combustion) used to operate a central regenerator (section) that supplies a multiplicity of contactor units illustrated in Figure 10 each contactor section supplies preconditioned dehumidified air to a multiplicity of air handling units serving various occupied building spaces. The central regenerator is continuously operated at substantial (or full) capacity based on moisture loading requirements of the combined space areas to be conditioned. When the desiccant liquid level rises to a pre-determined maximum level in any one contactor unit, desiccant is automatically diverted to the working fluid storage tank. A portion of diluted desiccant liquid is continuously removed from the tank and supplied to the central regeneration unit while concentrated desiccant liquid is simultaneously returned to storage tank from the central regeneration unit, etc. From concentrated desiccant is automatically pumped to a multiplicity of contactor units from the storage tank as required to maintain satisfactory hydraulic balances throughout the interconnected desiccant distribution system. As shown, each contactor supplies treated ventilation air to one or more separate air handling units, etc. For example, in Figure 10, there are two contactor units and four air handling units shown, with each contactor unit serving two air handling units, etc. As shown in Figure 10, each contactor unit is served from a mechanically chilled and/or evaporatively chilled water source, the proportion of which is dependent upon interior space demands, ambient conditions, etc.

As shown in Figure 10 points 3 and 4 are identified for interconnection between the respective piping sections indicated. However, in practice, point 3 connects either directly to corresponding point 3 (and similarly for point 4), or partially through a network of overhead automatic fire sprinkler piping, serving as branch interconnections of either supply or return flow in such a way as to promote better circulation through the sprinkler mains (not shown) and in accordance with provisions of the National Fire Protection Association (NFPA) Pamphlet-13 for incorporating the distribution of circulated water from building HVAC systems.

The central regenerator section is arranged to serve a number of individual contactor units, (i.e. only two shown) is follows: Each contactor removes moisture from outside and space return air, not shown, to maintain space within predetermined comfort limits. Dilute desiccant is circulated to the working fluid storage tanks as shown. Desiccant from the storage tank is continuously recirculated to and from the central regenerator, so that the average desiccant concentration in the storage tank is capable of meeting the design needs of any contactor which is also supplied with regenerated desiccant from the storage tank. Note that cooling tower water is used to cool the contents of this tank as needed, to maintain a vapor pressure of mixed desiccant solution in the tank within the proper range at all times. In this way, DOS #4 can meet the co-incident peak design requirements of all interconnected, distributed contactor sections. The regenerator is self contained and utilizes jacket and/or engine exhaust waste heat for regenerating the desiccant solution from the tank.

The size of the central regenerator can be substantially less than the conventionally peak design requirement, since by providing a smaller unit but running it at essentially full (or constant) load for the duration of the buildings occupied hours, (and beyond), one can achieve the same <u>total</u> "system" moisture removal capability that a larger regenerator (or a series of coupled contactor/regenerator units), following the actual hourly moisture load would provide by tracking the load directly. Consequently, by coupling the available heat rejection of a package diesel (including jacket water not shown) or turbine primer mover skid mounted and matched to the smaller regenerator, one can increase the utilization of cogeneration equipment thereby assuring a constant cost effective supply of associated electrical power to the building for general building use, etc. This approach is used to select the optimum size of desiccant storage tank and desiccant regenerator, based on suitable, commercially available cogeneration package units. What we have is a situation analogous to TES, with desiccant storage and redistribution providing the 'bank" for leveling of moisture load, and in so doing providing a stable, thermal load for cogeneration load tracking.

BIBLIOGRAPHY

Meckler, Milton., editor, Retrofitting of Commercial, Industrial and Institutional Buildings For Energy Conservation; Van Nostrand Reinhold Co., New York, 1983.

Meckler, Milton., P.E., C.Mfg.E., Energy Conservation in Buildings and Plants; McGraw Hill, Inc., New York, New York, 1981 (ISBN 0-07-041195-6).

Meckler, Milton., and Wen-Yung Chen, editors, Pumps and Pump Systems; ASPE Handbook, American Society of Plumbing Engineers, Sherman Oaks, California, 1983.

Meckler, Milton., Recovering Heat From Pollution Control System; from POWER magazine, September 1973.

Meckler, Milton., On-Site Energy Systems Stage A Comeback; from ELECTRICAL CONSULTANT, March, 1975.

Meckler, Milton., Preparing for On-Site Power; from POWER magazine, April, 1975.

Meckler, Milton., How Energy Costs in Commercial Buildings Can Be Cut 20 to 50%; from ELECTRICAL CONSULTANT, July, 1975.

Meckler, Milton., Operation and Maintenance of On-Site Energy Systems; from MAINTENANCE ENGINEERING, November, 1975.

Meckler, Milton., Preparing an Economic Analysis of an On-Site Energy System; from CONSULTING ENGINEER, February, 1976.

Meckler, Milton., Improving Performance of Existing On-Site Energy Systems; from BUILDING SYSTEMS DESIGN, February/March, 1976.

Meckler, Milton., On-Site Systems Puts Squeeze On Consumption; from HOTEL AND MOTEL MANAGEMENT, March, 1976.

Meckler, Milton., Are Utilities Taking A New Look At On-Site Power Generation?; from ELECTRICAL CONSULTANT, August/September, 1976.

Meckler, Milton., Heat Reclamation Strategies; from BUILDINGS Magazine, Nov., 1976.

Meckler, Milton., Energy Conservation Experience on Large Scale HVAC Systems; from ASHRAE TRANSACTIONS, Vol. 83, Part 1, 1977.

Meckler, Milton., Promising Solar Energy Applications In Large Building Applications; from ASHRAE JOURNAL, November, 1977.

Meckler, Milton., Use of Rankine Cycle HVAC and Selective Power Systems for Cooling of Buildings; from proceedings of Second International Helioscience Institute Conference and Exhibition on Alternate Energy, University of California, Riverside, April 1978.

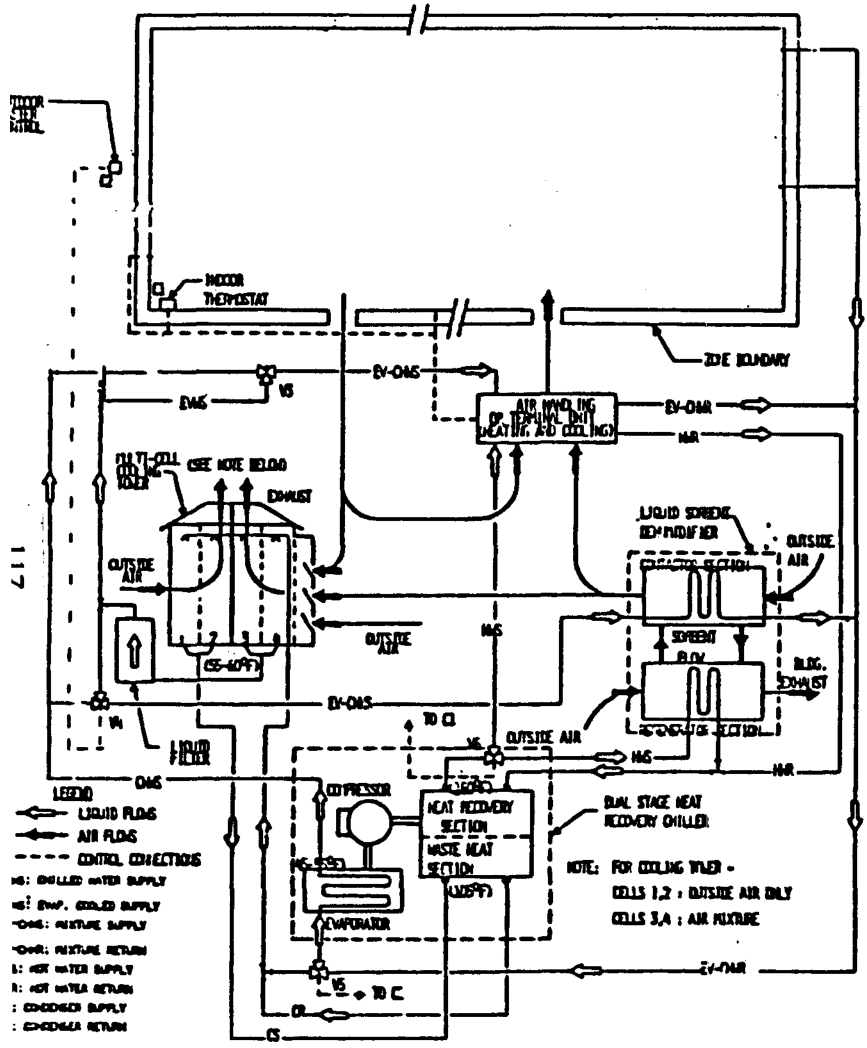
Meckler, Milton., A Novel Cooling System That Cuts Energy Usage (Part 1); from SPECIFYING ENGINEER, January, 1982.

Meckler, Milton., A Novel Cooling System That Cuts Energy Usage (Part 2); from SPECIFYING ENGINEER, March, 1982.

Meckler, Milton., Ventilation/Air Distribution Cure Sick Buildings; from SPECIFYING ENGINEER, January, 1985.

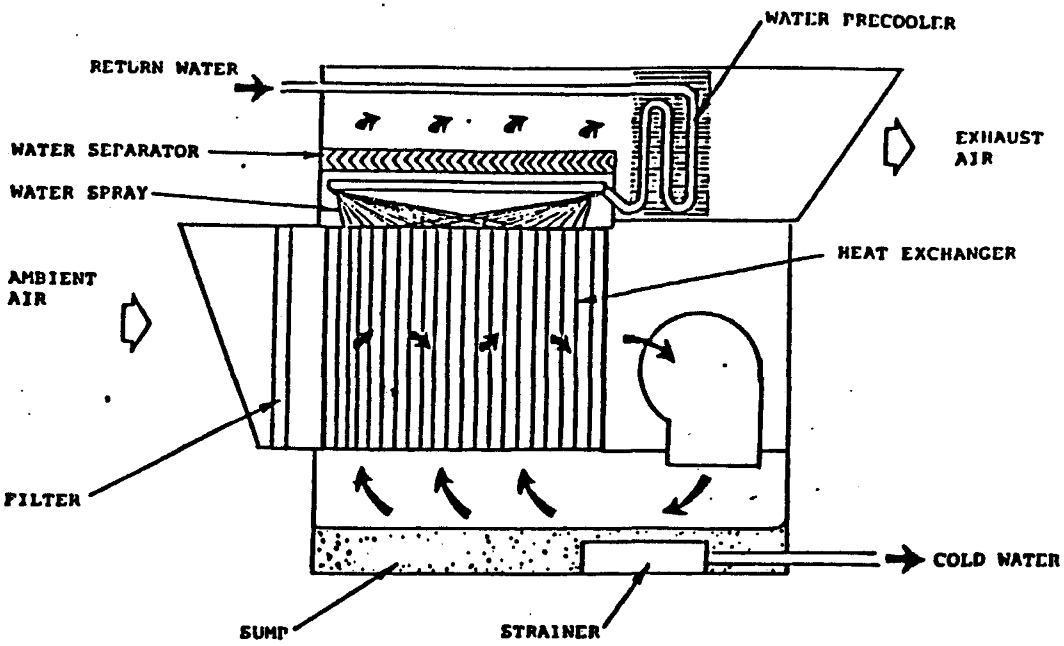
Meckler, Milton., Retrofitting of Cooling Towers For Cost Effective Evaporative Chilling - A Case History; from SPECIFYING ENGINEER, April, 1986.

US 4, 024,908; US 4, 300,623; US 4, 490,981; PENDING PATENT APPLICATIONS



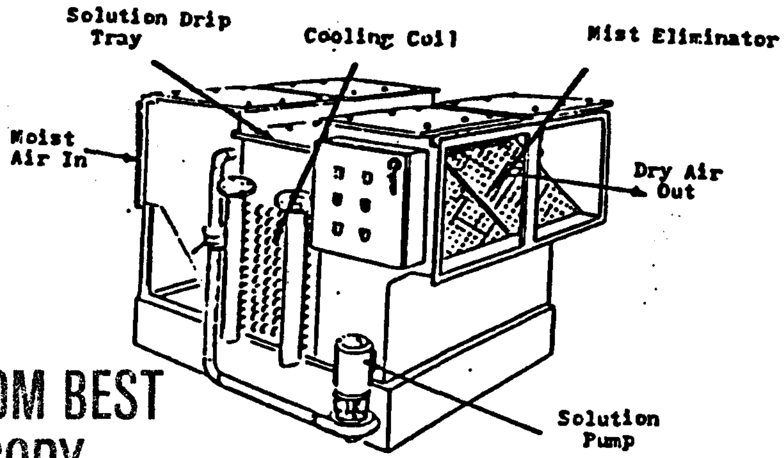
DOS #1 - DESICCANT COOLING SYSTEM

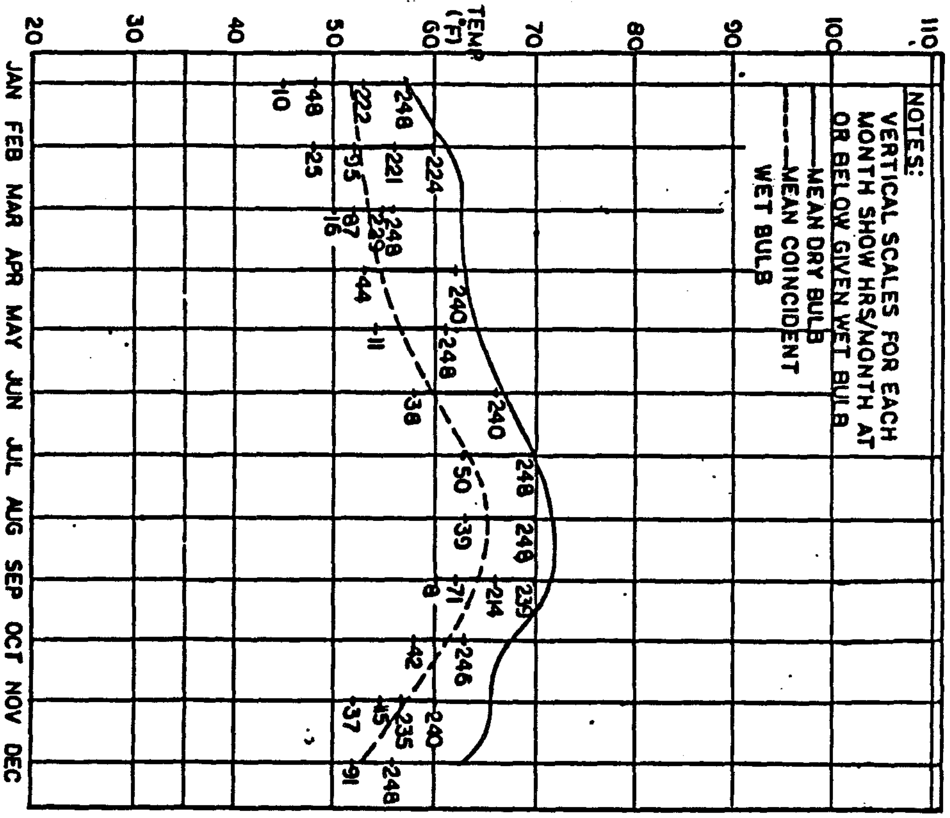
FIGURE 1



INDIRECT EVAPORATIVE CHILLER

FIGURE 2





LOS ANGELES DRY BULB AND COINCIDENT BULB COMPARISON

FIGURE 4

COMPARATIVE PERFORMANCE TWO STAGE ABSORPTION CHILLER

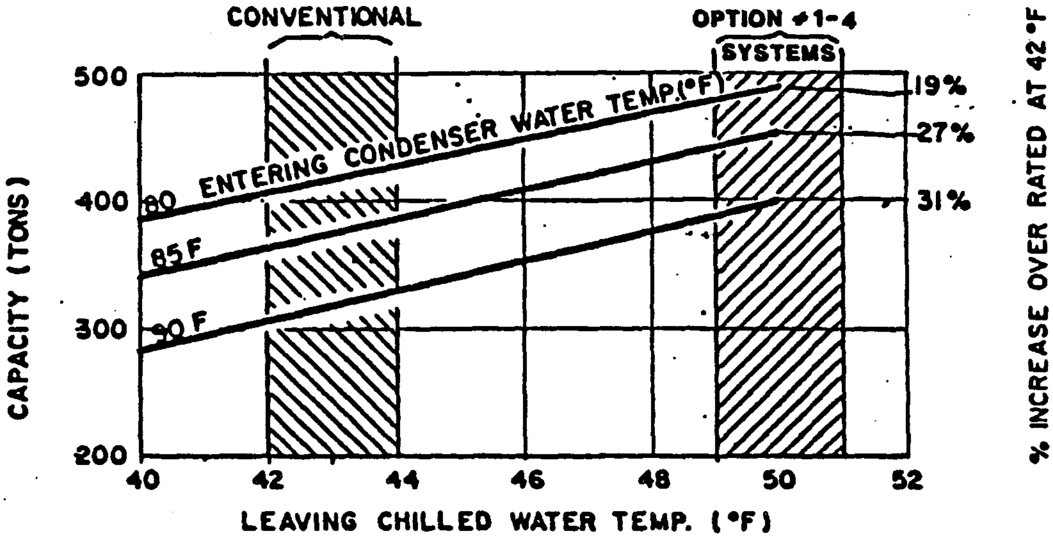
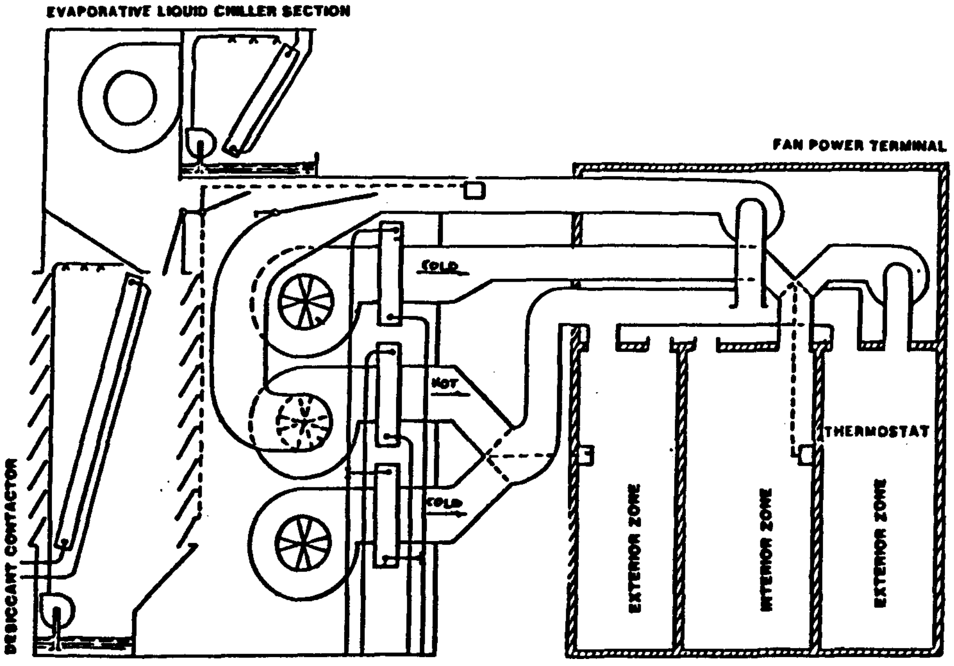


FIGURE 6

```
                              SUMMARIZED MONTHLY ENERGY COMPARISON

                 SYSTEM ONE (Existing System)                     SYSTEM TWO (With Energy Master)
              ----------------------------------------------   ---------------------------------------------
              PEAK    MID PK   OFF PK    TOTAL      DOLLARS     PEAK    MID PK   OFF PK    TOTAL     DOLLARS    DOLLARS        $ ENERGY CHARGE              $ DEMAND CHARGE
              KWH      KWH      KWH       KWH       TOTAL       KWH      KWH      KWH       KWH      TOTAL      SAVED      PEAK    MID PEAK  OFF PEAK    PEAK    MID PEAK
JANUARY       1226.      0.       0.     1226.       $53.       215.      0.       0.       215.       $9.       $43.    $0.0430  $0.0000  $0.0000    $5.650  $0.000
FE   RY       1635.      0.       0.     1635.       $70.       282.      0.       0.       282.      $12.       $58.    $0.0430  $0.0000  $0.0000    $5.650  $0.000
 A            7058.      0.       0.     7058.      $304.      1130.      0.       0.      1130.      $49.      $255.    $0.0430  $0.0000  $0.0000    $5.650  $0.000
 PI          22816.      0.       0.    22816.      $981.      6366.      0.       0.      6366.     $274.      $707.    $0.0430  $0.0000  $0.0000    $5.650  $0.000
 AY          48808.      0.       0.    48808.     $2099.     25002.      0.       0.     25002.    $1075.     $1024.    $0.0430  $0.0000  $0.0000    $5.650  $0.000
JUNE         93155.      0.       0.    93155.     $4024.     54582.      0.       0.     54582.    $2358.     $1666.    $0.0432  $0.0000  $0.0000   $14.850  $0.000
JUL         118795.      0.       0.   118795.     $5132.     66801.      0.       0.     66801.    $2886.     $2246.    $0.0432  $0.0000  $0.0000   $14.850  $0.000
 U          114551.      0.       0.   114551.     $4949.     64443.      0.       0.     64443.    $2784.     $2165.    $0.0432  $0.0000  $0.0000   $14.850  $0.000
SE   BER     77494.      0.       0.    77494.     $3348.     42279.      0.       0.     42279.    $1826.     $1521.    $0.0432  $0.0000  $0.0000   $14.850  $0.000
 C   R       30930.      0.       0.    30930.     $1336.     13974.      0.       0.     13974.     $604.      $733.    $0.0432  $0,0000  $0.0000   $14.850  $0.000
 OVEMBER     10394.      0.       0.    10394.      $447.      2845.      0.       0.      2845.     $122.      $325.    $0.0430  $0.0000  $0.0000    $5.650  $0.000
 ECEMBER      3373.      0.       0.     3373.      $145.       597.      0.       0.       597.      $26.      $119.    $0.0430  $0.0000  $0.0000    $5.650  $0.000
--  ---------------------------------------------------------------------------------------------------------------------------
Tot   :     530238.      0.       0.   530238.    $22887.    278516.      0.       0.    278516.   $12025.    $10863.


          SUMMARIZED MONTHLY DEMAND COMPARISION

                SYSTEM ONE (Existing system)   SYSTEM TWO (With Energy Master)                  TOTAL UTILITY COST            YEARLY SAVINGS              REP SYSTEM
               ---------------------------   -------------------------------                ------------------------   ---------------------------------  OPERATIONS
             PEAK   MID PK      DOLLARS      PEAK    MID PK   DOLLARS      DOLLARS       SYSTEM 1     SYSTEM 2     FST YR     SND YR     TRD YR    SYSTEM 1 SYSTEM 2
            DEMAND  DEMAND       TOTAL      DEMAND   DEMAND    TOTAL        SAVED          COST         COST      SAVINGS    SAVINGS    SAVINGS    HOURS     HOURS       EER1
JAN  ARY      64     0           $363         17        0       $98         $265           $416         $108       $308       $355       $408        27         0        8.11
FE   ARY      74     0           $419         17        0       $98         $321           $490         $110       $379       $436       $501        38         0        8.10
 A           130     0           $732         17        0       $98         $634          $1036         $147       $889      $1022      $1176       124         0       10.37
 P           249     0          $1405        154        0      $870         $535          $2387        $1144      $1243      $1429      $1644       323        23       11.58
 AY          264     0          $1492        163        0      $919         $573          $3591        $1994      $1597      $1836      $2112       465       162       12.71
JUNE         335     0          $4978        202        0     $3007        $1971          $9002        $5365      $3637      $4183      $4810       478       413       13.08
JUI          335     0          $4978        197        0     $2921        $2056         $10110        $5807      $4303      $4948      $5690       496       480       13.02
 U     T     335     0          $4978        191        0     $2839        $2139          $9926        $5622      $4304      $4949      $5692       497       467       13.02
SE    BER    335     0          $4978        189        0     $2810        $2168          $8325        $4636      $3689      $4243      $4879       468       307       12.96
 CTOBER      240     0          $3562        144        0     $2137        $1425          $4898        $2741      $2157      $2481      $2853       395        86       11.89
 OVEMBER     186     0          $1052        121        0      $682         $370          $1499         $804       $695       $799       $919       201         5        9.73
 EC   BER     85     0           $481         17        0       $98         $383           $626         $124       $502       $577       $664        76         0        8.58
--  ---------------------------------------------------------------------------------------------------------------------------------------------------------------------
Tot   :     2633     0         $29418       1430        0    $16577       $12841         $52305       $28602    $23703.    $26548.    $29734.    3589  1943

    ANNUAL AVERAGE: EER1= 12.19  EER2= 28.99                 TEN YEARS ENERGY SAVINGS:    $415966.


     AUGUST
     TDB   TWB    PEAK   MIDPK   OFFPK   SYS1 SUP.AIR   SYS2 SUP.AIR   BMIDB  BMIWB  TDIR   BWORK BUILD  SYSTEM1 SYSTEM2  SYS1    SYS2    SYS1 TOTAL   SYS2 TOTAL    KWH    TWR
      F     F     HOURS  HOURS   HOURS   TDB     TWB    TDB     TWB    DEG    DEG    DEG    TONS  LD TON  LD TON  LD TON   KW      KW        KWH          KWH        SAVED   TEMP
     51    48.8    1.2    0.0     0.0    70.0    61.4   70.0    61.4   49.5   48.1   48.5    6.8  158.9    0.0     0.0     0.0     0.0       0.0          0.0          0.0   86.0
     53    50.4    1.2    0.0     0.0    69.8    61.3   69.8    61.3   53.0   50.4   51.1    0.0  161.5    9.9     0.0    34.0     4.2      40.8          5.0         35.8   86.0
     55    52.0    1.2    0.0     0.0    69.7    61.2   69.7    61.2   53.0   51.1   51.6    9.3  164.0   21.0     0.0    36.9    17.4      44.2         20.9         23.4   52.8
     57    53.6    1.2    0.0     0.0    69.6    61.2   69.6    61.2   54.7   52.6   53.2   10.5  166.6   32.2     0.0    42.2    17.4      50.6         20.9         29.7   54.9
     59    55.5    2.5    0.0     0.0    69.5    61.1   69.5    61.1   56.6   54.5   55.1   10.8  169.1   46.5     0.0    50.8    24.9     128.0         62.7         65.2   57.5
     61    57.5    4.3    0.0     0.0    69.3    61.1   69.3    61.1   58.6   56.5   57.1   10.8  171.7   73.1     0.0    66.7    24.9     285.6        106.6        179.0   60.3
     63    59.4    7.0    0.0     0.0    69.2    61.0   69.2    61.0   60.6   58.4   59.0   11.1  174.2   99.5     0.0    93.0    24.9     651.1        174.3        476.8   62.9
     65    61.0   12.6    0.0     0.0    69.1    61.0   69.1    61.0   62.3   59.9   60.6   12.4  176.8  122.8     0.0   117.0    24.9    1473.7        313.7       1159.9   65.1
     67    62.6   18.2    0.0     0.0    68.9    60.9   68.9    60.9   64.0   61.5   62.2   13.6  179.4  146.9    73.8   128.4    89.3    2336.8       1625.2        711.6   67.4
     69    64.2   27.2    0.0     0.0    68.8    60.8   68.8    60.8   65.8   63.0   63.8   14.9  181.9  171.7    84.2   152.5    99.5    4142.5       2703.1       1439.4   69.6
     71    65.8   37.2    0.0     0.0    68.7    60.8   68.7    60.8   67.5   64.5   65.4   16.1  184.5  197.2    94.7   178.8   110.3    6659.0       4108.5       2550.5   71.9
     73    67.2   45.4    0.0     0.0    68.6    60.7   68.6    60.7   69.1   65.8   66.7   17.9  187.0  220.6   104.6   202.8   120.4    9215.6       5472.8       3742.8   73.9
     75    68.0   48.0    0.0     0.0    68.4    60.7   68.4    60.7   70.3   66.4   67.5   21.7  189.6  235.2   112.3   217.8   128.4   10454.7       6161.4       4293.2   75.0
     77    68.8   50.6    0.0     0.0    68.3    60.6   68.3    60.6   71.5   67.0   68.2   25.4  192.1  250.0   120.0   227.3   136.3   11493.7       6891.3       4602.4   76.1
     79    69.6   48.6    0.0     0.0    68.2    60.5   68.2    60.5   72.6   67.5   69.0   29.1  194.7  265.1   127.8   242.8   139.2   11789.4       6757.7       5031.7   77.2
     81    70.4   45.0    0.0     0.0    68.0    60.5   68.0    60.5   73.8   68.1   69.7   32.8  197.2  280.3   135.5   258.5   143.8   11641.3       6477.0       5164.3   78.4
     83    71.2   40.7    0.0     0.0    67.9    60.4   67.9    60.4   75.0   68.7   70.5   36.5  199.8  295.8   143.3   274.4   148.5   11173.1       6045.2       5127.9   79.5
     85    72.0   34.0    0.0     0.0    67.8    60.4   67.8    60.4   76.2   69.3   71.2   40.2  202.3  311.6   151.1   290.6   153.6    9878.9       5221.6       4657.3   80.6
     87    72.8   27.3    0.0     0.0    67.7    60.3   67.7    60.3   77.4   69.9   72.0   43.9  204.9  327.5   158.9   307.0   161.6    8374.2       4408.0       3966.3   81.8
     89    73.6   20.1    0.0     0.0    67.5    60.3   67.5    60.3   78.6   70.5   72.8   47.7  207.4  343.8   166.7   323.6   169.6    6498.8       3405.7       3093.1   82.9
     91    74.4   12.7    0.0     0.0    67.4    60.2   67.4    60.2   79.8   71.1   73.5   51.4  210.0  360.2   174.5   335.2   177.6    4263.7       2259.6       2004.1   84.1
     93    74.9    6.5    0.0     0.0    67.3    60.1   67.3    60.1   80.8   71.4   74.0   56.0  212.6  371.5   181.2   335.2   184.5    2185.5       1203.1        982.4   84.8
     95    74.5    3.8    0.0     0.0    67.1    60.1   67.1    60.1   81.1   70.4   73.4   63.4  215.1  366.4   184.4   335.2   187.8    1273.8        713.8        559.9   84.0
     97    74.1    1.1    0.0     0.0    67.0    60.0   67.0    60.0   81.5   69.5   72.9   70.9  217.7  361.4   187.6   335.2   191.1     362.0        206.4        155.6   83.3
     99    74.0    0.3    0.0     0.0    66.9    60.0   66.9    60.0   82.1   69.0   72.7   77.4  220.2  361.8   192.0   335.2   195.6      93.9         54.8         39.1   83.0
    101    74.0    0.1    0.0     0.0    66.8    59.9   66.8    59.9   82.7   68.6   72.5   83.6  222.8  364.0   196.7   335.2   200.5      40.2         24.1         16.2   82.9
      MONTHLY ENERGY CONSUMPTION SYSTEM ONE= 114551.    MONTHLY ENERGY CONSUMPTION SYSTEM TWO=  64443.     MONTHLY ENERGY SAVED=  50108.    EER1= 13.02   EER2=    23.14
```

REPRESENTATIVE DBS COMPUTER SIMULATION RUN

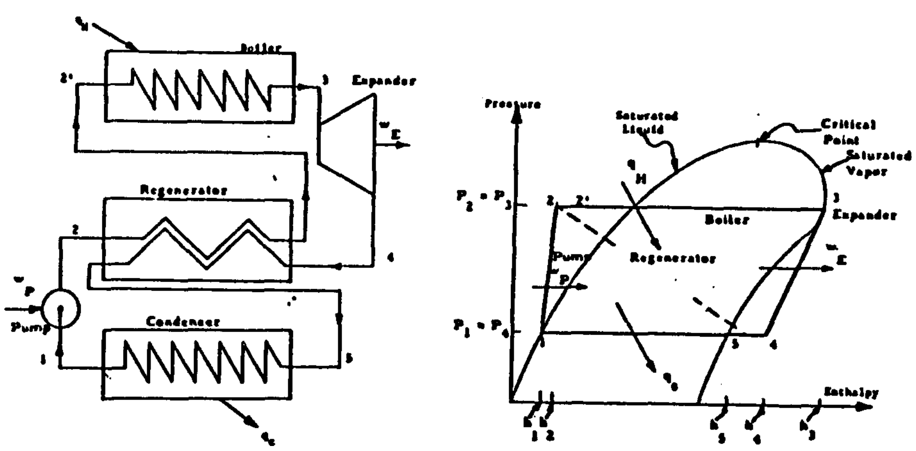
FIGURE 5



THREE TIER INTEGRATED DESICCANT CHILLING AIR HANDLING SYSTEM

FIGURE 7

RANKLINE CYCLE SCHEMATIC and GENERALIZED PRESSURE - ENTHALPY DIAGRAM



$$\eta_o = \left[\frac{W_E - W_P}{Q_H}\right] = \left[\frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2) - (h_4 - h_5)}\right] \qquad (1)$$

and if we let $\epsilon = \dfrac{[\Delta m\,\Delta w\,^{H}(g1)]}{[l(h_5 - h_1)]}$,

the revised regenerative cycle efficiency n'_o becomes:

$$n'_o = \left[\frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_2) - (h_4 - h_5) - \epsilon(h_5 - h_1)}\right] \qquad (2)$$

FIGURE 8

TABLE 1—CHILLER EFFICIENCY/CAPACITY SELECTION CRITERIA

Compressor type	*Capacity (tons)*	*Coefficient of performance*
Hermetic, water-cooled reciprocating	10 to 185	3.65
Hermetic, air-cooled reciprocating	2.5 to 165	2.85
Open, water-cooled reciprocating	10 to 240	3.87
Open, air-cooled reciprocating	10 to 240	3.02
Hermetic, screw	100 to 750	3.3 to 4.3
Open, screw	500 to 750	3.5 to 4.3
Hermetic centrifugal (factory-assembled)	800 to 2,000	4.1 to 4.65
Open centrifugal (factory-assembled)	90 to 1,250	4.2 to 4.9
Open centrifugal (field-assembled)	700 to 13,000	—

(Full load value/ranges)

Source: Central Cooling-Compressive Chillers, Argonne National Laboratory, Pub. ANL.CES-TE 78-2

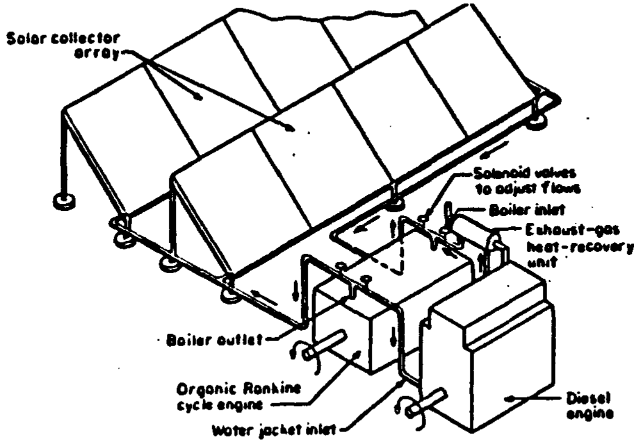
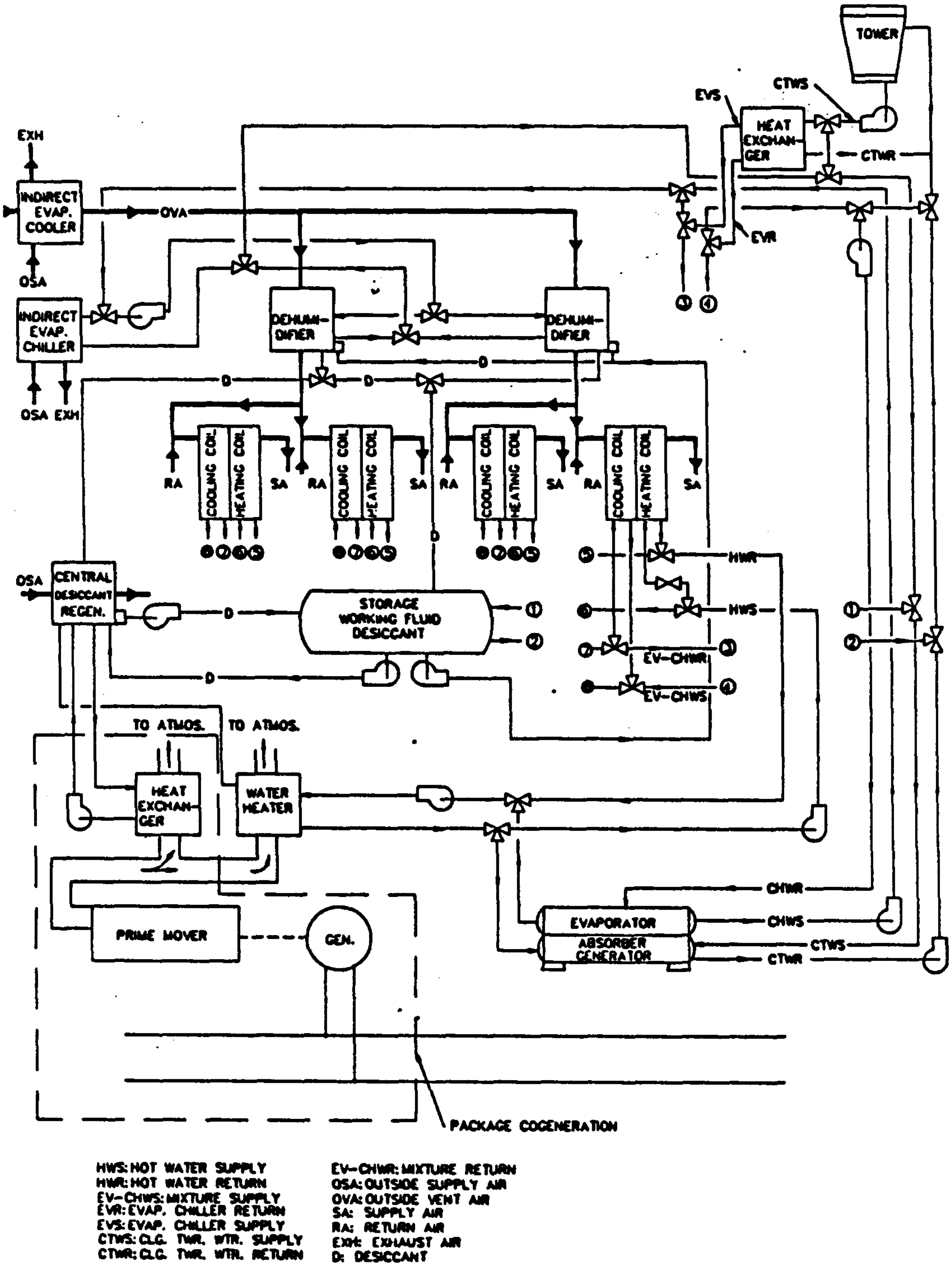


FIGURE 9



CENTRAL DESICCANT REGENERATION/CO-GENERATION SYSTEM

FIGURE 10

DESICCANT MATERIALS

by

Dr. R.K. Collier, Jr.

Enerscope, Inc.

ABSTRACT

The effect that the desiccant adsorption properties have on the performance of open-cycle desiccant cooling systems can be quite marked. The desiccant property that has the greatest effect is the shape of the adsorption isotherm. The isotherm shape determines the shape that the resultant heat and mass transfer wavefronts operating within the desiccant matrix will assume. The containment of these heat and mass transfer wavefronts is crucial to achieving the maximum possible performance from the cooling/dehumidification system. Isotherms that enhance wavefront behavior during the dehumidification process degrade that behavior during the regeneration process and vice versa. The most desirable isotherm shape therefore becomes a compromise between the two processes. The isotherm shape that yields the best compromise for adiabatic systems is a modified Brunauer Type 1.

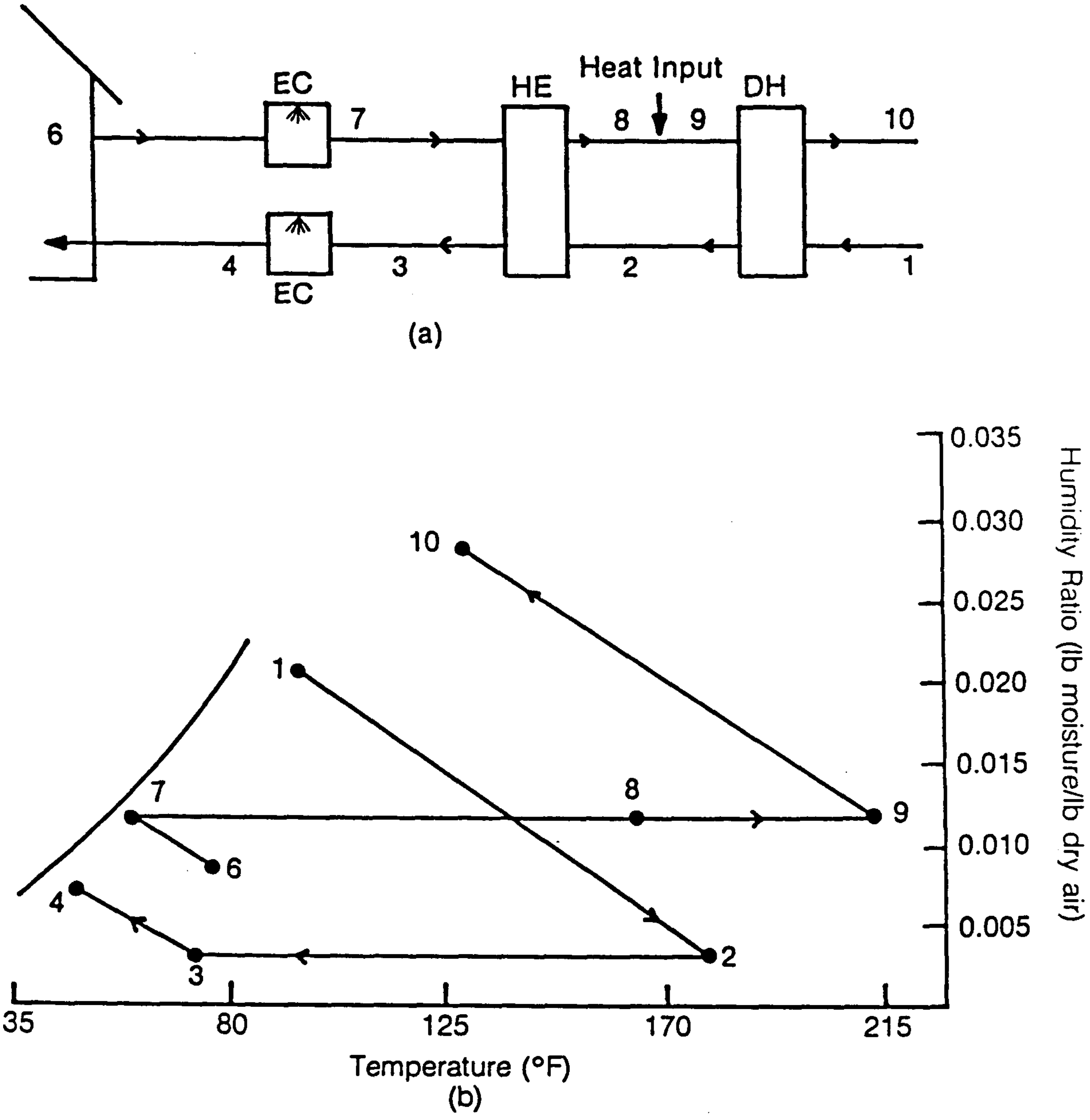
BACKGROUND

Open cycle desiccant cooling systems process water vapor between a conditioned space and the environment. The use of thermal energy creates a chemical potential that can be used to produce a cooling effect. In simpler terms, if the air can be made dry enough, an evaporative cooler will produce air that is as cold as a conventional electric air conditioner. The overall performance of these systems rely on the quality of the thermal energy input (availability) and the environment as both a cold sink and as a source of chemical potential (unsaturated air). The major solid-desiccant cooling cycles consist of a desiccant dehumidifier, a sensible heat exchanger and two evaporative coolers. There are two important modes of operation: 1) The ventilation mode where outdoor air is continually processed to produce low enthalpy air for the cooled space, and 2) The recirculation mode where air from the cooled space is continually processed to maintain low enthalpy air conditions in the space.

A schematic of the ventilation mode and a psychrometric representation of the cycle is shown in Figure 1. Ambient air at (1) is adiabatically dehumidified by the desiccant (DH). The hot, dry air at (2) is cooled by the sensible heat exchanger (HX) to create dry cool air (3). This air is then adiabatically humidified by the evaporative cooler (EC) to produce cold, nearly saturated air (4) that enters the building. Simultaneously, an equal amount of building air (6) is adiabatically humidified to (7). This produces the cold sink for the dry air (3). The air is then heated by the same heat exchanger to (8). This is an attempt to recouperate as much of the heat of adsorption as possible from the dehumidification process. The enthalpy of the air must now be increased to (9) by a thermal energy input. This air is then passed through the desiccant in order to regenerate it. The warm, humid air that exits the desiccant (10) is then exhausted to the atmosphere.

A schematic of the recirculation mode and a psychrometric representation of the cycle is shown in Figure 2. This time ambient air (1) is adiabatically humidified to (2). This air now becomes the cold sink for the cycle. It is heated by the sensible heat exchanger to (3) and then further heated to (4) by an external thermal input. This air is then used to regenerate the desiccant material. The warm, humid air that exits the desiccant (5) is returned to the environment. Simultaneously, room air (6) is adiabatically dehumidified by the desiccant to produce warm, dry air (7). This air is then heat exchanged with the humidified and cooled ambient air to create cool, dry air (8). It is then adiabatically humidified to produce the cold, near-saturated air (9) which is returned to the building.

Both of the operational modes previously described operate on a continuous basis. That is, both the dehumidification and the regeneration processes occur simultaneously. In order to accomplish this, the desiccant is deployed into a rotating wheel or drum that continuously cycles the desiccant between the



EC = evaporative cooler
HE = heat exchanger
DH = dehumidifier

Fig. 1 Ventilation mode of desiccant cooling system

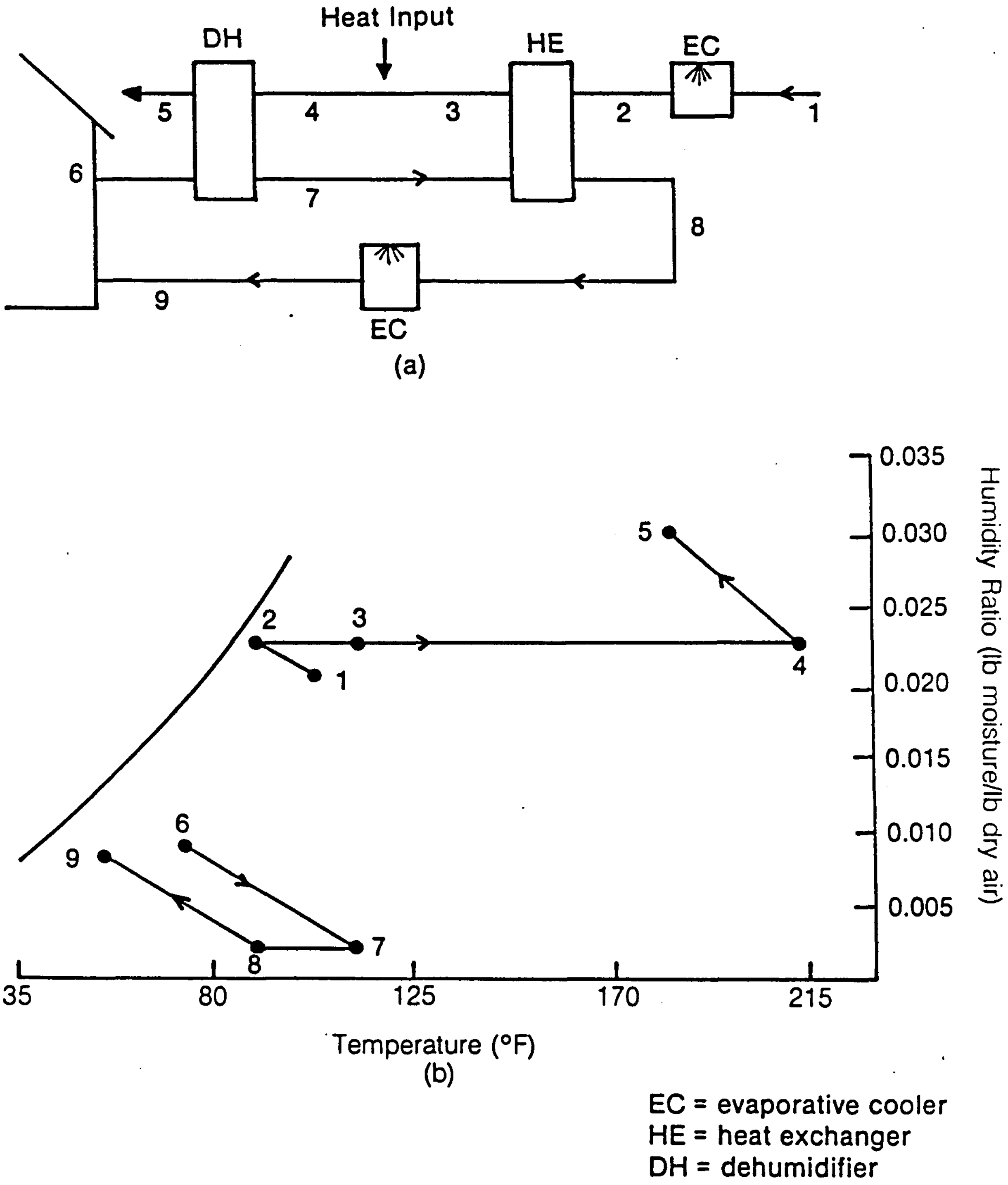


Fig. 2 Recirculation mode of desiccant cooling system

dehumidification and regeneration air streams. The heat exchanger may or may not be of a rotating design and the evaporative coolers must always be distinctly separate units.

There are four important, interrelated performance parameters that determine the viability of desiccant cooling systems. They are:

1) Thermal coefficient of performance (COP).

2) Energy efficiency ratio (EER).

3) Physical size

4) Specific cooling capacity (inverse of CFM/ton of cooling)

The thermal COP is a measure of the amount of cooling that is produced by a given amount of thermal energy input. It determines ultimately the cost of the thermal energy source (natural gas) required to cool the building. The EER is a measure of the amount of cooling, in thermal units, that is produced by a given amount of electrical energy input, in electrical units. It determines ultimately the cost of the electrical energy required to cool the building. The physical size of the equipment is important because it not only effects the initial cost of the equipment, but also influences the aesthetic desirability of the equipment in the marketplace. The specific cooling capacity is a measure of the amount amount of cooling obtained for a given amount of air. The higher the specific cooling capacity, the less air is needed for a given cooling load. The lower the specific cooling capacity, the more air is needed for a given cooling load.

Examples of the trade-offs between these performance parameters are as follows: Increasing the thermal COP can serously decrease the specific cooling capacity. This will result in a increase in the amount of air needed to meet the building load which will decrease the EER (increase electricity cost) and/or increase the physical size of the equipment. Increasing the effectiveness of the sensible heat exchanger will increase both the thermal COP and the specific cooling capacity. This will result in a larger physical size and/or increased pressure drop which again will decrease the EER. Increasing the regeneration temperature will, with conventional desiccant materials, increase the specific cooling capacity, but it will also decrease the thermal COP of the system.

The effect that the desiccant material properties have on the system performance parameters can best be seen by examining psychrometrically the processes that that occur within the dehumidifier wheel. For both the dehumidification and the regeneration processes, there are two fundamental wavefronts that occur. The first and fastest wave is primarily a thermal front that is most affected by the total amount of thermal heat capacity associated with the dehumidifier. The second and slowest wave is the main concentration wavefront with strong associated thermal effects. The outlet air conditions associated with these two waves are shown on psychrometric chart in Figure 3. The locus of points show the time-dependent outlet air conditions for both the dehumidification and the regeneration processes. A constant elapsed time exists between each point plotted. For the dehumidification process, the first wave is represented by the locus of points between the regeneration condition to a position of minimum absolute humidity. The outlet air remains at or near this condition for some time and then experiences the gradual breakthrough of the main concentration wave which creates a locus of points that connect this minimum absolute humidity point with the inlet air conditions. This point on the psychrometric chart has been called the MZ point or middle zone condition that exists between the two passing wavefronts. An identical condition to that of the dehumidification process also exists for the regeneration process. The first and fastest wavefront creates a locus of points that begins at the dehumidifier inlet conditions and progresses quickly to a point of maximum outlet humidity. The air remains at this condition for a period of time and then experiences breakthrough of the main concentration front with the outlet conditions gradually approaching the inlet regeneration air conditions. It is important to notice that the outlet air conditions depicted in Figures 1 and 2 are average outlet conditions that represent the time-averaged effects of both the first and second waves associated with the heat and mass transfer within the desiccant matrix. As will be shown later, the most thermodynamically advantageous conditions for optimal cycle operation are the two MZ points associated with both the dehumidification and regeneration processes.

On the dehumidification side, the position of the average outlet air condition on the psychrometric chart is critical to determining the ultimate cooling performance of the system. The lower its absolute humidity, the higher the cooling capacity of the system. The higher its temperature, the higher the preheat temperature of the regeneration air will be. This reduces the amount of external thermal energy required to regenerate the desiccant and subsequently increases the thermal COP. The effect of the first wave is to increase the average outlet humidity while increasing the average outlet temperature. The effect of the second wave is to increase average outlet humidity and to decrease average outlet temperature. Therefore, in order to achieve the maximum possible performance, it is necessary to avoid any significant breakthrough of the second wavefront and to increase the speed of the first. This will result in the average outlet air condition being as close to the MZ point (the thermodynamic optimum) as possible.

On the regeneration side, a similar situation exists although the ramifications of the average outlet air conditions on system performance are not so widely appreciated. The average outlet air condition for the regeneration process should represent the minimum thermodynamic availability that is consistent with the

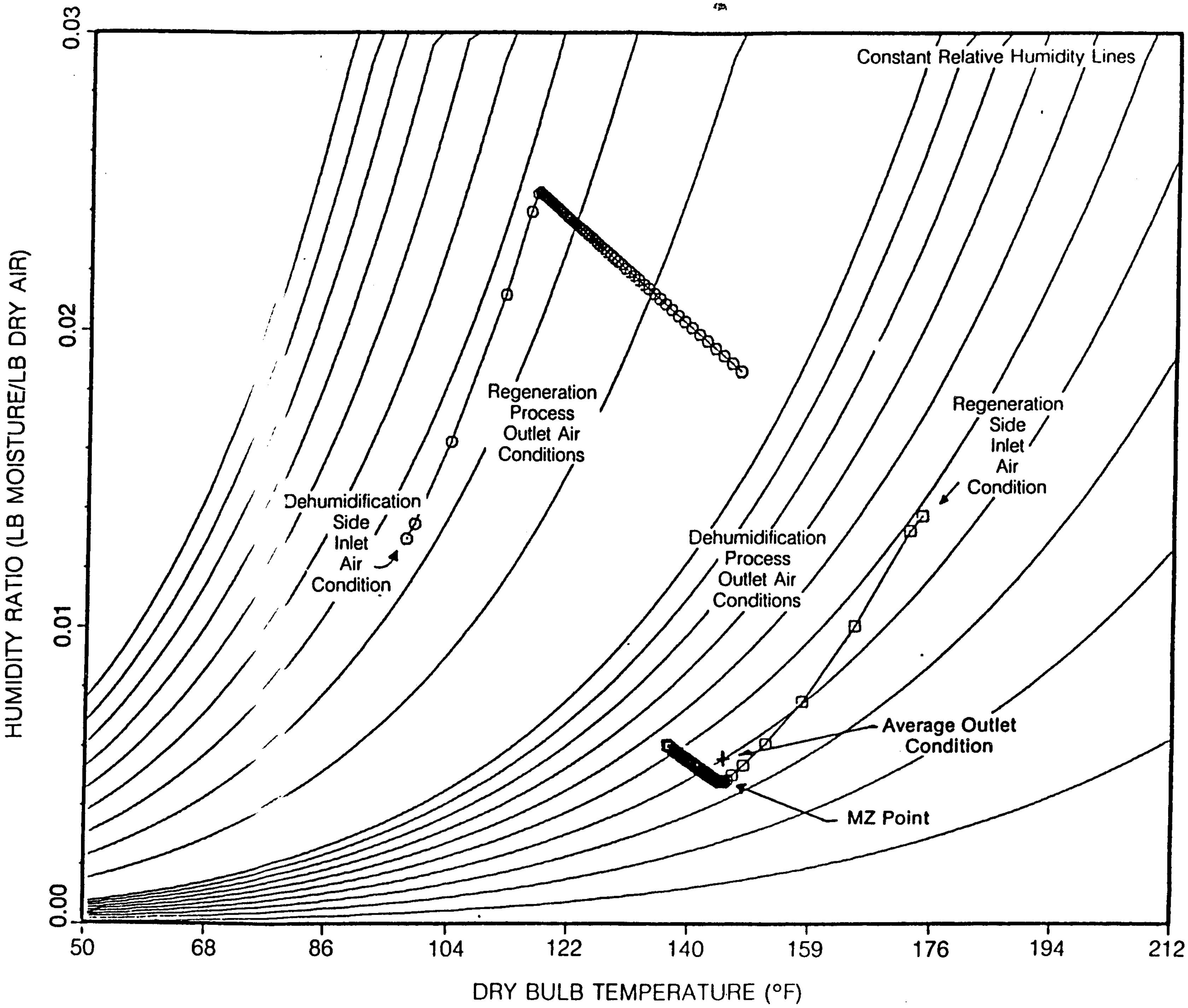


Fig. 3 Time-dependent air conditions exiting each side of dehumidifier

Clausius-Clapeyron limits of the cycle. The air condition consistent with this hypothesis is the MZ point for regeneration. This again indicates that avoiding breakthrough of the concentration wavefront is paramount to attaining maximum efficiency.

The preceeding discussion has been an attempt to show that the primary function of the desiccant material in an open cycle desiccant cooling system should be to produce the sharpest possible concentration wavefronts for both the dehumidification and the regeneration processes. The determination of just how sharp these wavefronts should be will be dictated by the performance parameters discussed earlier. The most important being the physical size of the equipment and the EER. The desiccant matrix itself has certain physical limitations. The most obvious one is it's length in the direction of air flow. Difficulties with weight, manufacturability and pressure drop all influence the maximum length that is possible. Another important aspect is the geometry used to deploy the desiccant within the matrix. For example, a packed bed of spherical or nearly-spherical desiccant particles would probably be the cheapest method of manufacture. However, because of the limit on the allowable pressure drop within the system, this configuration is limited to lengths of two inches or so. On the other hand, laminar flow channel geometries with desiccant-laden walls could be up to two feet long without suffering from excessive pressure losses and the concommitant loss in EER. These designs suffer more from a manufacturability limit rather than a pressure drop limit. This manufacturability limit appears to be on the order of eight to ten inches. These constraints then dictate the degree of wavefront steepness required to meet our design goal of cyclic operation of the desiccant cooling device without experiencing concentration wave breakthrough with either the dehumidification or the regeneration processes.

DESCRIPTION OF DESIRED PROPERTIES

The desiccant material properties that most influence the ability to achieve the goals of the dehumidifier device described in the previous section are as follows:

1) Isotherm Shape

2) Adsorption Energy

3) Maximum Uptake

4) Chemical and Physical Stability

ISOTHERM SHAPE

The desiccant isothem shape is the single most important factor in determining the wavefront shapes within the dehumidifier of the cooling system. One isotherm shape that will result in the steep wavefronts needed to be contained within the 8 to 10 inch length of the dehumidifier matrix is shown in Figure 4. This isotherm is characterized by the constant separation factor equation:

$$SC = FC/(R + FC - R*FC)$$

where: SC = relative solid concentration
FC = relative fluid concentration
R = separation factor

The separation factor used for the isotherm in Figure 4 is 0.1. It represents a compromise between the concentration wavefront behavior on both the dehumidifying and the regeneration processes. Separation factors less than 0.1 steepen wavefronts during the dehumidification process, but broaden wavefronts during the regeneration process. For separation factors greater than 0.1 the opposite effect results. Wavefronts are broadened during the dehumidification process, but steepened during the regeneration process. A parameter that plays an important role in determining the compromised isotherm shape is the regeneration temperature. High regeneration temperatures steepen the regeneration wavefronts as the value of the separation factor decreases and broaden the dehumidification wavefronts as the value of the separtion factor increases. On the other hand, low regeneration temperatures broaden the regeneration wavefronts as the value of the separation factor decreases and steepen the dehumidification wavefronts as the value of the separation factor increases. It is important to notice that the so-called "linear" isotherm (R=1) does not produce symmetrical wavefronts between dehumidification and regeneration due to the process being adiabatic and the saturation pressure of water vapor in air being a strong function of temperature. The regeneration temperature that produces the acceptably steep wavefronts for R = 0.1 is 320 deg F. The effect that regeneration temperature has on the performance of the cooling cycle is very important in determining the proper isotherm shape for the desiccant. As discussed in the previous section, the specific cooling capacity of the cycle increases with increasing regeneration temperature. There is a point of diminishing returns, however, because the moisture in the air goes to zero asymptotically. Factored into this is the efficiency of the heat exchanger between the gas flame and the air stream. The net result is that the highest possible regeneration temperature that is consistent with efficient heat transfer to the regeneration air stream and the chemical and physical stability of the desiccant matrix is desired. The higher this regeneration temperature, the lower the value of the separation factor that yields optimal system performance.

ADSORPTION ENERGY

The second most important property of the desiccant is the adsorption energy. There are two aspects to the adsorption

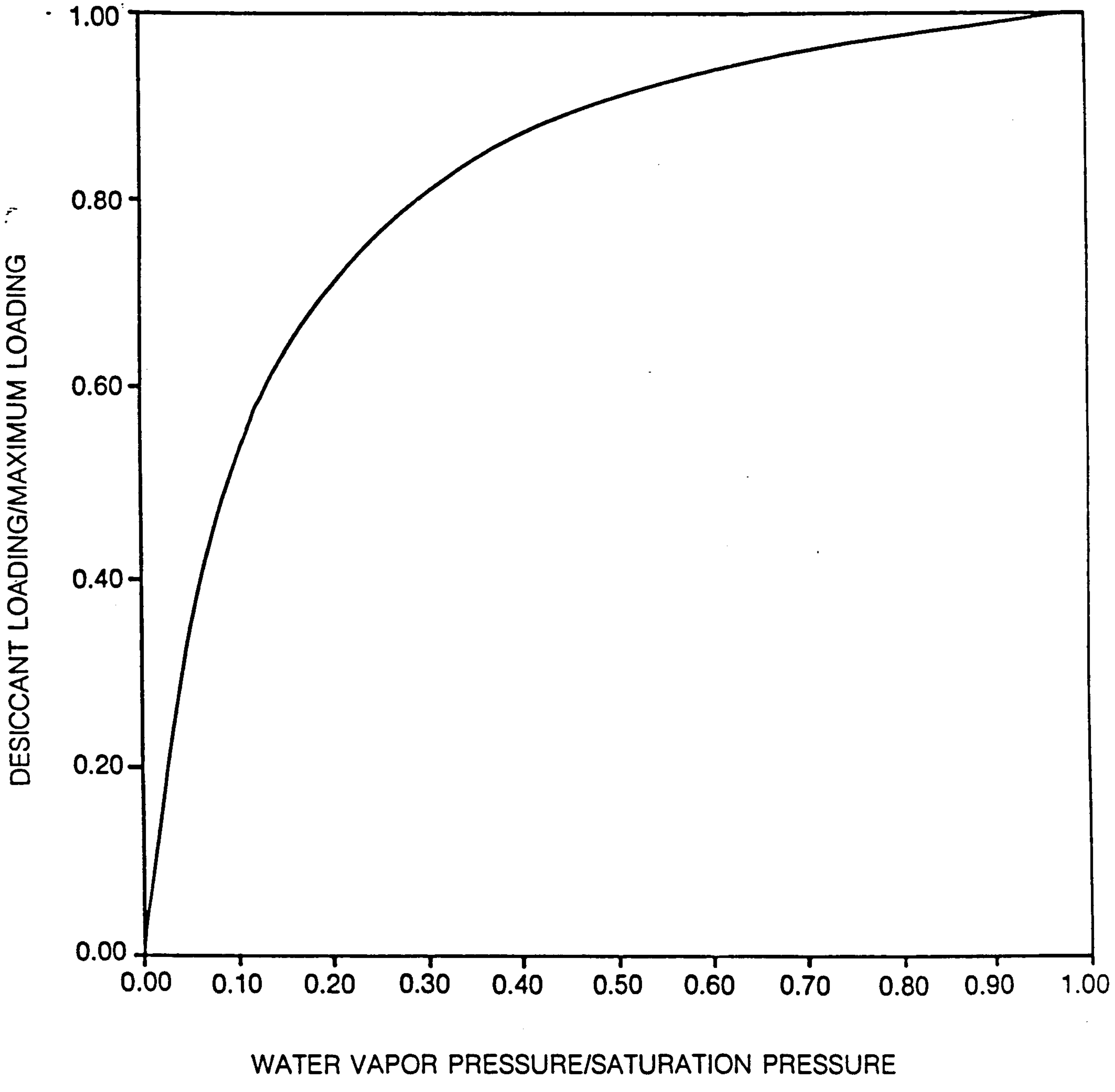


Fig. 4 Sorption properties of desired desiccant material

energy that are important. They are: 1) The maximum value of adsorption energy that occurs at zero loading, and 2) The rate that the adsorption energy decreases to become the heat of vaporization for water at higher loadings. The effect of high adsorption energies is generally bad from both a systems and a wavefront standpoint. From a systems standpoint, high adsorption energies decrease specific cooling capacity for a given regeneration temperature. From a wavefront standpoint, high adsorption energies decrease the effective value of the separation factor for low temperatures and increase it for high temperatures. This effect results in both the dehumidification and the regeneration wavefronts being broadened. When the adsorption energy quickly decreases with loading, the regeneration wavefront is most affected. When the adsorption energy slowly decreases with loading, the dehumidification wavefront is most affected. Generally speaking, the optimal value of the separation factor will decrease (become more Type I) as the adsorption energy increases. Even though the system performance is decreasing, the decrease can be minimized by altering the isotherm shape slightly. Also, it is best to have the adsorption energy decrease quickly as possible with loading. This will allow regeneration wavefronts to be steepened by increasing regeneration temperature.

The Type I isotherm depicted in Figure 4 is not the only possible isotherm shape that could conceivably produce the sharp wavefronts needed. Desiccant cooling equipment will rarely experience relative humidities over 60%, and when they do, the exposure may be for a short period of time where the loss in system efficiency due to an inappropriate isotherm shape would not seriously affect the seasonal performance. This could open the possibility of a Type II or Type IV isotherm being appropriate as long as the second derivative of the isotherm is sufficiently negative over the range of relative humidities most encountered by the cooling system. The word sufficiently should be emphasized here. The dehumidification wavefronts will quickly broaden if not. It can be definitely stated that Type III, Type V and Linear isotherms will not satisfy the design conditions imposed.

MAXIMUM UPTAKE

This desiccant property has historically been considered one of the more important ones. This was probably a carry-over from the large, static packed column applications where differences in the maximum uptake of the desiccant material could save hundreds or thousands of dollars in material costs. The amount of desiccant in a rotating system is negligible by comparison. The differences in wave speed (or breakthrough time) due to differences in the uptake by the desiccant are handled by merely changing the rotational speed of the desiccant matrix. There are absolutely no inherent advantages in the concentration wavefront behavior associated with the slower wave speeds of desiccants with large maximum uptakes. There can be a problem, however, with the thermal wavefront associated with the heat capacity of the system. For a given amount of heat capacity, the speed and magnitude of the first thermal wavefront will be almost independent of the maximum uptake of the desiccant. However, the speed and magnitude of the main concentration wave will be strongly affected by the maximum uptake of the desiccant. The result is that the outlet air conditions associated with the first wave are a larger fraction of the total cycle outlet air conditions as the maximum uptake of the desiccant decreases. This causes the average outlet air conditions of the cycle to be further removed from the MZ conditions with the concommitant reduction in system performance. Merely increasing the amount of desiccant per unit volume of matrix is not a solution because the amount of heat capacity will also increase which will slow the first wave keeping it's percentage of the total cycle time nearly the same.

Computer simulations have shown that the degradation in cooling system performance associated with the heat capacity of bulk desiccant materials (about 0.2 BTU/lbm deg F) and a maximum uptake of 0.4 lbm water/lbm desiccant, is very small. However, for the same uptake, increasing the heat capacity to 2 BTU/lbm deg F causes a severe performance loss. Not all of the heat capacity in a desiccant matrix is associated with the desiccant material itself. Many designs require a support structure to hold the desiccant in place. The heat capacity of this support structure adds to the heat capacity of the desiccant to result in sometimes very large overall heat capacities compared to that of the bulk desiccant material alone. It is possible therefore, that a dehumidifier matrix made up entirely of self-supporting desiccant material with a relatively low maximum uptake, would yield higher system performance than a desiccant with a relatively large uptake that required a significant supporting structure for deployment.

CHEMICAL AND PHYSICAL STABILITY

It is intuitively obvious (to coin the hated textbook phrase) that chemical and physical stability are properties that any desiccant should possess. However, there are two important aspects to this stability that deserve special attention. They are: 1) The ability to withstand high regeneration temperatures, and 2) The ability to be reliably regenerated by ambient-source air.

Many desiccants rely on molecular surface structures that are not stable at high temperatures. The requirements for desiccant cooling cycles dictate that the desiccant sorption properties should be stable at temperatures between 300 and 400 deg F.

Some desiccants are "fouled" by the presense of certain molecules that have a greater affinity for the active sites than water vapor. Others undergo a change in chemical composition in the presense of certain compounds that changes their water sorption characteristics. There are contaminants and pollutants in the air used for processing as well as byproducts from the combustion of natural gas (oxides of carbon, nitrogen, sulfur, particulates, etc.). Exposure to these substances must not inhibit or change the water sorption characteristics of the desiccant material.

SHORTFALLS IN CURRENT DESICCANT MATERIALS AND COMPONENTS

We do not believe that any of the commercially available desiccant materials were developed for the specific purpose of providing space cooling. In most present-day applications, the necessity of achieving efficient regeneration as well as deep drying of the air has not been a consideration. The requirement of attaining the very sharp adsorption wavefronts associated with the molecular sieves along with the more efficient regeneration characteristics of the silica gels is what makes this application truly unique commercially. It is not surprising therefore, that none of the commercially available desiccants match the properties needed. The isotherms of various commercial and laboratory-developed desiccants are shown in Figure 5. The trend is to see linear or nearly linear (Type II) isotherms or extreme Type I. Also, the extreme Type I isotherms most always are associated with very high adsorption energies. This is because the Type I behavior is often obtained by ionic dopants. These dopants form a limited number of very favorable high energy adsorption sites. When these sites are filled, the less energetic parent material sites are filled with a concommitant change in the isotherm shape.

A myriad of other water sorbing materials exist and have been identified in the literature. To date, over 150 different materials have been identified as potential desiccants. Unfortunately, most of the properties necessary to properly evaluate these materials are lacking. Only the commercially available desiccant materials are understood well enough to make meaningful judgements.

RESEARCH ISSUES AND GOALS

The goals of the desired research are as follows:

A desiccant material is needed:

1) Having an isotherm with a separation factor of between 0.1 and 0.07.

2) With a adsorption energy equal to the heat of vaporization for water.

3) With an internal mass diffusivity of not less than 10^{-8} ft^2/sec.

4) With the ability to be regenerated for a minimum of ten

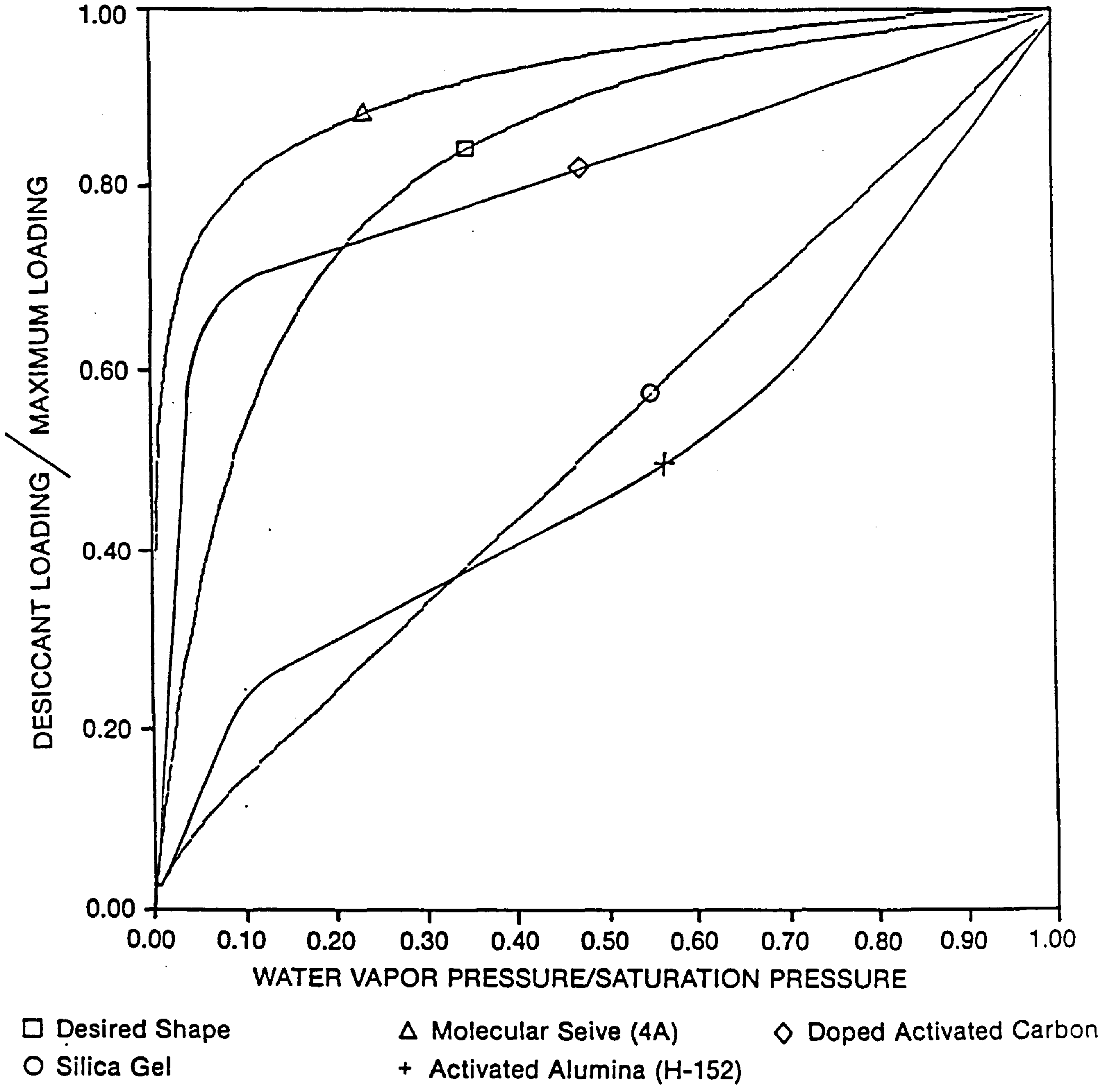


Fig. 5 Sorption properties of various desiccants

years continuous operation.

5) That can be fabricated into a honeycomb structure about 10 inches long with channel sizes of 0.06 to 0.04 inches consisting wholly of desiccant material.

Some possible questions associated with the desired material are:

1) Is it possible to achieve a large rate of uptake at low relative humidities that gradually decreases as loading increases, with a low adsorption energy? It is certainly thermodynamically possible, but may require fresh thinking about Type I isotherms to make it physically possible.

2) Is it possible to modify existing materials to achieve the properties desired? Could variations in process control, material purity or composition of existing desiccants be successful?

3) Could mixtures of existing desiccants be the solution? The patent literature (mostly foreign) contains many references to desiccant mixtures based on salts mixed with common adsorbents.

4) Are there any new fabrication techniques that would allow bulk-form desiccants to be efficiently integrated into a honeycomb matrix? Could this technique be applied to any material that exists in powder or particle form or is it restricted to a particular desiccant or process? Both cases would be applicable to this solicitation.

5) Are there any entirely new classes of materials capable of achieving the properties desired? For example, some polymers are known for their absorption behavior. Could the problems of swelling and unfavorable isotherm shape be overcome? Any material that could be made self-supporting would certainly be amenable to configuring into the desired honeycomb configuration.

POLYMERS: OPTIONS AS ADVANCED DESICCANT MATERIALS*

A. W. Czanderna and T. M. Thomas
Materials Research Branch
Solar Energy Research Institute
Golden, CO 80401

ABSTRACT

Background information will be presented that addresses the key issue: why consider using polymers for the desiccant material in desiccant cooling systems (DCS)? Of the experimental methods for measuring water vapor sorption by desiccants, a quartz crystal microbalance (QCM) for measuring sorption isotherms, rates, permeation rates, and cyclic stability has been chosen. Background information is given about the QCM, including the quartz crystal resonator itself, the necessary vacuum system and auxiliary apparatus, and the advantages and limitations of a QCM. The apparatus assembled and placed into operation at SERI during calendar 1985 is briefly described. The criteria for narrowing the potential candidates as advanced desiccant materials for the initial studies are given. A list of principal candidates of 14 materials identified based on the criteria and available data in the literature are then given. Finally, the potential advantages and possible limitations of using polymer materials as the desiccant in DCS are listed. More details about the experimental method and polymers for desiccants are given in a recent SERI report (reference 6).

---

* Work Supported by USDOE under Contract No. DE-AC02-83CH10093

1.0 INTRODUCTION

1.1 Why Polymers?. This paper is concerned with discussing the feasibility of polymeric materials as candidates for use in regenerative desiccant cooling systems (DCS). By considering a polymeric desiccant as a component in a DCS, we note that:

- o polymers have the potential for sorbing water from 5 to 2000% of their own weight,
- o polymers have the potential for being readily fabricated into shapes required for DCS, e.g., a honeycomb structure 10 x 0.04 x 0.06",
- o polymers have the potential of being modified so sorption isotherms of the desired shape and heats of adsorption of about 10.8 kcal/mol are both obtained,
- o polymer structures have the potential of being fabricated to provide rapid diffusivities of water vapor through the material,
- o polymers have the potential for being regenerated by using heated air for the thermal desorption of sorbed water,
- o polymers have the potential for maintaining long-term stability through thousands of sorption-desorption cycles, and
- o commercially available polymers are available at less than $1.00/lb, or comparable to the cost of commercial grade silica gel.

Accordingly, polymeric materials could not only serve as both the desiccant and support structure in a desiccant wheel, but also could be replaced easily and inexpensively if their water sorption capacity degrades for any reason. The primary research needs are to develop and to characterize the water sorption performance of polymeric materials that could serve in an economically competitive DCS.

1.2 Desiccant Cooling Systems. We are concerned with the possibility of using solid polymeric materials as desiccants in desiccant cooling systems (DCS) that process water vapor in an atmosphere to produce cooling. Systems with solid desiccants presently used in industrial air-drying applications and DCS have been described in other parts of these proceedings. For DCS, a desiccant-laden wheel is typically used in which air may flow in the axial direction only. The solid desiccant (state-of-the-art; silica gel) is mounted onto a wheel and the air to be dried flows through one side of the wheel, while the desiccant on the other side of the wheel is being dried by an externally heated air stream. These two air streams must be kept physically separated to maintain the distinctly separate functions of air dehumidification and desiccant regeneration. The historical development that has led to the present commercial systems has been summarized [1].

The commercial systems are primarily intended for dehumidifying air and do not produce a significant net cooling. There are important differences in design philosophy between regenerative desiccant cooling systems and commercial desiccant dehumidifiers. Some of these differences have also been described in these proceedings. Most importantly, it is apparent from thermodynamic analyses of recirculation cycles that the adsorption characteristics of the desiccant can have a large influence on the cooling capacity and COP of the cycle [1].

Silica gel is presently considered to be the most important candidate material for DCS. Extensive optimization studies of the engineering design of DCS units have been carried out using silica gel. Models in systems analysis studies for estimating the achievable thermal and electrical coefficient of performance (COP) also utilize the properties of silica gel, since they have been documented extensively over several decades [2]. Since the water sorption properties of the desiccant are crucial to the performance of any commercial system, we initiated a desiccant materials research effort to search for and measure the properties of materials that could become better desiccants than silica gel.

1.3 Purpose of Advanced Desiccant Materials Research. One purpose for performing long-range research on materials is to secure an understanding of the behavior of low-cost, high performance technological materials, with a goal of extending the lifetime of these materials or to identify new materials that will offer new options for components used in operating systems. The cost-effective deployment of DCS is currently limited by the sorption performance, durability, and life cycle cost of the inorganic materials used. Long-term materials development is needed which focuses on improving the properties of advanced desiccant materials [3] and on identifying the reasons for degradation of their sorption performance. The purpose of advanced desiccant materials research and development (R and D) is to provide new materials options for DCS. The ultimate purpose is to identify materials with optimal performance in the temperature range used in a DCS in which the sorption activity of the desiccant can be routinely regenerated with an appropriate energy source. The performance of these materials can be ranked as compared to the performance of silica gel, the industrial standard. The life-cycle cost (i.e., initial cost, performance, and durability) has a direct relationship on the cost-effective deployment of any regenerative DCS. Identification of materials with extended lifetimes, enhanced reliability, and sustained desirable sorption properties for use in DCS, can be anticipated from a sustained R and D effort.

1.4 Relationship of Desiccant Materials to Desiccant Cooling Systems. Desiccant materials must have a favorable performance/cost ratio, provide satisfactory performance, and have cost-effective lifetimes; widely available materials need to be certified, modified, or in some cases, developed specifically for use in DCS. Some of the important parameters for the water solid-desiccant-material system include the isotherm shape, sorption capacity, the heat of sorption, the rate of sorption at or near the desiccant bed temperature (~ 25 to 55°C), the rate of desorption at an elevated temperature, the physical and chemical stability of the desiccant, and the cyclic repeatability of the sorption amount and rates. Some important conclusions and recommendations on these parameters were reached in the recent modeling study [4], but the possibility of identifying new materials with optimal properties was not considered. The problems associated with silica gel and zeolites (e.g., isotherm shape, large heat of adsorption, high regeneration temperatures, and hysteresis on cycling) restrict their potential for use in DCS [1,4]. Therefore, a search for an advanced desiccant material is a natural research need for enhancing the potential of deploying cost-effective DCSs.

1.5 Background for Studies of Advanced Desiccant Materials. The arguments for desiccating water from air in an open-cycle DCS are well known [1]. The sorption of water vapor by the desiccant is given by the reaction scheme

$$D(s) + H_2O(v) \xrightarrow{1} D{\cdot}H_2O_{surface} \tag{1}$$

$$D(s) + D{\cdot}H_2O_{surface} \xrightarrow{2} D{\cdot}H_2O_{internal} + D(s) \tag{2}$$

where D(s) is a solid desiccant adsorption site. Water vapor adsorbs onto the surface (step 1) and permeates into the solid by several possible diffusion processes (step 2) to provide the total sorption. Empty surface sites are filled by further adsorption (step 1). Desorption of water vapor from the desiccant occurs by the reverse sequence

$$D{\cdot}H_2O_{surface} \xrightarrow{3} D(s) + H_2O(v) \tag{3}$$

and $$D(s)_{surface} + D{\cdot}H_2O_{internal} \xrightarrow{4} D{\cdot}H_2O_{surface} + D(s)_{internal} \tag{4}$$

where empty surface sites are generated (Eq. 3) by desorption and water sorbed in the material must first occupy a surface site (Eq. 4) prior to desorption via Eq. 3. While the details of the molecular processes are much more complex than illustrated by Eqs. 1-4, they illustrate that the sorption capacity is gained by at least two kinetic steps involving the rate of adsorption (Eq. 1) and the rate of permeation into the solid (Eq. 2). Equations 3 and 4 illustrate the reverse steps of desorption and diffusion. The diffusion (permeation) steps provide the "solid-side" resistance in the sorption-regeneration sequence. The principal limitations are related to finding a desiccant material with the required combination of isotherm shape, sorption capacity, sorption heat, sorption rates at the bed T, evolution rate at a minimally elevated temperature, cyclic stability of the sorption properties, and stability of the desiccant itself. These parameters are connected with the energetics and kinetics associated with gas-solid interactions.

From sorption isotherms, the binding energy of the water adsorbate to the solid can be determined. The rate of diffusion through the porous material can be calculated and, in principle, the rate of evolution from the solid can be determined and compared with experimental results. The latter must be taken on materials that at least approach the configurations used for the calculations. Practically, the sorption rate processes will be reduced by pore volume diffusion and intraparticle diffusion in a packed bed of particles. Although some sorption data are available in the literature [5] for candidate desiccant adsorbent materials, these data have not been analytically scrutinized to determine the suitability of new materials in advanced desiccant applications.

Recent articles show both experimental activity in water-vapor adsorbent systems and analysis of the associated phenomena [6]. In most of these studies, the conventional volumetric, SEM, and electron microscopic techniques used provide only a limited understanding of fundamental processes. Surface compositional analyses of solids have not been correlated with changes in sorption behavior. Bonding information available from infrared spectroscopy has been coupled with adsorbed amounts in only a few papers. The quartz crystal microbalance (QCM) has not been used extensively to obtain adsorption equilibrium and rates [6]. We plan to exploit the great potential of the QCM for both screening and evaluating advanced polymeric desiccant materials.

2.0 EXPERIMENTAL MEASUREMENT OF WATER VAPOR SORPTION BY DESICCANTS

The experimental apparatus for measuring the water vapor sorption by a solid is conceptually quite simple. An arrangement is needed where a solid can be surrounded by pure water vapor (or a partial pressure of water in the ambient atmosphere) and the number of water molecules that bond to the solid can be measured. A vacuum system can be used to control the gas (vapor) pressure over the solid. There are three basic methods for measuring directly the rate of gas adsorption and the equilibrium amount of vapor adsorbed by a solid. These are the gravimetric, volumetric, and radiotracer methods. The last is excluded because it would require tritiated water and offers no obvious advantages over gravimetric techniques. Gravimetric techniques have many advantages over volumetric techniques [5,7], especially for water vapor sorption studies. Water is one of the most difficult vapors to work with for analysis when using vacuum systems, but the determination of adsorption isotherms and the rate of adsorption and desorption of water can be carried out routinely with gravimetric techniques.

Of the available gravimetric techniques, the beam balance and the quartz crystal microbalance (QCM) are the two best choices when the advantages and limitations of all microbalances are considered [7]. We have chosen to use a QCM for our initial studies because of the commercial availability of QCMs and their ease of use for studying the sorption of water by materials at or near 25°C [8]. The experimental system, which is shown schematically in Fig. 1, consists of a vacuum system, a residual gas analyzer, a QCM, and a temperature bath for the QCM. These components and their operation have been described in detail [6].

If a desiccant material with an overall density of 1.4 g/cm^3 is placed on a quartz crystal, then a desiccant film 0.1 µm thick would have a mass of 14 µg, so the limit of detectability of mass change would be 0.1%/Hz. [100% x (14 ng/Hz)/14 µg], where ± 1 Hz can be routinely monitored with commercially available oscillator circuits. For thicker desiccant films, the percent mass sensitivity is even smaller. Since the interest is for desiccants with percent mass gains in water of over 5%, the QCM sensitivity is more than sufficient, even if an active quartz crystal area of only 0.25 cm^2 is utilized. The mass measuring range of the QCM will permit studying polymer thicknesses that vary by over 100 times, so permeation rates can be deduced.

For operation in vacuum, the quartz crystal holder is an integral part of a vacuum system apparatus. The essential components of this holder include a 2.75-in. stainless steel Conflat^R vacuum flange with feedthroughs for a thermocouple, cooling water, and the electrical leads needed for operating the oscillator.

The advantages of the QCM for measuring the sorption of water vapor by advanced desiccant materials are as follows:

- The fundamental water sorption properties of desiccant materials can be studied in the absence of any engineering design restrictions.
- An appropriate mass sensitivity (14 ng/Hz cm^2) is obtained using commercially available equipment. For example, a 100 nm-thick polymer film with a mass of 14 ng on a 1-cm^2 QCM results in detectable mass changes of 0.1%, or a single monolayer of an adsorbed gas is detectable for contaminant studies.
- The response time is fast; i.e., 2 s per point.
- It is UHV compatible, and can be baked to 450°C.

- It can be used in pure water vapor at pressures comparable to the relative humidity in actual desiccant systems. Thus, the sorption kinetics of the desiccant can be studied in the absence of any gas-side diffusion effects.

- Once constructed, it is simple to use and is not sensitive to shock or vibration.

- An accuracy of < 2 % can be achieved.

- It can be easily adapted for simultaneous use with IR or surface analysis equipment.

Some of the principal limitations of the QCM are:

- Only one temperature (or a narrow temperature range) can be studied with a given crystal at the maximum sensitivity.

- Samples must be coated onto the quartz crystal and must adhere to it.

- Liquid samples are not easily studied.

- Condensation of unknown gases on the sample or volatile materials in the sample may cause problems in securing the actual water sorption.

3.0 POTENTIAL POLYMERIC ADVANCED DESICCANT MATERIALS FROM ANALYSIS OF THE LITERATURE

About 300 articles were identified in a literature search for potential candidate advanced desiccant materials. Since sorption capacity is crucially important for use as a desiccant, the computerized literature search was carried out for articles containing the words water vapor and polymers. About half of the literature citations were for articles in Japanese, Russian, or Eastern Bloc languages; these were not translated based on our best judgments from the English abstracts. This decision was based principally on making the most prudent use of the funds available for the work.

3.1 Criteria used for Prioritizing Potential Candidate Materials. There are two sets of parameters that could be applied for ranking the identified materials in a list of potential candidate advanced desiccants. The first set is as follows: (1) Cost, both initial and life cycle; (2) Performance; (3) Durability, lifetime, and how performance parameters depend on temperature; (4) Availability; (5) Processibility; (6) Modifiability; (7) Stability at temperatures below 100^{o}C; and (8) Specific heat. For polymeric materials, the most important criterion is (2). As a first approximation, polymers should have relatively uniformly good prospects for (1), (4), (5), (6), and (7), and marginal values for (8). It was not expected that durability of polymers used as desiccants, i.e., (3), would have been evaluated, and this expectation was correct.

The second set of parameters considered are performance related, and were divided into three categories; i.e., primary, secondary, and other criteria. These criteria were ranked and reviewed by several members of the SERI desiccant team before analyzing the available literature references. The criteria, which are more detailed than previous listings [1] and were generated by SERI researchers [9], are as follows [6]:

- Primary Criteria ($X = P/P_0$, w = weight gain)
  - Sorption capacity; isotherm data on capacity and shape, where $dw/dx > 0.5$ for $0.05 < x < 0.8$
  - Permeation data, rate, or diffusivity where a diffusivity $> 10^{-9}$ m^2/s is desired and surface diffusion versus Knudsen diffusion needs to be considered. Pore size, pore size distribution, and their influence on capacity and sorption kinetics are included here.
- Secondary Criteria
  - Isotherm data yielding heats of adsorption and that indicate degree of hysteresis, if any
  - Specific heat, where $(dw/dx)/C_p > 0.5$ g K/J is desired from 0.05 to 0.8
  - Desorption kinetics at $T < 100^{\circ}C$ for regeneration.
- Other Criteria (some may be primary and others secondary) include

  cost, processibility, mechanical properties, commercial availability, stable at $T < 100^{\circ}C$, effects of impurities on isotherm data, and effect of cycling to an "elevated" T on the data for the primary and secondary criteria; i.e., chemical stability.

Since the goal of our research is to measure the performance parameters of the material itself, the obvious engineering-related parameters (e.g., bed geometry, material form, desiccant size and shape, packing density, pressure drops, and heat and mass transfer) were not considered.

3.2 Potential Advanced Desiccant Materials. A prioritized listing of potential polymeric advanced desiccant materials is given in Table 1. This listing is based on the relatively sparse information available in the open literature. Further refinements should be possible from product information available from various manufacturers. The only primary information that we were able to use were the sorption capacity and, in most cases, the isotherm shape.

There were no permeation or diffusivity data given when isotherm data were given. The permeation coefficients available were typically in the 10^{-10} to 10^{-12} m^2/s range, or slower than desired in the primary criteria. The permeation rates in polymers depend strongly on the properties, but no evidence was found for any systematic synthesis effort to improve the diffusivities of water in polymers. In fact, synthesis efforts have been concentrated on retarding diffusivities for the obvious applications in corrosion protection.

Finally, no important data published about the kinetics of water adsorption or desorption were found nor were any studies of the cyclic reproducibility for either the capacity or sorption rates reported. The only type of comments found were "adsorption is fast" or the isotherms were reproducible after "several cycles of measurement." Some polymers swell during water vapor adsorption, and this could be detrimental to pressure drops in a working system.

Further refinement of the ranked lists in Table 1 will be possible when other factors can be considered. In some cases, data should be available from the manufacturers of the polymers, but in other cases, the properties will have to be measured. In addition, the list in Table 1 is being expanded by direct contact with additional manufacturers, suppliers, or investigators with laboratory formulations. The search for additional hygroscopic polymers or inorganic solids will be an ongoing process while the QCM equipment is being used for the preliminary screening of the current lists of candidates in Table 1.

4.0 KEY RESEARCH ISSUES FOR POLYMERIC DESICCANTS

The key research questions in seeking candidate polymers for DCS are:

- Will any commercially available polymer have the properties required for use in a DCS?
- If commercially available polymers are not suitable, can laboratory-prepared polymers be synthesized with suitable sorption, desorption, and stability properties?
- Do impurities in a gas-fired air stream result in loss in sorption performance of candidate polymers?
- Are there other degradation processes that cause a candidate polymer to lose sorption performance? and
- Can the polymeric material be modified to improve performance properties such as isotherm shape, heat of adsorption, regeneration temperature, cyclic stability, and diffusivity?
- Do the performance properties depend on the sample size and configuration, e.g., 0.1 mg (QCM), 100 mg (beam microbalance), and SERI test-loop scale and configuration.

In the last two years, we have initiated work that provides a rapid QCM method for evaluating the performance properties of advanced desiccant materials with an emphasis on polymers. These include measuring sorption isotherms, sorption and desorption kinetics, sorption capacity, permeability, and the cyclic stability of advanced desiccant materials. In the future, polymers will be studied that are either commercially available or synthesized specifically to serve as a desiccant. Those polymers with the best properties will be subjected to simulated use conditions and cause for any performance losses will be identified.

5.0 REFERENCES

1. Collier, R. K., R. S. Barlow, F. H. Arnold, "An Overview of Open-Cycle Desiccant Cooling Systems and Materials," J. Solar Energy Engr., 104 (1982) 28.

2. Parent, Y., Davison Division, W. R. Grace, is an excellent contact for literature about silica gel and its properties.

3. Shelpuk, B., Proceedings of a Desiccant Cooling Conference of 11/16/77, SERI-22, Golden, CO: Solar Energy Research Institute, April 1978.

4. Collier, K., Principles of Open Cycle Desiccant Cooling System Operation and Design, TE-4315-169-85, Phoenix, AZ: Enerscope, Inc. (Draft), October 3, 1985.

5. Gregg, S. J. and K. S. W. Sing, Adsorption, Surface Area, and Porosity, Academic Press, NY, 1982.

6. Czanderna, A. W. and T. M. Thomas, Advanced Desiccant Materials Research, SERI/PR-255-2887, June, 1986, Solar Energy Research Institute, Golden, CO (Available NTIS).

7. Czanderna, A. W. and S. P. Wolsky (eds.), Microweighing in Vacuum and Controlled Environments, Elsevier, Amsterdam, 1980.

8. Lu, C., and A. W. Czanderna (eds.), Applications of Piezoelectric Quartz Crystal Microbalances, Elsevier, Amsterdam, 1984.

9. Maclaine-Cross, I., A. a. Pesaran, T. Penney, A. W. Czanderna, L. Flowers, and T. M. Thomas, as given in ref. [6].

Table I. Leading Organic Polymeric Candidates for an Advanced Material for Use in Desiccant Cooling Systems

Material	Sorption Capacity (wt. % H_2O at $x = 0.8$)	Isotherm Shape	Other Factors
Polystyrene sulphocationates	30-55	Linear to Type 2	Swelling of 1.3 to 5 times
Cationites based on methacrylic acid	18-55	Nearly ideal[a]	Swelling of 1.2 to 3 times
Aminoalkylalkoxysilanes	10-90	Type 2	Supported by glass fibers; most uptake is between x = 0.6 and and 0.9
Copolyoxamides	8-68	Type 2	Made for membrane applications
Wool plus 8% Calgon Polymer 261	28	Type 2	Other modified wools also studied
Viscose Rayon	18	Nearly Ideal[a]	Sorption studies at 35°C so uptake may be greater at 25°C; cyclic studies done
Triethylbenzl ammonium cationites	18-22	Nearly Ideal[a]	Swelling of 1.2 to 5 times
Perfluorosulfonic acid membranes (du Pont Nafion)	13	Nearly Ideal[a]	Influence of processing on isotherm shape and capacity studied; heat of adsorption determined
Cellophane	15	Nearly Ideal[a] to x = 0.8	Capacity increases to over 20% at x = 0.9
Vinylon	12	Linear to Type 3	Capacity ranges from 4% to 16% between x = 0.5 and 0.9
Polyurethanes, poly (ethylene oxide)	13	Type 3	Diffusivity is 4×10^{-7} cm^2/s and the sorption kinetics are fast; capacity range is 3% to 23% from x = 0.6 to 0.9
Styrene divinylbenzene copolymer	9-18	Nearly Ideal[a]	Influence of amino groups on sorption capacity was studied
Epoxy Resins	10	—	Sorption kinetics measured slow uptakes
Poly (cross-linked) methacrylates	10	Ideal[a]	Sorption at 35°C and larger uptakes from x = 0.8 to 0.9

[a] The ideal isotherm shape was taken to be linear for a plot of sorption capacity vs. x (relative pressure of water vapor). Most of the "nearly ideal" isotherms had slight curvatures of a type 2 isotherm with a small BET "c" value. These isotherms might be excellent fits to those modeled in the most recent systems study [11].

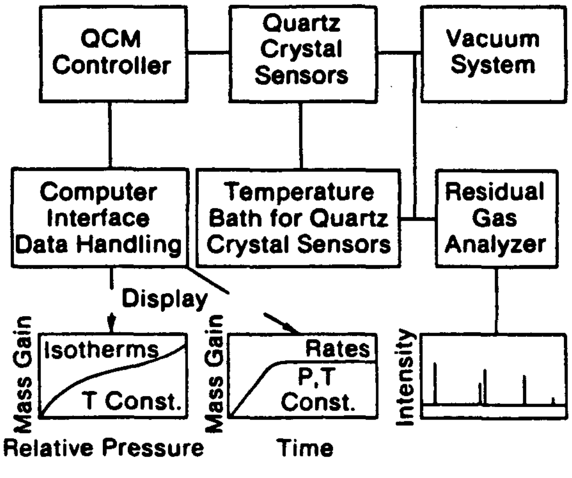


Figure 1. Block Diagram Showing the Principal Components of a Quartz Crystal Microbalance (QCM).

Other Papers Submitted

Southern California Gas Company Perspective on Gas Cooling

Steve Simons, Southern California Gas Company

Today, I would like to present to you the Southern California Gas Company's perspective on the market potential for gas cooling. We see desiccant cooling as one type of system to develop in pursuing this lucrative market.

Historically, SoCalGas was heavily involved in the development and promotion of gas cooling equipment. Throughout the 1960's and early 70's special programs to promote gas cooling were available to residential and commercial customers. Specifically, SoCal offered advertising campaigns and educational classes, reduced gas rates, assumed the responsibility for equipment warranties, and provided a field service/maintenance organization. SoCalGas was able to sell and install over 50,000 tons of cooling through promotion of these programs, primarily in the residential market.

However, in the early 70's two major factors caused SoCalGas to terminate its gas cooling programs. Gas cooling equipment became a liability with the company due to poor equipment reliability and the corresponding high maintenance costs SoCalGas absorbed in honoring warranty contracts. The second factor which hurt sales of gas fired air conditioning units was the regulation-induced gas supply shortages. Gas utilities were forced to eliminate "load building" marketing programs which included rate incentives for gas cooling. However, it is my understanding that SoCalGas would have discontinued these programs with or without the gas supply problem. It was evident during this period that manufacturers had little commitment in supplying the market with efficient, reliable, cost competitive equipment. Over the years minimal improvements in the technology or action has been taken by the manufacturers or our company to promote gas cooling. The result is obvious! SoCalGas's percent of market share has declined significantly. Estimates of the gas air conditioning market share are: residential - 2%, secondary schools - 13%, health care facilities - 11%, restaurants - 9%, commercial offices - 8% , and hotel/motels - 4%. These percentages are residual from the 1960's and continue to decline.

This situation offers gas utilities and manufacturers an excellent opportunity. With declining gas prices, stable gas supplies, and projected high electric rates, an opportunity to move back into this market exists. What is needed to take full advantage of the current situation? We need gas cooling equipment that is competitive with electric systems; equipment that is energy efficient and reliable. We need manufacturers that are dedicated to improving their product line and who will commit to an extensive distribution and service network.

What is SoCalGas's involvement and future plans? We are currently looking at several types of gas cooling systems, including absorption systems, gas fired heat pumps, and desiccant cooling systems.

We perceive desiccant systems to be directed towards residential and small to medium sized commercial applications up to 100 tons. SoCalGas will conduct laboratory tests on a 4 ton unit starting in June, 1986, and proceed with field tests of at least one unit during 1987. Field testing of a 25 ton desiccant unit was completed in late 1985 in a greenhouse flower application. By controlling temperature and humidity in the greenhouse, we attempted to decrease growing time, improve flower quality, and reduce the need for insecticide sprays. Test results were unimpressive, as were the design and operation of the desiccant system. Larger desiccant systems for the commercial market, such as Cargocaine's Super Aire System are being considered. Another application of interest is the integration of desiccant cooling with small packaged cogeneration.

SoCalGas is also considering several new marketing programs to assist in the promotion of gas cooling. New programs under consideration are builder incentives and advertising promotions. Minimal maintenance services for gas air conditioners might also be offered. Several promotional programs SoCalGas probably wouldn't offer are special gas rates, warranty work, and extensive service/maintenance programs.

In closing, let me say that SoCalGas is extremely interested in improving its position in the gas cooling market. However, we will only promote equipment that is efficient,reliable and offers customers a cost advantage over the electric competition. We do not want to become the major marketing agent for manufacturers. SoCalGas is willing to offer assistance to manufacturers, but manufacturers must consider establishing distribution, marketing, and service organizations.

GAS RESEARCH INSTITUTE DESICCANT COOLING PROGRAM

D. R. Kosar   Project Manager, Cooling Systems
Gas Research Institute

INTRODUCTION TO GAS COOLING

The issue of the future potential of gas cooling in general has become of intense interest not only to gas utilities and the pipelines and producers that serve them, but also to energy consumers and, especially, to the many commercial users that are experiencing rapidly mounting electricity costs. During the long period of falling or stable real electricity prices and looming gas shortages and gas price flyups, the market share of gas cooling equipment virtually disappeared. This period provided little incentive to pay the large installed cost premiums for gas equipment that failed to keep pace with the rapid advances in electric equipment. The recent dramatic turnaround in the competitive environment created by rising electricity prices and stable or falling gas prices has already opened a "window of opportunity" for some existing, improved gas cooling technologies. However, even more significant is the fact that this window need not close in 1990, but can stay open well beyond 2000 with advancements in gas cooling technologies.

The cooling (as well as heating) performance of gas and electric equipment is measured by a thermal coefficient of performance at standard conditions - COP for short. It is the useful cooling (or heating) capacity delivered, divided by the gas or electric energy input. A COP of 1.0 would be equivalent to 100% "efficiency." The reason COPs can be higher than 1.0 (or "efficiencies" greater than 100%) is that the utility energy input is not used directly but, instead, is used to "pump" heat from a lower to a higher temperature by means of a refrigerant compression or absorption cycle. Electric cooling equipment usually has COPs of 3.0 or higher, whereas existing gas equipment has cooling COPs of 1.0 or less. The new gas cooling technologies whose development GRI is supporting is narrowing this performance gas - they have COPs well above 1.0 and some might even reach more than 2.0.

However, generally, their installed cost is still higher than that of electric equipment. Fortunately, these performance and installed-cost differences will not put these new gas cooling technologies at a competitive disadvantage. At the high prevailing and projected ratios of electric to gas energy costs - about 4:1 today and no less than 3:1 over the longer range - the higher first costs and lower COPs of gas equipment are offset by lower utility costs. In fact, the differentials between electricity and gas costs will generate substantial savings over the life of the equipment for owner-users. This is not all - these savings will be large enough to pay back the installed cost differences in just a few years and sometimes in as little as one year! This is typical of all of GRI's investments in new or improved end-use technologies. GRI, in its R&D program, strives not only for substantially lower life cycle costs, but for quick payback of any first cost premium - normally 2 to 4 years. This is a more realistic test of commercial viability than competitive life cycle costs because definitive studies have shown that purchasers of space conditioning equipment and other capital goods that provide energy services will generally not accept simple payback times in excess of 5 years. It has been said that only the U.S. Government buys on the basis of the lowest projected life cycle costs.

BACKGROUND ON GAS COOLING

In the course of GRI's program planning process and in market-oriented studies conducted by GRI and other organizations, it has become apparent that gas cooling in general could provide enormous benefits to the gas industry and to residential and commercial energy users if a secure market foothold can be established by the end of the decade. The factors underlying this renewed interest in gas cooling include:

- The recent sharp drop in oil prices has resulted in a significant decline in gas prices and a move toward a more efficient and market-oriented gas industry. As a result, the competitive position of gas has significantly improved relative to electricity. Furthermore, gas prices are expected to remain relatively stable over the next several years.

- In addition to the decline in gas prices and the dramatic turnaround in the near-term supply situation, the long-term prospects for natural gas supply have also changed for the better in several significant ways. This is the result of the ongoing reassessment of the economics of recovery of the unquestionably huge resource base, in part due to the development of advanced recovery techniques that make formerly marginal supplies producible at market-clearing prices. Another important contributor to the greatly improved prospects for supply and price stability is the favorable outlook for rising Canadian imports at costs competitive with Lower-48 States gas prices until at least 2000. These positive fundamentals speak for adequate supply at only modest price increases well after the current surplus deliverability has been dissipated, and wellhead prices have again achieved equilibrium with true replacement costs.

- Growth in electricity demand through the balance of the 1980s and into the 1990s is expected to closely track the rate of growth in GNP. A realistic assessment of the likely schedules of completion and of probable deferrals and cancellations of currently planned utility plant additions indicate a high probability of substantial shortfalls in central generating capacity by the mid-nineties. GRI's own assessment has identified a potential 60 gigawatt deficiency by 2000. In response to any shortfalls, utility regulatory commissions can be expected to establish rate structures which will penalize peak load users in an attempt to curb load growth. This will act to maintain the window of opportunity for gas-fired cooling beyond the current circumstances.

- The majority of new construction is taking place in the South and Southwest where the substantial energy load is cooling. Many commercial buildings, regardless of location, require long cooling seasons. At present, less than 150,000 out of 2.4 million commercial buildings use gas cooling equipment.

- New equipment now available or under development offers superior performance and reliability over previous gas-fired systems marketed in the 1960s and '70s. At least two American and four Japanese manufacturers are currently marketing advanced gas cooling equipment (absorption chillers/heaters) for commercial buildings.

- At least 25 gas utilities now offer seasonal incentive rates for gas air conditioning and several offer rebates of $100/ton or more to builders, developers, owners or operators of commercial buildings that install gas cooling.

These trends illustrate the opportunity at hand and the potential benefits of an increase in gas cooling installations. For the gas industry, gas cooling offers a way to increase sales during the summer "valley" and offset the cost of providing gas service to new customers. Because of the extensive gas delivery system that is already in place, these advantages can be realized with little, if any, additional capital expense. For regulators and ratepayers, gas cooling can provide a cost-competitive option, particularly where electric rates and demand charges are high. By reducing electric consumption, gas cooling can also help to defer the construction of additional power generation plants and thus reduce upward pressure on electric rates.

To improve the efficiency, reliability, and cost-effectiveness of gas cooling equipment, GRI has put in place an ambitious R&D effort with the goal of introducing several advanced systems circa 1990. The planned expenditures for this effort have increased steadily over time. The total expenditure planned last year for the period 1986-90 was $155 million. The total expenditure now planned for the period 1987-91 has risen to $184 million, or an average of $37 million per year. This does not include the roughly $50 million of expected cofunding or coordinated funding by participating manufacturers. The technologies addressed by GRI's gas cooling program include engine driven heat pumps and chillers, absorption heat pumps and chillers, cogeneration package cooling systems, and desiccant cooling systems. A significant fraction of the 1986-1990 planned expenditures, approximately $23 million, is for desiccant cooling system R&D. Likewise, the planned expenditure for the 1987-1991 period has risen to over $26 million for desiccant cooling system R&D. The current 1986 year plan expenditures for the 1986-1990 time period are shown in Figure 1 for systems/components, materials, and analytical tools R&D for applications of desiccants to residential and commercial space conditioning.

SUMMARY OF GRI DESICCANT COOLING PROGRAM

Most cooling systems - whether gas or electric - actually provide two types of cooling: sensible cooling, or temperature reduction, and latent cooling, or dehumidification. In a desiccant-based cooling system, the latent cooling load is met by means of a desiccant, or drying agent, that removes moisture from the air. The desiccant is regenerated by the application of a heat source that releases the moisture, which is rejected to the outdoors as exhaust air. Sensible cooling can be incorporated by including standard electric cooling components with reduced capacity. To further reduce the amount of electric vapor compression air conditioning, a more advanced desiccant-based system incorporates the use of indirect evaporative cooling. It is also possible to completely eliminate the need for electric air conditioning by adding a stage in which the bone-dry, warm air produced by the desiccant is brought to comfort levels by evaporating water into it - the old swamp cooler idea that works quite well in hot, dry climates such as Arizona. The options for substituting gas-fired dehumidification for electric air conditioning are shown schematically in Figure 2.

An early result of GRI's R&D work on desiccant systems is SuperAire I, an integrated gas desiccant/electric air conditioning system introduced by Cargocaire Engineering Corporation in early 1984. First application of this system is targeted toward supermarkets. Forty one units have already been sold. SuperAire I provides the equivalent capacity of an 80-ton electric air conditioner by substituting 15 tons of desiccant and 60 tons of electric cooling. In a typical 30,000 square foot supermarket located in the humid Southeast, the desiccant-based system reduces the electrical load to the customer by 27% while increasing the gas consumption by 750 to 1,500 Mcf per year. The overall benefit to the supermarket owner is a 10% reduction in cooling and refrigeration costs, and greater comfort of shoppers, and enhanced appearance of frozen foods because of better humidity control. Because the system requires smaller ductwork than a conventional air conditioner, it can be installed in new stores at no cost premium. As a replacement unit in an existing store, energy savings result in only a two-year payback over a standard system. The system also qualifies for an investment tax credit, further increasing its advantages to the user.

GRI is now supporting the development of SuperAire II, an advanced system that incorporates high-efficiency gas-fired heat pipe heat exchangers and indirect evaporative cooling to further reduce electric air conditioning capacity. This system will use 50% less electricity than a comparable all-electric system and provide a net energy saving of 20% for air conditioning and refrigeration. Field testing is currently underway at an H. E. Butt Company Store in San Antonio, Texas, with a second unit at a Kroger Store in Murfreesboro, Tennessee. Other specialized commercial and institutional applications which can also economically benefit from the low moisture level setpoints and tight humidity control, such as electronics, food, and pharmaceutical production, are now under investigation by Cargocaire to broaden the applications for SuperAire.

To expand the manufacturer base and further broaden application for integrated desiccant systems, a Request for Proposal (RFP) was released in 1985 by GRI soliciting additional commercial and residential system development. The RFP identified commercial building ventilation air treatment and residential dehumidification as potential integrated system markets. One pending project, to be performed by the leading U.S. liquid desiccant equipment manufacturer, will develop a nominal 20 to 100 ton rooftop integrated (with electric air conditioning) system for low rise commercial buildings. A second selected project team, including a major absorption chiller manufacturer/marketer, will develop a greater than 100 ton central integrated (with absorption chiller) system for high rise commercial buildings. Research to adapt the integrated desiccant technology to residential applications is also being initiated this year as an integral part of the desiccant cooling program.

To eliminate the need for any electric cooling components, GRI is also sponsoring work with Thermo Electron Corporation on total desiccant cooling systems in sizes spanning 5 to 20 tons. The work is still in the laboratory prototype stages, but performance levels are extremely encouraging. Field tests of these advanced systems are scheduled for 1987 and beyond. Furthermore, a unique closed cycle arrangement for a cooling (and heat pump) device employing desiccants is under laboratory development with the Zeopower Company.

In support of these system/component development efforts, GRI is also sponsoring research on advanced desiccant materials for improved performance/lower cost components and systems and on analytical tools for more accurate system and building simulation. A RFP released earlier in 1986 is soliciting both basic and applied R&D of desiccation processes and materials. Continuation of work with the Florida Solar Energy Center on more realistic latent and sensible building load modeling and mechanical and desiccant cooling performance simulation is also planned to deliver a public domain computer program for researchers and A/Es in the late 80s.

CONCLUSION

This brief outline of GRI's efforts related to gas cooling technologies, and desiccant system development in particular, demonstrates our level of commitment to this vital and viable opportunity to provide benefits to the gas industry and to residential and commercial energy users. GRI not only has well-formulated strategies designed to address the particular characteristics of many building types and market sectors, but has developed an approach that ties together activities in numerous related efforts in order to gain the maximum value of research results and speed individual products toward the marketplace. A major element of this approach is managing the interactions among R&D contractors, equipment manufacturers, member companies, and other parties to stimulate technical achievement based on realistic cost and performance goals. With a continuation of our effort, GRI can and will make important contributions toward establishing desiccant cooling as a viable option during this tempting window of opportunity and to the creation of the technology base needed for growing long-term market penetration.

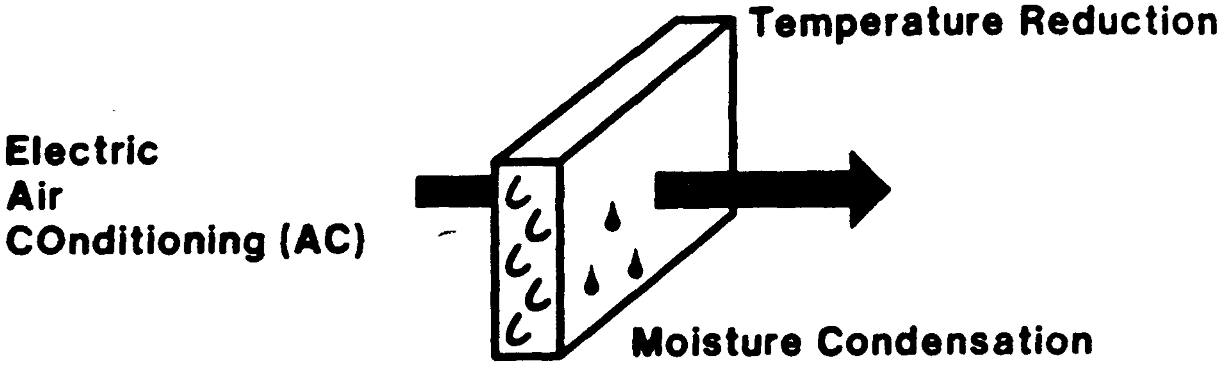
Figure 1

GRI DESICCANT DEVELOPMENT PROGRAM
1986 PLAN BUDGET ($000)

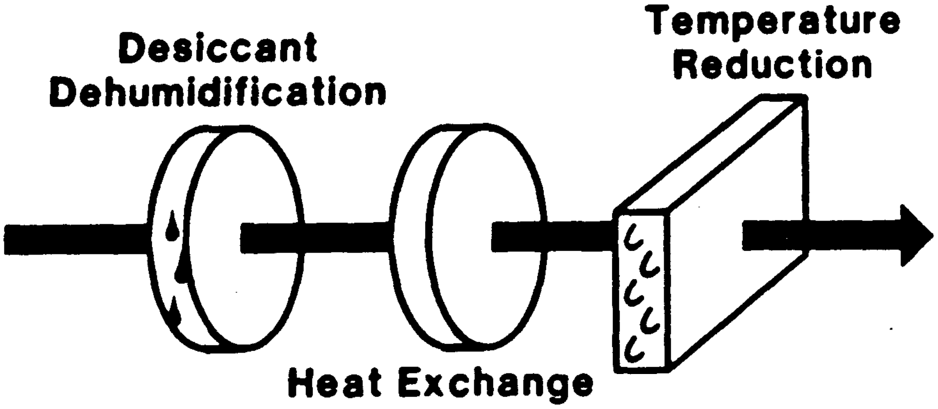
	'86	'87	'88	'89	'90
Total Cooling Systems/Components	800	1,400	1,800	2,400	3,100
Integrated Cooling Systems/Components	1,200	2,200	1,950	1,900	1,800
Materials	400	400	600	900	900
Analytical Tools	200	200	200	200	200
Total Budget	2,600	4,200	4,550	5,400	6,000

Figure 2

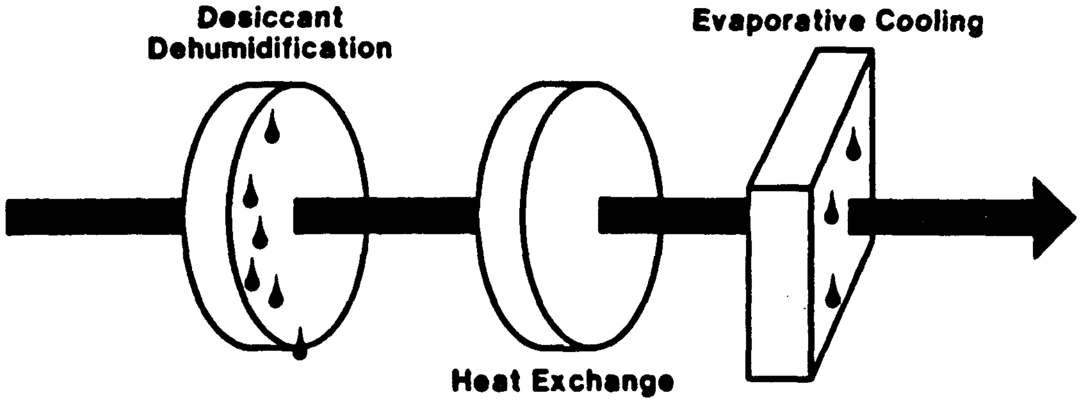
EVOLUTIONARY DESICCANT COOLING SYSTEM DEVELOPMENT



Step 1: Downsize Electric AC and Add Gas Dehumidifier



Step 2: Remove Electric AC, Upsize Gas Dehumidifier and Add Evaporative Cooler



EPRI/GRI/TVA Workshop on:
Desiccant Cooling & Dehumidification
Opportunities for Buildings
June 10-12, 1986, Chattanooga, TN

Glenn T. Chinery
Tennessee Valley Authority
1S 70A Signal Place
Chattanooga, TN 37402

ELECTRICAL UTILITY PERSPECTIVE
-on-
DESICCANT COOLING & DEHUMIDIFICATION

ABSTRACT

In the last several years electric utilities, including TVA, have discovered that when "new capacity" is needed to augment their power system's generation base, conservation strategies, including load management, are a less expensive option to implement than the building of new power plants. When properly designed and controlled, desiccant air conditioning systems have operating characteristics which make them potentially ideal devices for peak shaving, peak shifting and strategic conservation. Since penetrations of desiccant systems in large enough numbers to impact electric utilities' demand loads are not expected until perhaps the mid-1990's, the present overcapacity situation faced by many utilities will probably not be adversely affected by the development of a large desiccant market share. By the time such a market share penetration is likely to occur, utilities may be able to defer or outright cancel the construction of expensive new capacity which would have had to have been built to meet the air conditioning/heating load met by desiccant systems. Consumers may additionally benefit by the potentially high SEER's and good humidity control such systems can deliver compared to conventional vapor compression machines.

INTRODUCTION

According to TVA's recently published Energy Services Plan (ref. 1), TVA is "---strongly committed to energy conservation and end-use management programs in order to help delay expensive new power plant construction and to help our customers use energy wisely." In the same document it was stated that "(TVA) will improve the economic use of --- existing generating facilities by reducing energy use during the peak periods and by selling power in offpeak periods"---and ---"will demonstrate and promote new techniques and technologies that enable us to serve our customers better and more economically and enhance environmental qualities." This approach set the stage for TVA's interest and involvement in desiccant air conditioning activities. Widespread penetration of desiccant systems in the coming 20 years may assist TVA (and other utilities) in meeting these types of goals.

POTENTIAL BENEFITS TO UTILITIES AND THEIR CUSTOMERS

In meeting their load electrical utilities experience the daily roller coaster phenomenon of peak demand during certain times of the day and paucity of demand at other times. In the summer and winter, in particular, the differences between the "peaks" and "valleys" become extreme. This is well illustrated by the graphs in figures 1 & 2, which show the actual load curves experienced by TVA for a typical summer day and a typical winter day in 1984, respectively. A majority of the power system demand profile is made up of residential and C&I loads, and a significant portion of the residential and C&I loads are composed of space conditioning. As an example, residential air conditioners are responsible for about 40 percent of the residential summer peak.

TVA's generation mix is composed of a variety of generation sources ranging from least expensive baseload generators (nuclear, large coal plants and some hydro), through more expensive intermediate generators (coal plants), to most expensive peak sources (combustion turbines, purchased power, additional hydro). With minor exceptions, the plan followed in committing generation units to meet the load is in order from the least expensive to most expensive, with respect to operating and maintenance cost. Accordingly the marginal cost of producing power is highest during the peak hours and lowest during the offpeak hours. Table 1 shows TVA's actual 1984 cost of producing power as a function of generation source.

It would be financially advantageous to the power system (and eventually to the rate payers) to reduce or "shave" electrical energy use during expensive peak periods, since this would reduce the average cost of all power production. It would be additionally advantageous to "shift" some electrical energy use from peak periods to offpeak periods, since additional load during inexpensive offpeak periods would also serve to lower the average cost of all power production. Making up shifted demand during offpeak periods is particularly beneficial for utilities with overcapacity, which is the present situation with TVA.

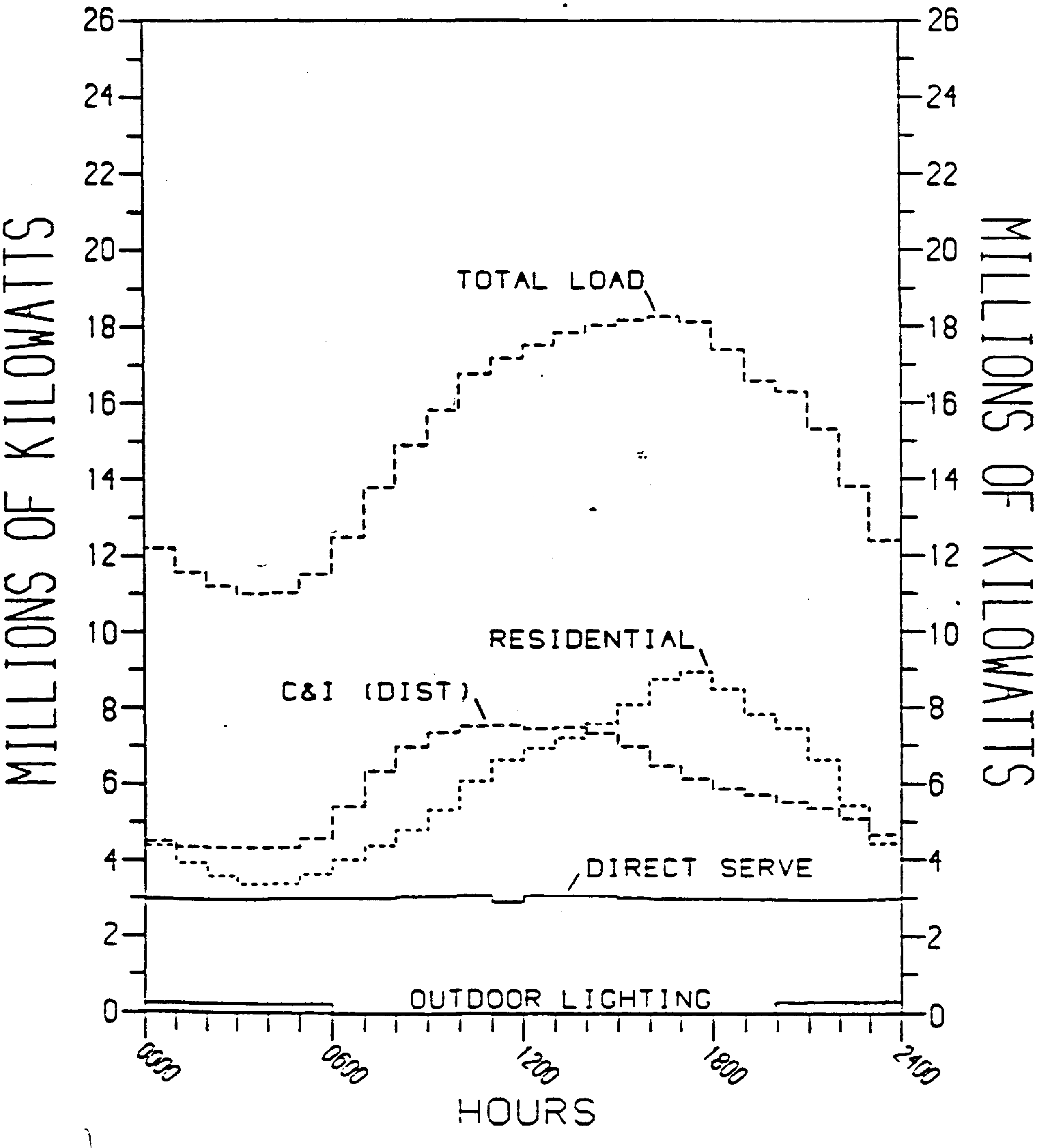
Except for electrical energy to power fans, pumps and controls, most of the energy required to run desiccant systems is thermal energy used to desorb the desiccant. The source of the thermal energy can be diverse. Natural gas, solar energy, waste heat from internal combustion engines, and condenser waste heat from vapor compression machines are a few of the sources that have been utilized successfully to date. Thermal energy, as opposed to electrical energy, also has the distinct advantage of being easily storable. This means that if it is produced partially or wholly by electrical means as (i.e. vapor compression machine or electrical resistance heating element) it can be produced during offpeak periods, offering a tremendous potential for peak shifting.

If a vapor compression machine is used to produce the thermal energy to desorb a desiccant (i.e. from the heating effect of the condenser coil) a free by-product is the cooling effect of the evaporator coil, which can be used to assist the desiccant system in the building cooling/dehumidification process. Machines that operate on this principle are termed hybrid vapor compression/desiccant systems, and their electrical COP is vastly improved over systems which use electrical resistance heaters or don't take advantage of the vapor compression evaporator coil cooling effect.

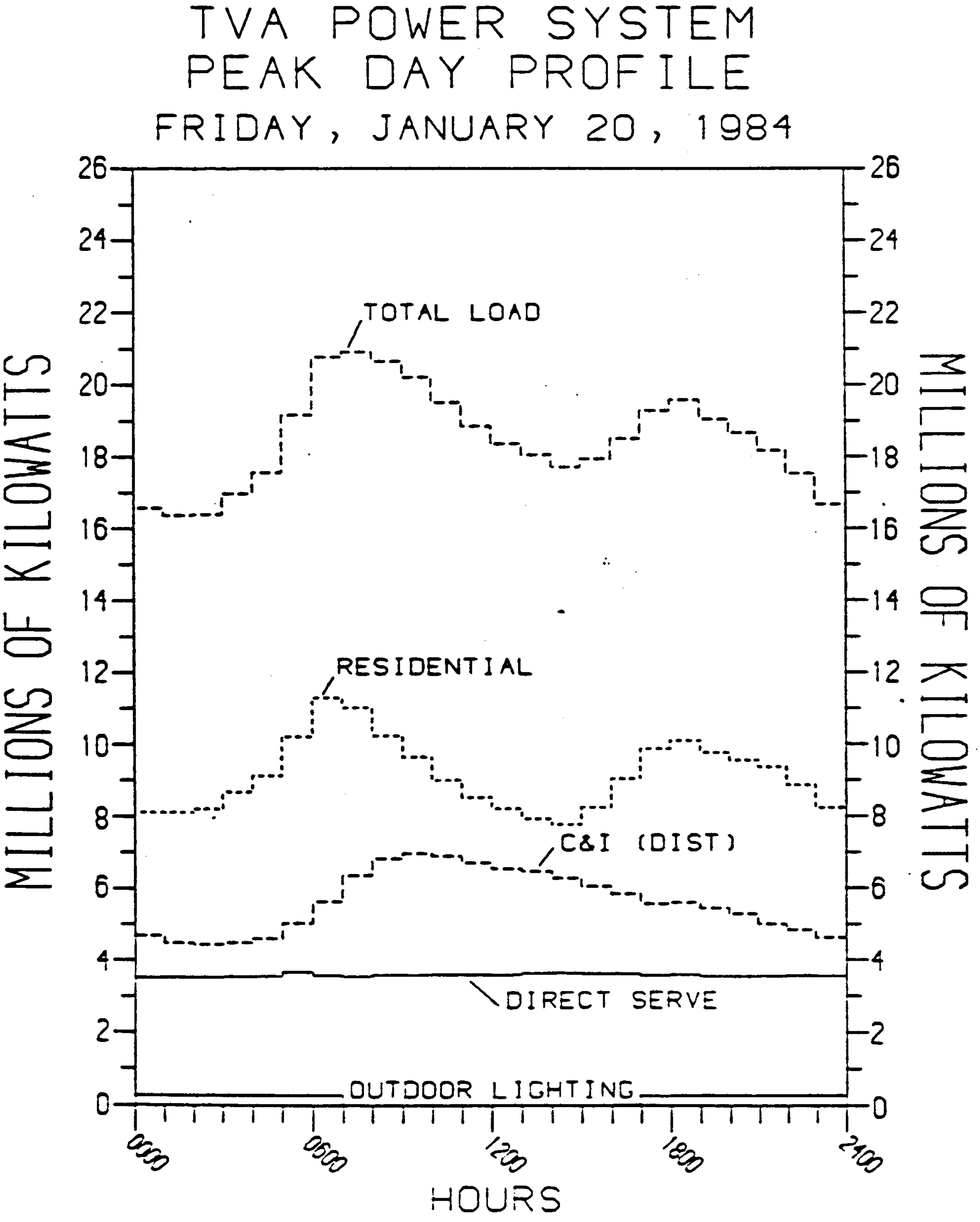
Desiccant systems whose thermal energy is derived partially or wholly from non-electrical sources (ex. solar with electric backup, or natural gas) can

-FIGURE 1-

TVA POWER SYSTEM
PEAK DAY PROFILE
WEDNESDAY, JULY 11, 1984



-FIGURE 2-



TVA POWER SYSTEM
PEAK DAY PROFILE
FRIDAY, JANUARY 20, 1984

-Table 1-

TVA's 1984 Generation Mix, Generation Contribution, and Incremental Cost of Power with Respect to Source

Generation Source	Percentage of Total Capacity (%)	Contribution as a Percentage of Generation Throughput %	Generation Costs (c/kWh)
Combustion Turbine	7.8	0.039	28.694
Coal-fired	55	60	2.205
Nuclear	18.4	21.6	1.823
Hydro	10.3	17.4	0.184
Pumped Storage	4.8	0.3	not avail.
Purchased	3.7	0.32	not avail.
Interchanged With Other Utilities	--	0.4	not avail.

be controlled not only to shave the peak demand, but also to conserve total electrical energy use. It is estimated that these types of systems could perform with dimensionless electrical COP's of 5 to 25, compared to 2 to 3 for conventional vapor compression air conditioners. These types of systems would most benefit the TVA power system in the mid to late 1990's when additional capacity is projected to be needed.

Assuming they are load-managed using thermal storage, it is estimated that residential-sized desiccant air conditioners could shave/shift up to 4 kW per residence off the peak demand during the extreme peak hour of a hot summer afternoon, if installed in place of a typical 3-3.5 ton vapor compression unit. Up to a 2 kW average peak demand reduction could be realized for the entire multi-hour daily peak period (10 a.m. to 10 p.m. for TVA) during the cooling season. Table 2 shows the potential for peak load reduction as a function of the number of residential desiccant systems installed in the TVA service area (TVA serves about three million customers in a 7 state region). The estimated penetration in the year 2005 was projected using very conservative "S-curve" market penetration assumptions. For comparison with this projected megawattage reduction, the nameplate rating of a typical TVA nuclear unit is 1100 MW.

Preliminary studies (refs. 4,5) have indicated that desiccant systems are capable of handling the entire (sensible and latent) cooling load of a residence in the Tennessee Valley, which is fairly representative of the humid southeastern portion of the country. Furthermore, when compared to a conventional vapor compression system in computer studies, a desiccant system with humidistat control was projected to be more capable of keeping conditions in a residence within the ASHRAE comfort zone, because of better humidity control.

TVA DESICCANT ACTIVITIES

TVA has inaugurated a strong program of national importance in the testing and demonstration of desiccant air conditioners for residences and commercial buildings. Current activities include:

Testing of three residential-sized units installed in test houses at TVA's Energy Use Test Facility (EUTF) in Chattanooga.

- Illinois Institute of Technology system - house #4, offpeak electric resistance or heat pump water heater driven with solar assist
- American Solar King Energymaster system - house #2, offpeak electric resistance or heat pump water heater driven with solar assist (also can be gas-driven)
- Rhodes Design A system - house #1, offpeak vapor compression driven

Monitoring of a commercial-sized unit in Richmond, Virginia.

- Meckler liquid desiccant system, Science Museum, uses diesel generator waste heat

Air quality studies.

- Rhodes Design B system - EUTF high bay air quality chamber, being tested for efficacy of desiccant system in removing hydrocarbons, toluene, carbon monoxide, sulfur dioxide, etc., and water vapor from a room-sized test chamber

-Table 2-

TVA Power System
Potential Summer Peak Load Reductions Due To Various Penetrations Of Residential Desiccant Air Conditioning Systems
(MW)

(No. of Residences)	Hourly Diversified Load (HDL) Power Savings Per Air-Conditioned Residence (kW)				
	0.5	1.0	2.0**	3.0	4.0***
1,000	0.5	1	2	3	4
10,000	5	10	20	30	40
100,000	50	100	200	300	400
250,000	125	250	500	750	1,000
286,939*	-	-	574	-	1,148
500,000	250	500	1,000	1,500	2,000
1,000,000	500	1,000	2,000	3,000	4,000
2,000,000	1,000	2,000	4,000	6,000	8,000

*Estimated situation in year 2005
**Projected seasonal HDL saving
***Estimated peak summer day HDL saving

Funding and technical support of research on hybrid vapor compression/desiccant air conditioners.

- o Mississippi State University, ED&T IECRP program,

and hosting of the EPRI/GRI/TVA-sponsored desiccant cooling workshop in Chattanooga in June, 1986.

TVA would welcome inquiries by outside agencies and institutions regarding additional desiccant testing at its Energy Use Test Facility. The facilities and staff include a full NBS-traceable instrumentaion calibration facility, and fully-staffed instrumentation & monitoring, and analysis sections equipped with state of the art computer and data acquisition equipment.

CONCLUSIONS

TVA has recognized the potential of desiccant air conditioning systems to assist its power program in the attainment of peak shaving, peak shifting and strategic conservation goals over the next two decades. High SEER's and good humidity control are seen as additional potential benefits to power system customers who install desiccant systems in place of vapor compression air conditioners. Accordingly TVA has instituted a strong program of testing and monitoring of desiccant air conditioning systems at its Energy Use Test Facility in Chattanooga. The use of desiccant systems for air pollutant removal is also being investigated.

REFERENCES

1. "A Bright Future Through the Wise Use of Electricity," TVA/TVPPA, Energy Services Plan, 1986.
2. Office of Energy Use, Tennessee Valley Authority, TVA Monthly Load Data Report, Fiscal Year 1984.
3. Power and Engineering Information Staff, Tennessee Valley Authority, "TVA Power Pocket Profile," FY 1984.
4. Majumdar, P., W.M. Worek and A. Lavan, "Simulation of a Cooled Bed Solar Powered Desiccant Air-Conditioning System," Illinois Institute of Technology.
5. Betbeze, A.R., "Analytical Evaluation of a Solar Heating and Cooling System in the Tennessee Valley," TVA internal report, 1983.

FIGURES

TABLES

Technical Note presented to the Desiccant Cooling Workshop,
Chattanooga, TN, June 10-12, 1986.

CORRELATING FIELD TESTS ON SOLID DESICCANT OPEN-CYCLE COOLING SUB-SYSTEMS

I.L. Maclaine-cross

School of Mechanical and Industrial Engineering
University of New South Wales
P.O. Box 1
Kensington NSW 2033
AUSTRALIA
Telephone +61 2 6974112

1. Introduction

Solid desiccant open-cycle (SDOC) cooling systems can use low temperature heat efficiently (Penney and Maclaine-cross 1985). They are now commercially available (Venhuizen 1984) and a number are currently under field tests. More field tests may be necessary in the future on SDOC systems with lower cost or higher performance. It is desirable that the results of all tests be reported in a consistent manner so valid comparisons between performance and economics of different systems can be made.

Closed cooling cycle performance depends mainly on the evaporator and condenser refrigerant temperature. For vapor compression cycles, only two independent variables are important: evaporator and condenser temperature. Correlation methods and correction to standard conditions are well understood.

SDOC cooling cycle performance depends on ambient and return temperature and humidity. In addition, the variation of performance for off standard or design conditions may vary considerably from machine to machine.
The purpose of this note is to propose correlating equations (Section 3) based on the known characteristics of SDOC cycles (Section 2). These equations may be used to calculate the performance parameters usually quoted (Section 4).

Traditionally cooling systems have been compared using fixed standard return and ambient states (ARI). These comparisons can be highly misleading for domestic installations, as to both relative capital cost and energy consumption (Coellner 1985). Economic comparisons between desiccant cooling and vapor compression can be based on computer simulations using correlating equations but field tests over at least one year under the climatic conditions of interest is perhaps the most reliable as well as the most expensive.

2. Characteristics of SDOC Cycles

The generic characteristics of SDOC cooling machines are discussed briefly in this section. It is assumed that the internal air states in the machine are inaccessible. Some manufacturers have prohibited access to the inside of machines, but in any case, internal states are not relevant to the overall performance of machines.

Machines may be indirect or direct fired. It is desirable that the same parameters be used in correlating both types of machines. Heating water mass flow rate does not exist for direct fired machines so it will not be used for indirect fired machines. For direct fired machines, the regeneration temperature t_{reg} is the bulk mean temperature of the regeneration stream entering the dehumidifier. If this is not accessible, it cannot be measured. For direct fired gas or electric machines, it is not necessary to know the regeneration temperature because the heat available is not affected by this. Solar machines are all indirect fired. If water is the heat transfer fluid, for typical heat exchanger efficiencies the lower or return water temperature will be closer to the regeneration temperature t_{reg} than the supply or mean water temperature. The return water temperature also appears in the $\bar{\phi}$,F chart method (Duffie and Beckman 1980) which is probably the most appropriate simple system performance correlation for predicting solar cooling auxiliary energy consumption.

A generic SDOC machine is shown schematically in Fig.1. Regeneration heat is transfered at rate, $\dot{Q}_{reg}$ (W) and temperature t_{reg} (°C). The ambient air state A has dry bulb temperature t_A (°C) and wet bulb temperature t_A^*. Cooled air is supplied at state S, t_S, t_S^* (°C) and mass flow rate $\dot{m}_S$ (kg dry air/s) to the air conditioned space and has specific heat at constant pressure c_{pa} (J/kg K). The air returns to the machine at state R, t_R, t_R^* (°C). Electric power $\dot{W}_e$ (W) is supplied to the machine.

Atmospheric pressure effects the performance of SDOC machines in a number of ways. It should be recorded and reported. Fans tend to maintain a constant volume flow rate when driven at constant rotational speed. If the atmospheric pressure drops, the air density and mass flow rate drop reducing the capacity of the machine. Test at a high altitude should be corrected for the change in air density. The effect of atmospheric pressure is also manifest through the change in the position of the saturation line on the psychrometric chart. This may be minimized by describing performance relative to the saturation line by using dry and wet bulb temperatures in the correlating equation.

The four parameters of interest in describing the performance of SDOC machines are the cooling capacity = $\dot{m}_S(h_R-h_S)$ (W), the thermal coefficient of performance COP = $\dot{m}_S(h_R-h_S)/\dot{Q}_{reg}$, the regeneration temperature t_{reg} (°C) and the electric power consumption $\dot{W}_e$ (W). The cooling effect CE = h_R-h_S (J/kg K) is sometimes used in place of the cooling capacity. The electric power consumption of SDOC machines is small and almost independent of ambient, supply and return air states. There are thus three dependent variables which must be correlated with six independent variables.

Kang (1985) has made theoretical performance predictions for a number of SDOC machines. Kang found that $\dot{Q}_{reg}$, CE and t_{reg} are linear functions of the space R and ambient A states. Figure 2 shows this variation for the well known ventilation cycle with 95% component efficiencies and balanced mass flows. Linear correlating equations would seem appropriate and will allow the use of standard multiple linear regression computer programs in reducing the results. If these correlating equations are expressed relative to the saturation line, not only are atmospheric pressure effects minimized but the number of correlation coefficients to be calculated is reduced from 21 to 12 or from seven to four for each dependent variable.

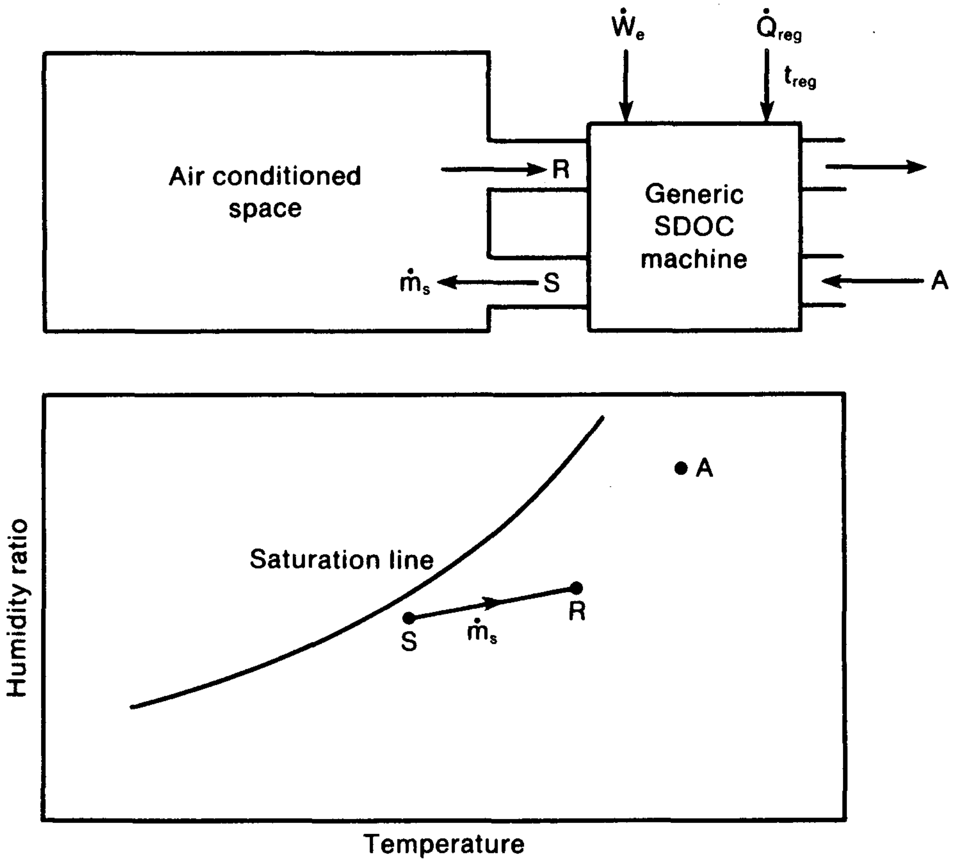


Figure 1 Schematic and Psychrometric Chart for a Generic SDOC Machine

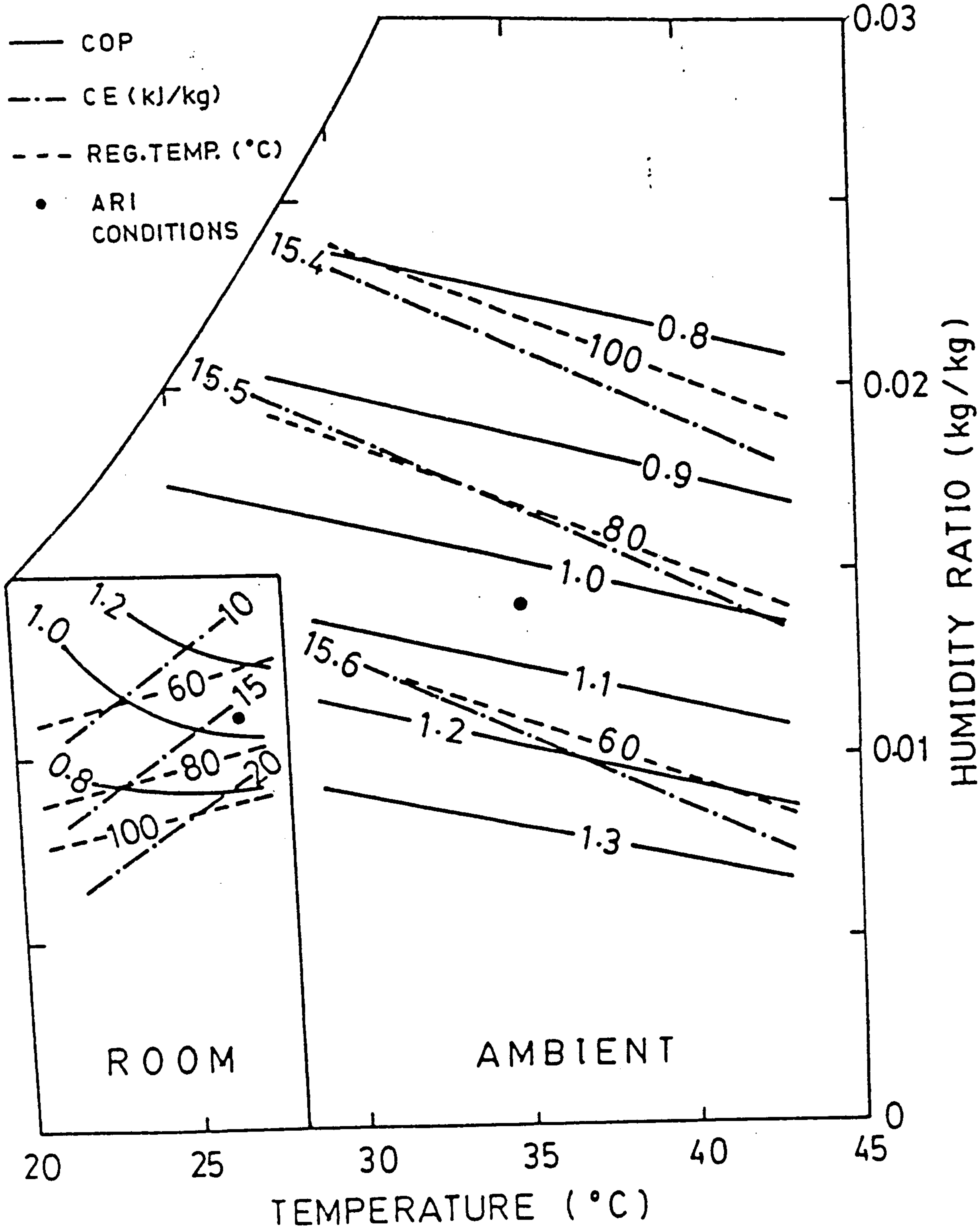


Fig. 2 Performance map for ventilation cycle.

There will be many thousand data scans to be correlated if a system has its performance measured hourly over a period of a year. Equations of simple form are thus highly desirable to minimize variable conversions. Dimensionless equations minimize the possibility of unintended unit conversion errors and make the results meaningful to professionals regardless of the unit system they are accustomed to.

The moisture capacity of dehumidifiers in SDOC coolers is substantial giving a response times of the order of an hour. This can cause underestimation of the steady state performance from tests on real buildings if the system is turned off every day. In estimating steady state performance from the results of such tests it is recommended that at least the first hour of operating data for each day be discarded.

SDOC machines can have a number of operating modes, e.g. two speed fans, dehumidification on or off. Each operating mode should be correlated separately.

The performance of SDOC machines depends also on a number of mass flow ratios, e.g. return to supply air flow ratio, regeneration fluid to supply air mass flow ratio. These should be recorded and reported.

An energy balance on the complete machine will indicate the accuracy of the measurements and is an important experimental check. SDOC machines should also satisfy conservation of water mass.

3. The Proposed Correlating Equations

The proposed correlating equations below are linear with dimensionless coefficients A_i, B_i, C_i, D_i. They were selected using the characteristics of SDOC cycles discussed above.

$$t_R^* - t_S^* = A_1(t_S - t_S^*) + B_1(t_R - t_S) + C_1(t_A - t_A^*) + D_1(t_A^* - t_R^*) \qquad (1)$$

$$Q_{reg}/(c_{pa} m_S) = A_2(t_S - t_S^*) + B_2(t_R - t_S) + C_2(t_A - t_A^*) + D_2(t_A^* - t_R^*) \qquad (2)$$

$$t_{reg} - t_A = A_3(t_S - t_S^*) + B_3(t_R - t_S) + C_3(t_A - t_A^*) + D_3(t_A^* - t_R^*) \qquad (3)$$

The coefficients A_i, B_i, C_i, D_i, are obtained by multiple linear regression (Carnahan *et al.* 1969) of the experimental results using standard computer program packages *e.g.* IMSL, NAG, SPSS.

The left hand sides of equations (1,2,3) are related to cooling capacity, energy consumption and regeneration temperature, respectively. Since these are the variables of economic importance which must be estimated most accurately, statistical theory requires that they should be the dependent variables for the regression. The equations may be rearranged after the above regression coefficients have been calculated, with new dependent variables and new coefficients calculated from the regression coefficients.

These coefficients may also be calculated theoretically from the component efficiencies, mass flow ratios and desiccant, water and air properties. The expressions are, however, complex and outside the scope of this note.

Consistent SI units have used in this note, but the values of the dimensionless coefficients would be the same if consistent British units were used.

4. Use of the Correlating Equations

Equations (1,2,3) may be solved for CE, COP, t_{reg} and performance maps (e.g. figure 2) for standard atmospheric pressure (101325 Pa) plotted on psychrometric charts similiar to Kang (1985). SDOC systems may be compared by comparing the performance maps, however, the regression coefficients would be used to calculate the coefficients of new equations for comparisons based on systems simulations.

The calculation of CE, COP and t_{reg} using equations (1,2,3) is not obvious and will be outlined here for given return and ambient states and latent to sensible heat ratio, R. Equation (1) reduces to the following linear relation between the supply air wet and dry bulbs when the known coefficients and temperatures are substituted.

$$t_s^* = A+Bt_s \qquad (4)$$

If the space load line is plotted on a psychrometric chart and equation (4) in its vicinity, their intersection will give the supply air state S. CE may be calculated from specific enthalpies at R and S read from the chart. Q_{reg} may be calculated from equation (2) and hence COP. t_{reg} may be calculated from equation (3).

Computers cannot read a psychrometric chart so they have to calculate from the equations of the chart. These equation are often poorly arranged in psychrometric programs resulting in long computation times. For this problem the four equations which must be solved simultaneously are the psychrometer equation, the saturation line, the load line and equation (4). Simple linearizations may be used to give an approximate explicit solution which may be made arbitrarily accurate by recalculating the constants with the most recent values of the dependent variable. The return dew point is recommended as an initial approximation to the supply dry bulb temperature, t_s.

The initial supply wet bulb temperature, t_s^*, is then calculated from equation (4). The coefficients of the linearized saturation line, a and b, are calculated at t_s^* using

$$b = dW_s^*/dt^* \qquad (5)$$

$$a = W_s^*+bt_s^* \qquad (6)$$

and the usual psychrometric functions (*e.g.* Maclaine-cross 1974). The psychrometer constant, K, is obtained from

$$K = (c_{da}+W_s^*c_v)/(h_v-h_f^*) \qquad (7)$$

where the specific heat of dry air, c_{da}, is evaluated at $(t_s+t_s^*)/2$, the specific heat of water vapour, c_v, is almost independent of temperature, the specific enthalpy of water vapour, h_v, is evaluated at t_s and the specific enthalpy of water, h_f, at t_s^*. The load line constant, K_1, is obtained from

$$K_1 = R(c_{da}+W_sc_v)/h_v \qquad (8)$$

where c_{da} is evaluated at $(t_R+t_S)/2$ and h_g at t_R. The supply dry bulb temperature may now be calculated from the following simultaneous solution to the linearized psychrometric equations

$$t_S = (W_R-K_1t_R-a-A(b+K))/(K(B-1)-K_1+bB) \qquad (9)$$

The calculations may be repeated from the top of the previous paragraph a second time. Further iterations are unnecessary because not only are the equations a very good approximation initially but convergence of the algorithm is quadratic for the usual psychrometric functions.

5. Conclusion

Experimental results are useless if they cannot be compared consistently with theory and other results. Equations (1,2,3) are proposed to enable this to be done.

There are many alternatives or equivalent approaches and equations (1,2,3) arose from discussions between SERI, CSU and ASK researchers and engineers. The author would welcome further comments or suggestions and they may be sent to him or Terry Penney at SERI.

6. Acknowledgements

The author is grateful for suggestions and encouragement to write this note from George Lof and Terry Penney. Jim Coellner and John Mitchell made written reviews of the first draft. Sibel Beba of CSU was the first experimenter to use equations (1,2,3) and her comments and difficulties have inspired this revision.

7. References

B. Carnahan, H.A. Luther and J.O. Wilkes (1969), "Applied Numerical Methods", Wiley, New York.

J.A. Coellner (1985), American Solar King Corporation, letter to author December 6, 1985.

J.A. Duffie and W.A. Beckman (1980), "Solar Engineering of Thermal Processes", Wiley, New York.

T.S. Kang (1985), "Adiabatic Desiccant Open Cooling Cycles", M.Eng.Sci. thesis, School of Mechanical and Industrial Engineering, University of New South Wales, Australia.

I.L. Maclaine-cross (1974), "A Theory of Combined Heat and Mass Transfer in Regenerators", Ph.D. thesis, Department of Mechanical Engineering, Monash University, Australia.

T.R. Penney and I.L. Maclaine-cross (1985), "Promising Advances in Desiccant Cooling", Proc. Solar Buildings Conference, Washington, DC, U.S. Department of Energy, pp. 72-85, March 18-20.

D. Venhuizen (1984), "Solar King's Cooling Gambit", Solar Age, 9, 10, pp. 25-27.

CLOSED CYCLE DESICCANT REGENERATIVE HEAT PUMP

Under GRI Contract #5083-243-0933 which started May 1, 1984, The Zeopower Company designed, constructed and tested a breadboard prototype of a half-ton zeolite regenerative heat pump. This prototype provided proof of the concept that combining energy regenerative heat exchangers with the closed cycle solid zeolite-refrigerant gas system results in an adsorption heat pump with low initial cost and high seasonal heating and cooling COPs and therefore low operating cost.

The state-of-the-art in adsorption technology before the commencement of this contract was defined by low cooling COPs (less than 1.0 for single-effect systems) and high initial costs and parasitic energy demands usually associated with cooling towers, large numbers of pumps, fans, etc. Higher COPs could be obtained by double-effect or two-stage systems, however, at the price of even higher initial costs and parasitic energy demands. These facts have resulted in very low market penetration of commercial systems and a residential heat pump market completely dominated by electrical vapor-compression units.

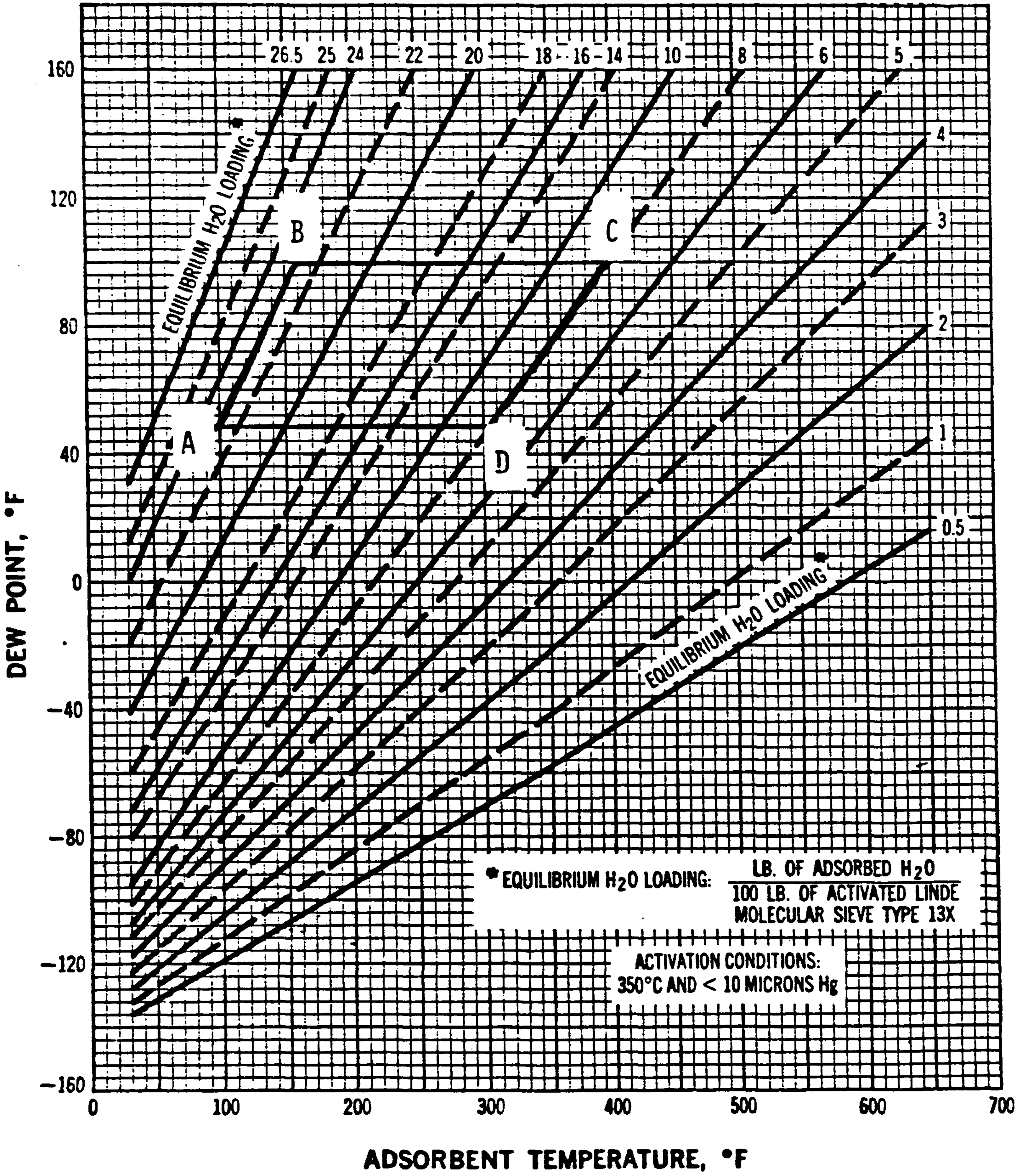
Zeolites have very nonlinear adsorption properties which result in a weak dependence of system performance on condenser temperature (similar to vapor-compression cycles) thus eliminating the need for cooling towers and reducing costs. The regenerative heat exchanger concept, on the other hand, permits recycling of as much as 80% of the thermal energy required resulting in a potential increase of cooling COP from 0.6 to 2.0 with a very slight increase in capital costs.

This cooling system is based on the solid zeolite-water vapor adsorption principle, and its method of operation is similar to other adsorption systems such as the ammonia-water and LiBr-water. Figure 1 represents the cycle: From Point A to B to C is the desorption part of the cycle -- the zeolite is heated to 400°F and water vapor is condensed at 100°F; from Point C to D back to A is the adsorption part of the cycle -- the zeolite is cooled back to 100°F and adsorbs water vapor at a boiling point of 47°F. About 15 weight percent of water is desorbed per cycle and, since the heat of vaporization of water is about 1,000 Btu/lb, 12 pounds of water per hour will be needed for 1 ton cooling (12,000 Btu/hr) which corresponds to 80 pounds of zeolite cycled once per hour.

In order to provide continuous cooling from an intermittent cycle, two separate zeolite containers are used -- one desorbing while the other is adsorbing as shown in Figure 2. This permits heat to be transferred from the

LINDE Molecular Sieve Type 13X Pellets
Water Adsorption Isosteres

FIGURE 1



* EQUILIBRIUM H_2O LOADING: $\frac{\text{LB. OF ADSORBED } H_2O}{\text{100 LB. OF ACTIVATED LINDE MOLECULAR SIEVE TYPE 13X}}$

ACTIVATION CONDITIONS:
350°C AND < 10 MICRONS Hg

ENERGY BALANCE DIAGRAM

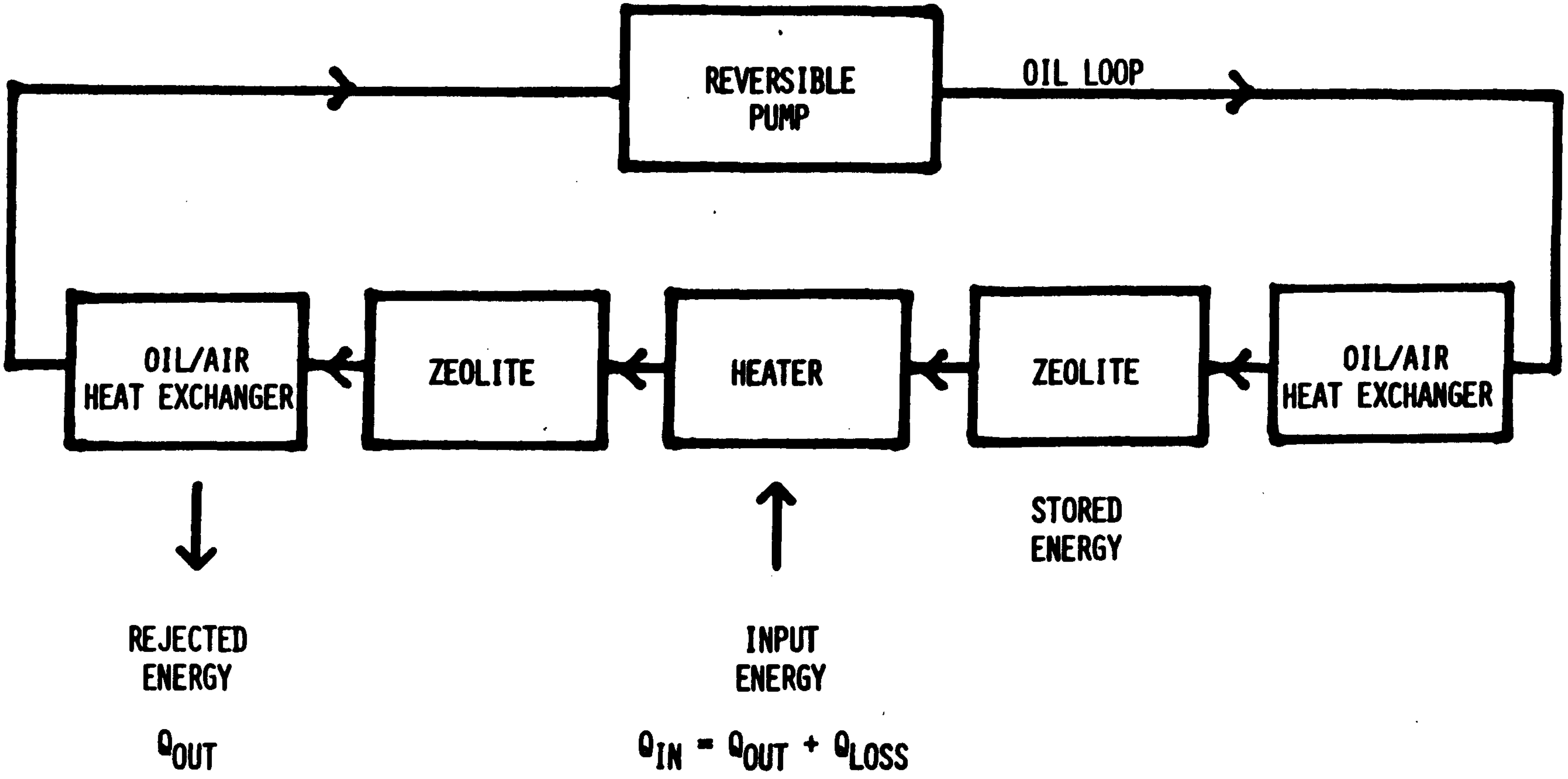


FIGURE 2 ZEOLITE HEAT PUMP CYCLE SCHEMATIC

cooling container to the heating container and reduces considerably the required energy input, thereby increasing the system COP to above 1. This figure also represents the block-diagram for detailed energy balance of the system. Careful analysis of the preliminary test results indicate that the energy supplied by the heater equals the sum of the energy rejected when cooling the output oil back to ambient temperature plus all losses. The losses are of two types: normal heat losses through the insulation of pipes and containers and recycling losses of the energy stored in the zeolite heat exchanger. At this time, we recycle up to 82% of the energy stored in this heat exchanger and hope to improve this even further. In Figure 3 the cycle COP of synthetic and natural zeolites are compared with the natural being slightly better. Figure 4 compares the cycle COP and capacity of a 1-ton system as a function of maximum zeolite temperature.

After the cycle analysis, the system's components were analyzed. In a 36-foot long heat exchanger, the heating or cooling cycle can be completed in 30 minutes each permitting one complete cycle for each container with 40 pounds of zeolite in it. Figure 5 shows the normalized input and output temperatures of the heat exchanger during the cooling cycle while Figure 6 gives the same data for the one being heated.

The heat regeneration concept is presented in Figure 7. Here the burner has to supply energy only to keep the temperature constant at 100% (solid line) and this energy is exactly the same as the one rejected to the atmosphere (dotted line) as required by the law for conservation of energy.

The key R&D issues were the lack of information on zeolite properties such as specific heat, thermal conductivity, heat and mass transfer properties, etc., and the limited data available on regenerative heat exchangers where all previous work has been done involving materials with linear properties. Therefore, we have determined for the first time analytically and observed experimentally the propagation of temperature fronts through zeolites during adsorption and desorption, determined the rate limiting steps for, and the relationship between, heat/mass transfer in the solid and the effects of density and surface-to-volume ratio on the time response and performance of the system.

A novel serpentine-like regenerative zeolite-hot fluid heat exchanger was designed, constructed and tested. It exhibits the unique properties of using atmospheric pressure to obtain and maintain good thermal contact and physical integrity during thermal cycling while allowing vapor transport in/out of the zeolite. It minimizes fluid volume for low cost and short cycling time while

NATURAL VS. SYNTHETIC ZEOLITE

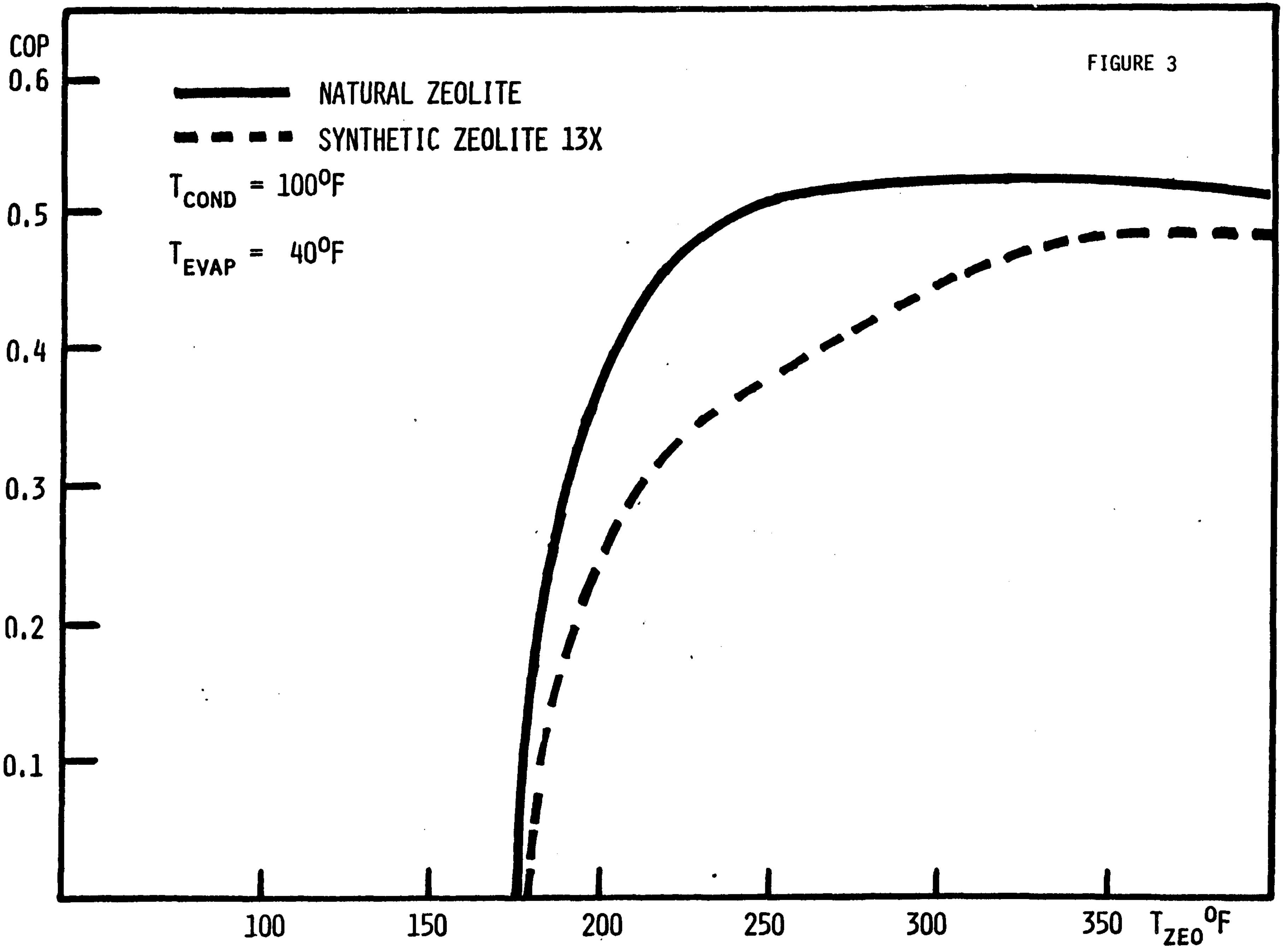


FIGURE 3

CYCLE COP & CAPACITY AS A FUNCTION OF ZEOLITE TEMPERATURE

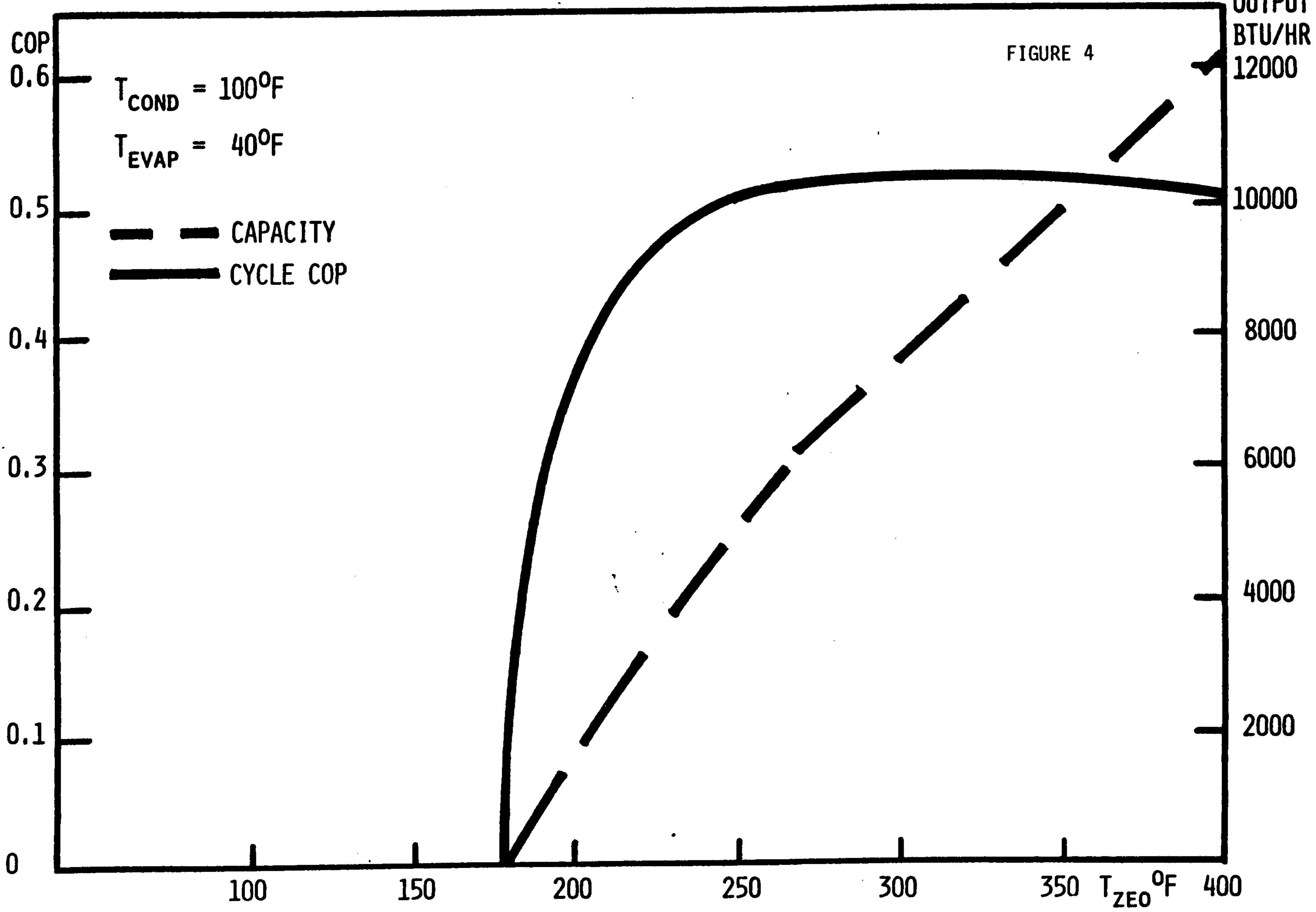


FIGURE 4

COLD LIQUID INPUT/OUTPUT CURVES FOR A 36-FOOT HEAT EXCHANGER

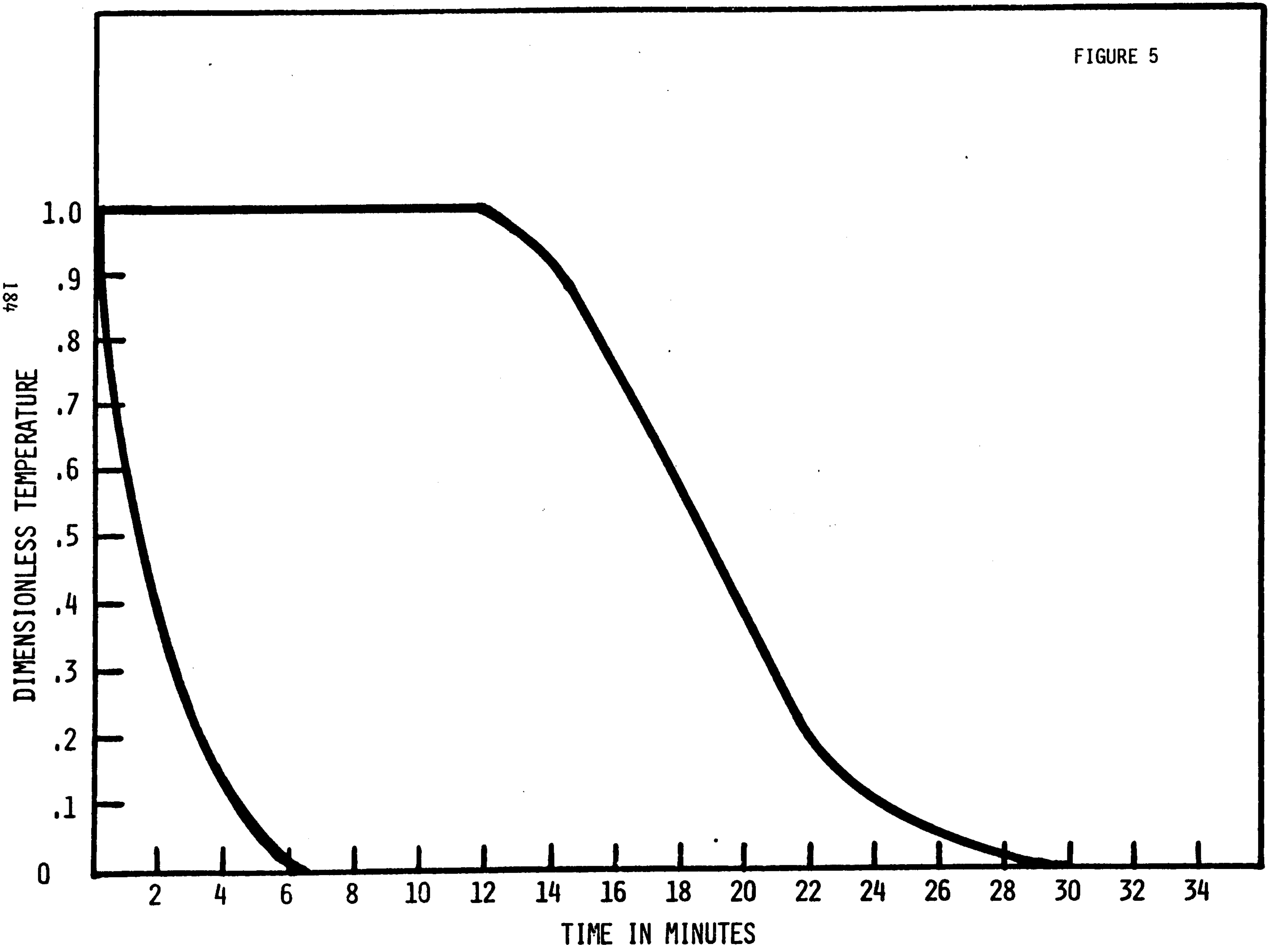


FIGURE 5

HOT LIQUID INPUT/OUTPUT CURVES FOR A 36-FOOT HEAT EXCHANGER

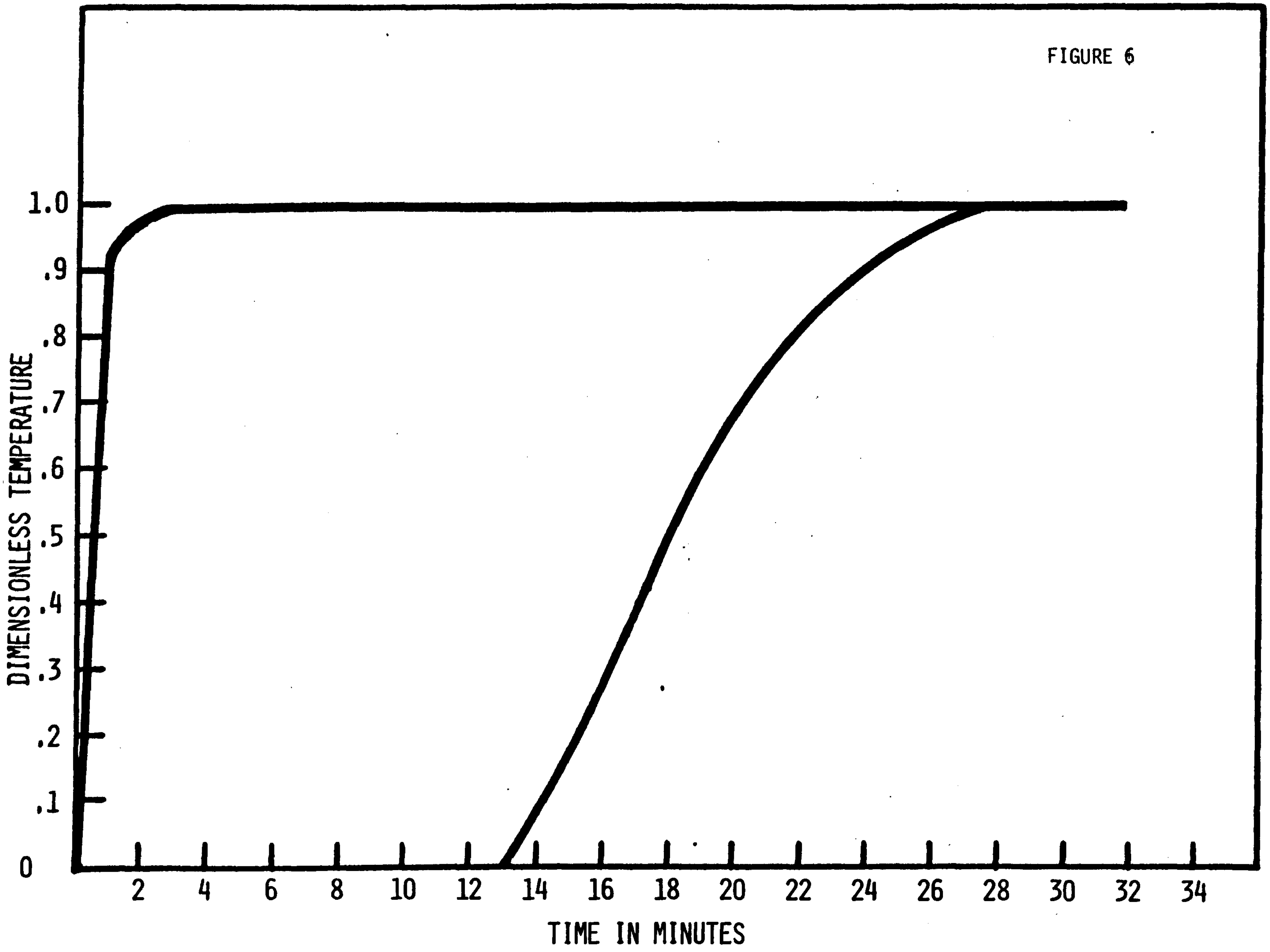


FIGURE 6

HEAT REGENERATION FOR A 36-FOOT HEAT EXCHANGER

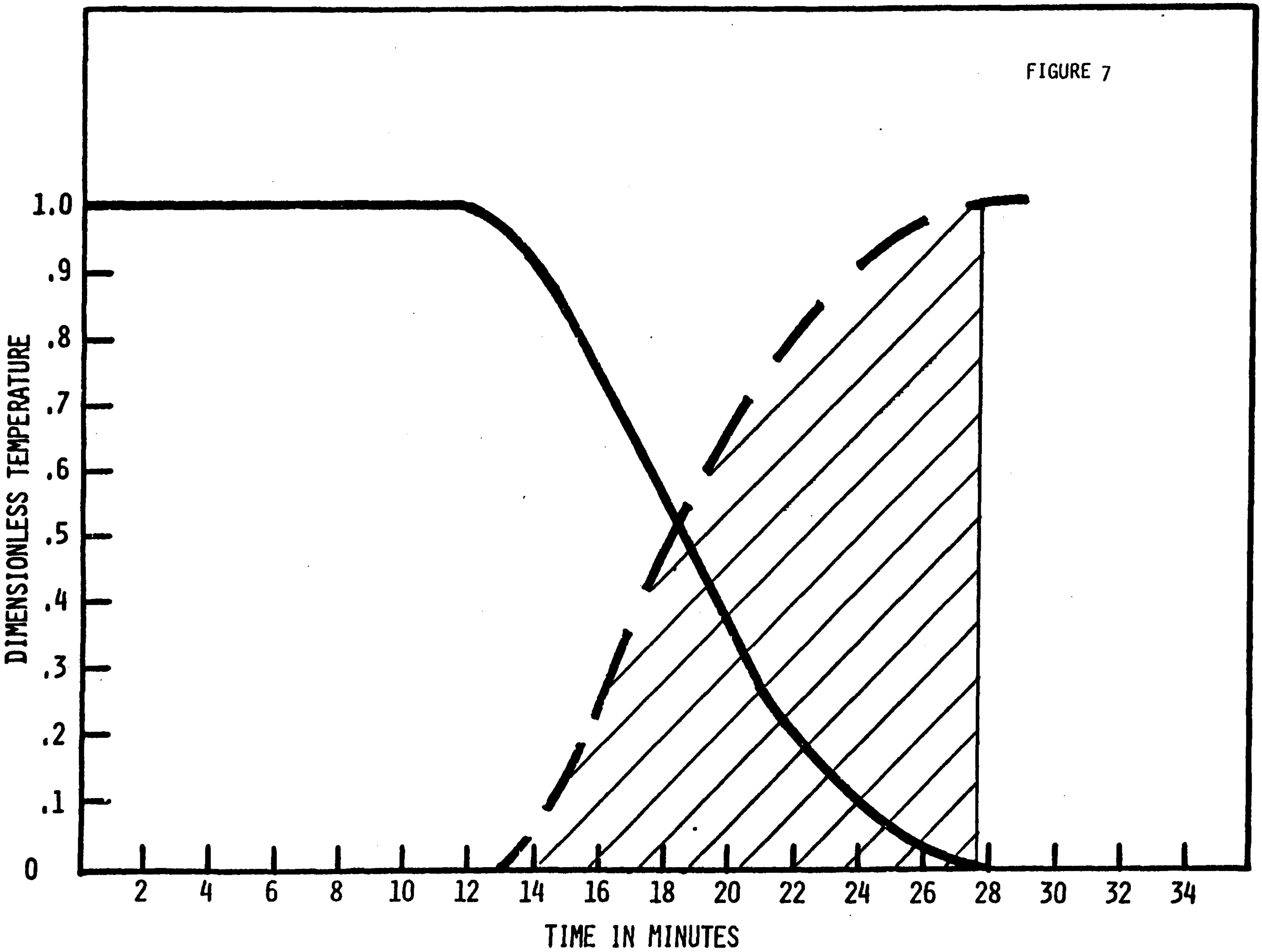


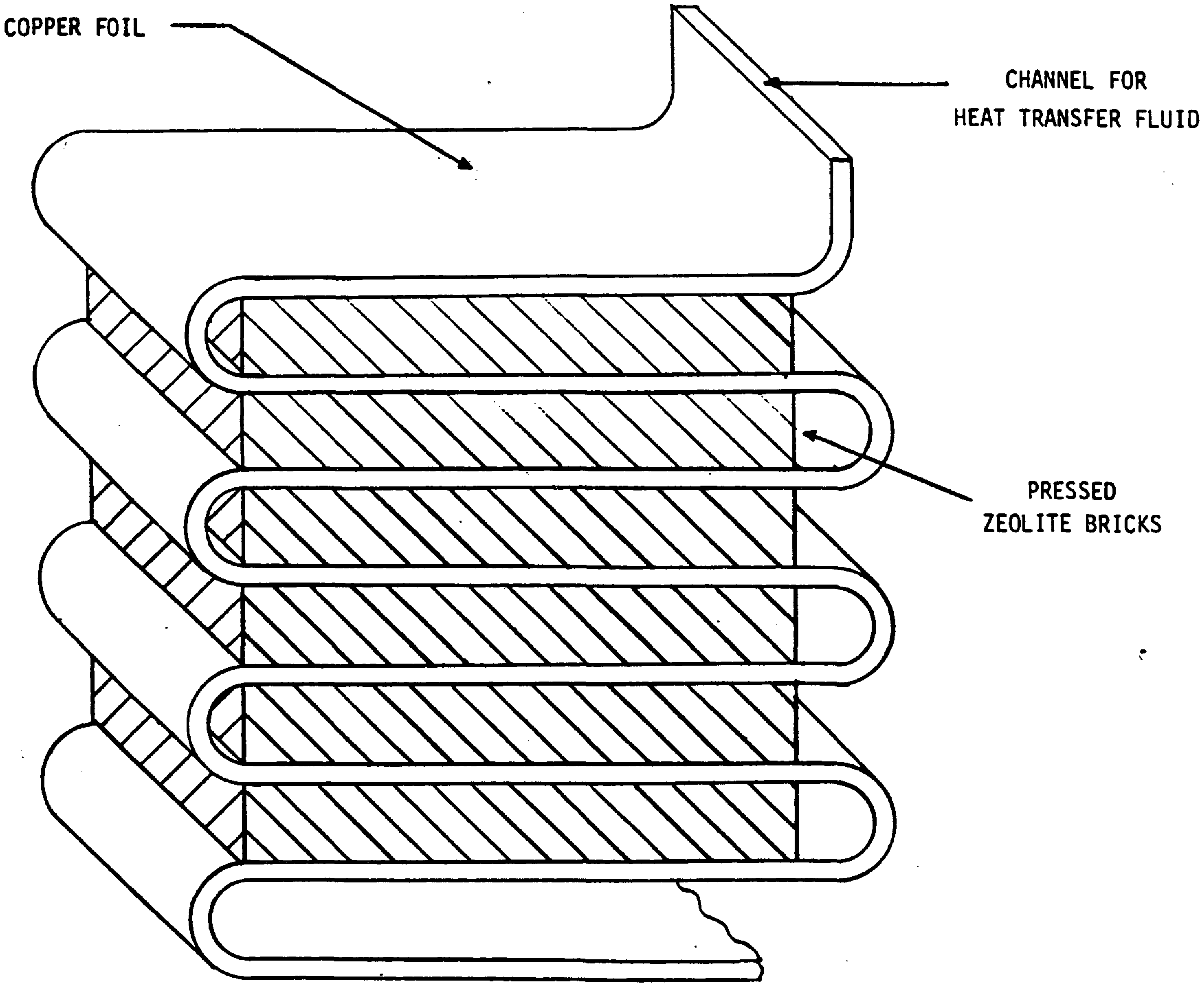
FIGURE 7

maintaining uniform fluid flow with small hydraulic resistance. We have further optimized its efficiency while minimizing weight, volume, and cost, and keeping thermal losses low. The design of the system was completed and a schematic cross section of the zeolite heat exchanger is shown in Figure 8. Construction of the breadboard prototype was completed and a cut-away view of the zeolite cylinders is shown in Figure 9.

We designed, built and tested a half-ton zeolite heat pump with modulation capability between 10% and 100% capacity consisting of two regenerative heat exchangers with 45 pounds of zeolite each (90 pounds total). The heat pump was operated using different configurations and control strategies over a period of 10 months and demonstrated a cooling COP of above 1.2 and a heating COP of above 1.8. The complete performance maps for the heat pump were generated and the expected weak dependence on condenser and evaporator temperature was demonstrated. After numerous measurements at different oil flow rates and cycling times the proper control strategy was established. This strategy is based on keeping the cycle time constant which causes a considerable increase in COP as the capacity is reduced as shown in Figure 10 and, since cooling systems operate most of the time below their maximum capacity, this control strategy results in a considerably higher seasonal performance factor (seasonal COP). For better clarity of presentation of the data we have combined the two curves of capacity and COP versus oil flow rate into one single curve representing the COP as a function of capacity. For the breadboard prototype steady state performance under ARI conditions (47°F evaporator, 95°F condenser) this performance curve is shown in Figure 10 together with the experimentally determined points from runs over a period of 6 months. This performance curve is for the configuration with two 48-foot heat exchangers and 90 pounds of zeolite.

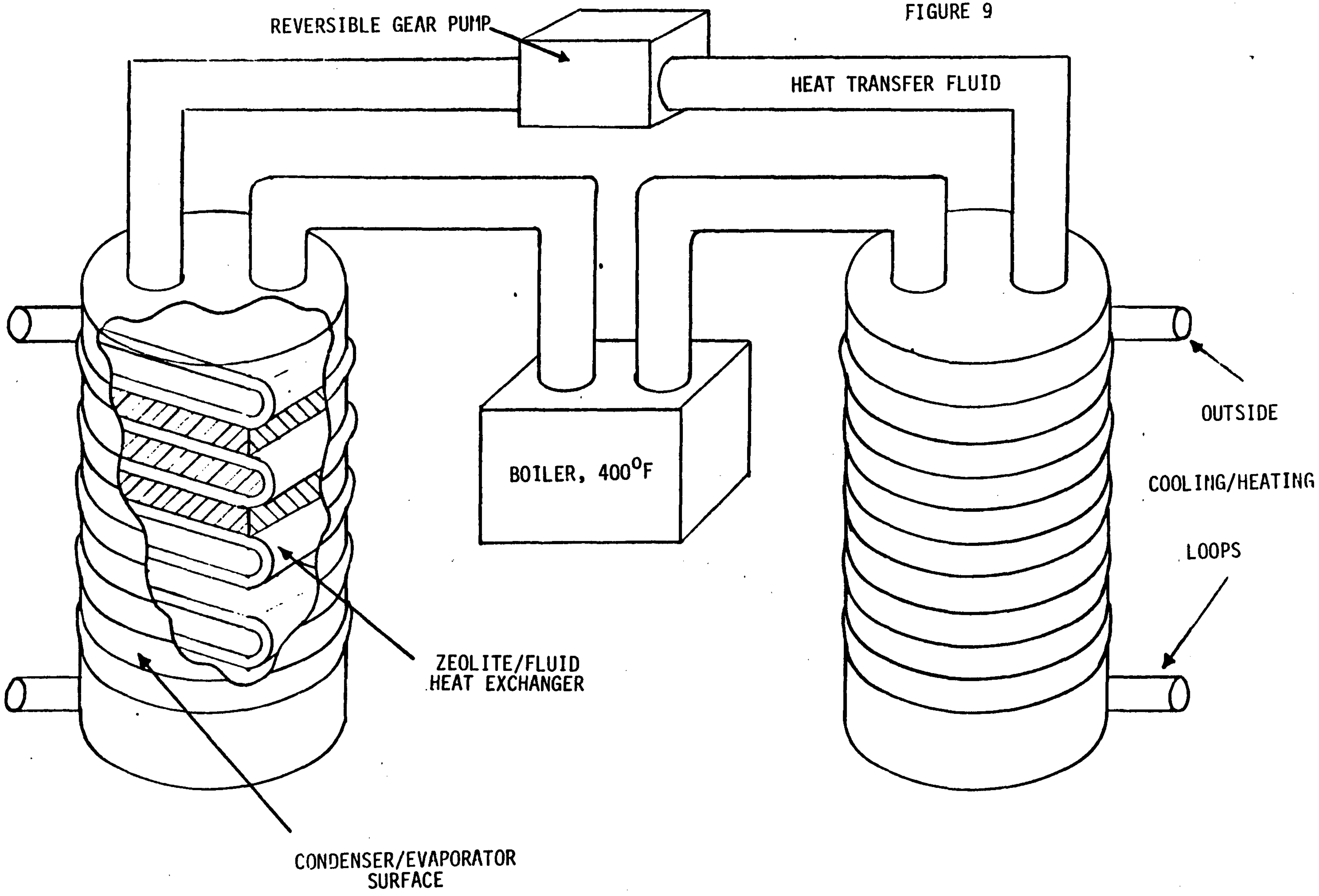
The system performance was then tested as a function of condenser and evaporator temperatures. The tests were performed under steady state conditions and constant oil flow rate with the evaporator held constant at ARI conditions of 47°F while the condenser temperature was varied from 90°F to 140°F. At each different condenser temperature the system was cycled until steady state was obtained before the capacity and COP were determined. The results of this test are shown in Figure 11 where the capacity curve is plotted against condenser temperatures. The data indicates that while the capacity is reduced with increasing condenser temperature the reduction is monotonic and reasonably small up to 140°F, which clearly demonstrates the system's capability to operate with air-cooled condenser and the elimination of a water cooling tower. This is a

ZEOLITE/OIL HEAT EXCHANGER



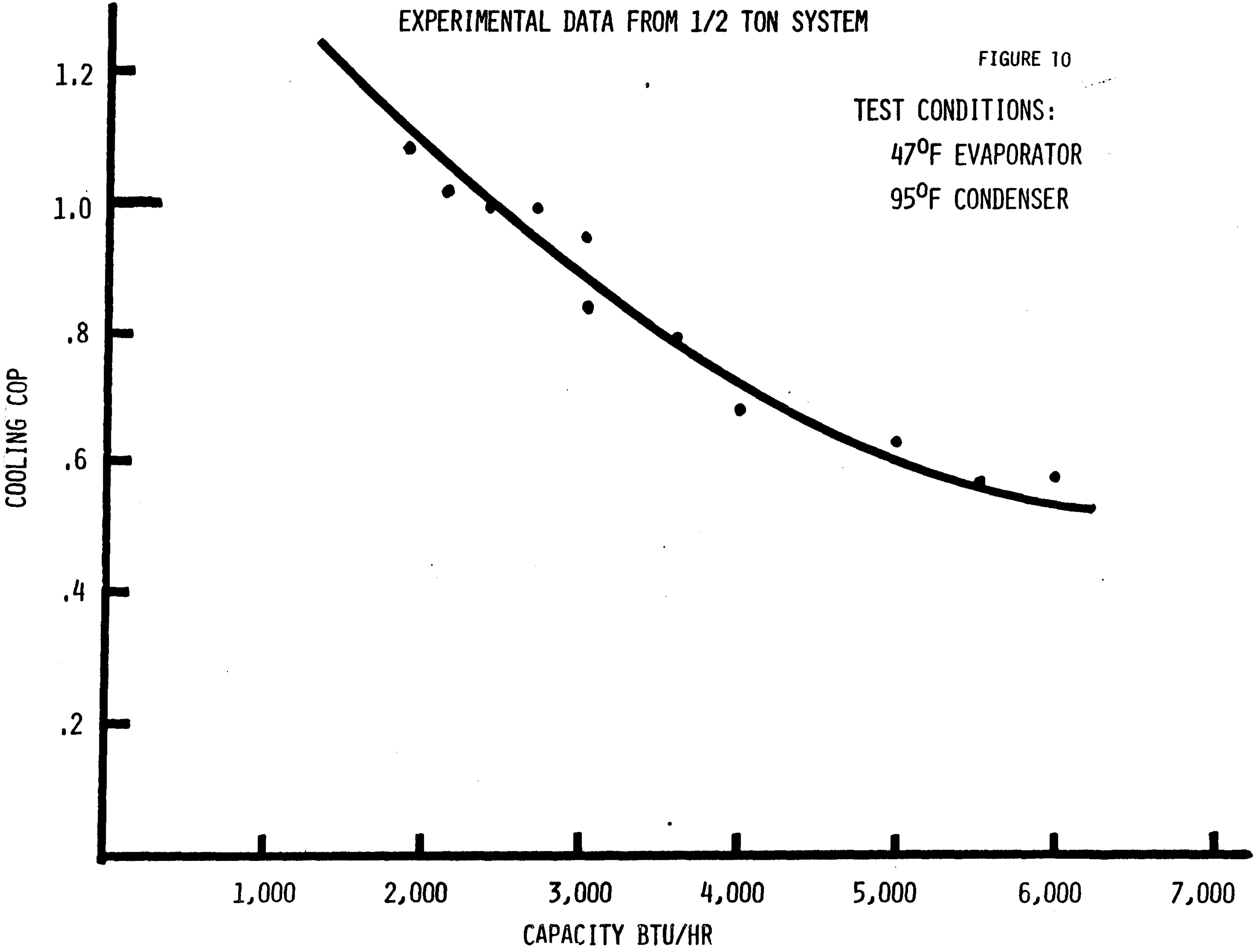
DESIGN CONCEPT FOR THE CONDENSER/EVAPORATOR HEAT EXCHANGERS

FIGURE 9



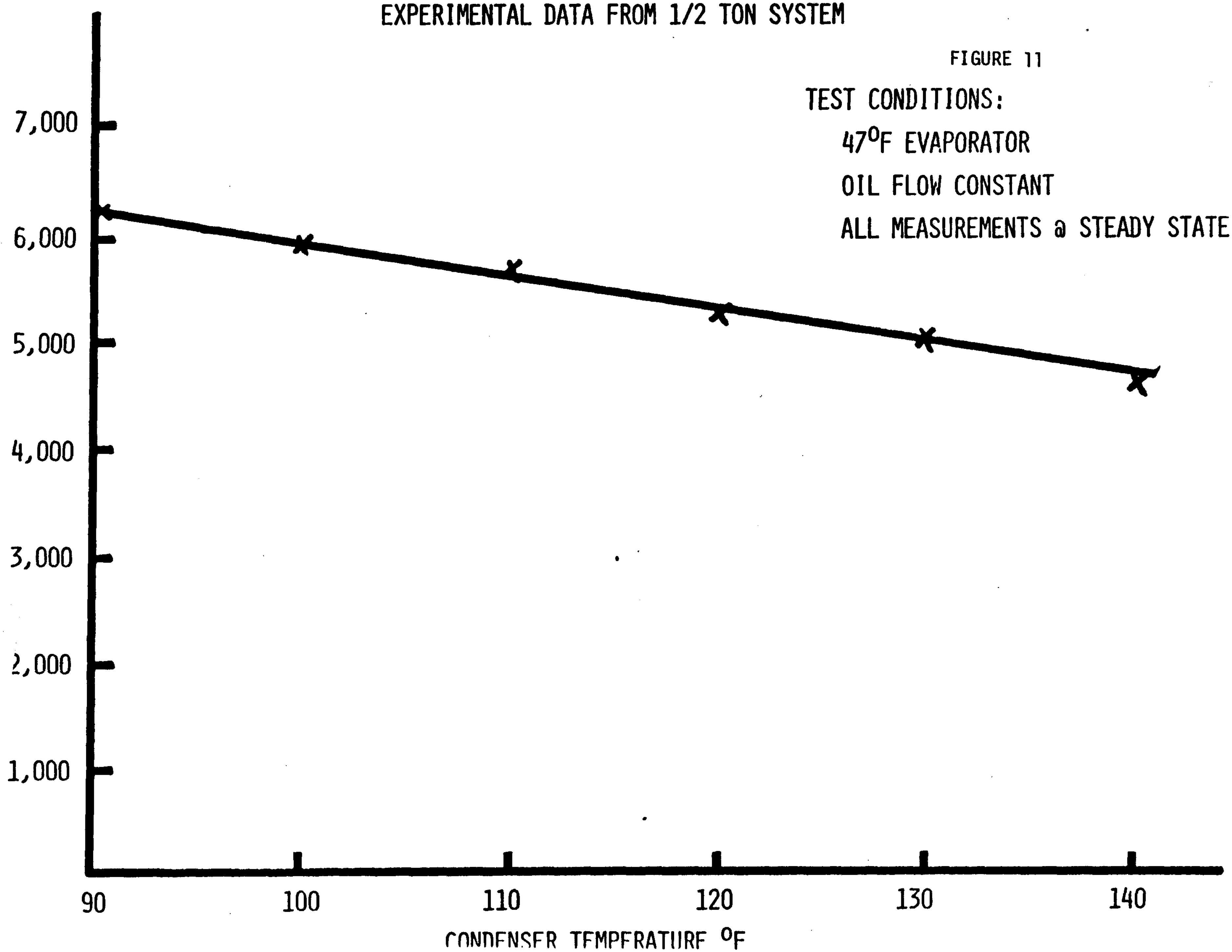
COOLING COP VS. CAPACITY
EXPERIMENTAL DATA FROM 1/2 TON SYSTEM

FIGURE 10



CAPACITY VS. CONDENSER TEMPERATURE
EXPERIMENTAL DATA FROM 1/2 TON SYSTEM

FIGURE 11


TEST CONDITIONS:
47°F EVAPORATOR
OIL FLOW CONSTANT
ALL MEASUREMENTS @ STEADY STATE

major breakthrough in adsorption cooling technology and results in major savings in costs and parasitic power thus making the regenerative zeolite heat pump more competitive in the marketplace.

At this point, the proof of concept of the zeolite heat pump was established successfully. Next, the modeling and design of a 3-ton zeolite heat pump for residential and small commercial use was completed.

Two different modeling methods were used: An analytical model was developed describing the interrelationships among physical properties, heat regeneration and first order approximation of the nonlinear properties. Concurrently, a finite element computer model was developed using numerical solutions to the differential equations describing the system and the computer results confirmed the predictions of the analytical model. Both models were then validated by comparison with the experimental data determined previously and the agreement between the three independently derived results was excellent.

With the confidence gained by the validation of the models, complete performance maps were generated describing the operation of the 3-ton heat pump. The cooling performance map is shown in Figure 12 while Figure 13 represents the heating performance map. These maps were then used for an independent performance evaluation and operating cost analyses for residential and small commercial applications in Dallas, Chicago and Boston. The seasonal heating efficiencies for Chicago and Boston are between 1.45 and 1.6 while seasonal cooling performance factors are about 1.6 in Chicago, 1.8 in Boston and 1.3 in Dallas. The annual operating cost savings were $361 in Chicago, $515 in Boston, and $206 in Dallas resulting in allowed installed system costs for 3 year payback of $4,678 in Chicago, $5,140 in Boston, and $4,152 in Dallas. The projected first cost for the Zeopower heat pump is (double the factory cost of $1,960) about $3,920 resulting in less than 2 year payback in Dallas, less than 1 year in Chicago, and 8 months in Boston. The independent evaluation definitely establishes the economic competitiveness of the zeolite heat pump.

Finally, the overall drawing view of the heat pump is shown in Figure 14 while an artist's rendition of the packaged unit is represented in Figure 15.

3-TON AIR CONDITIONING SYSTEM PERFORMANCE MAP

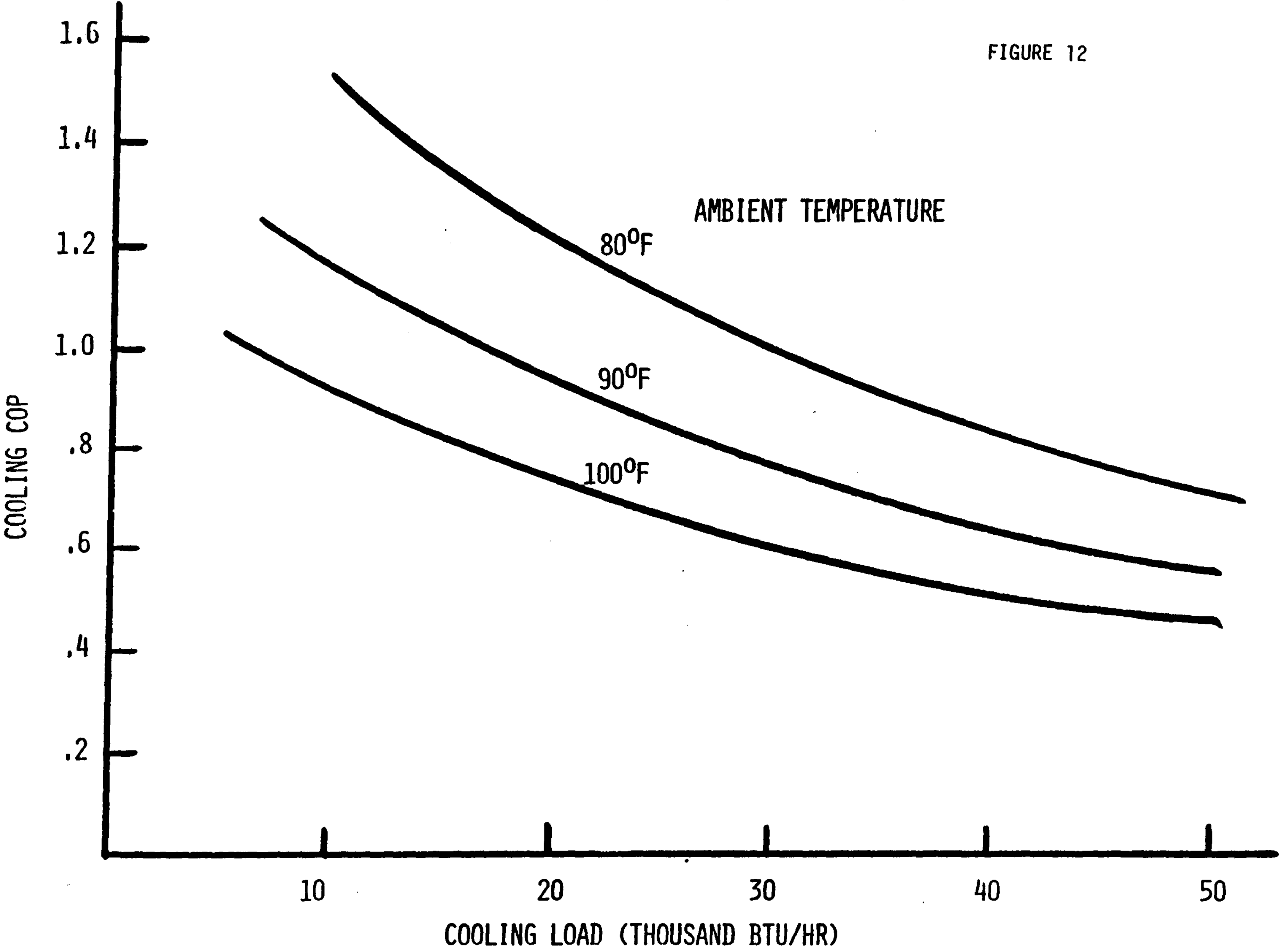
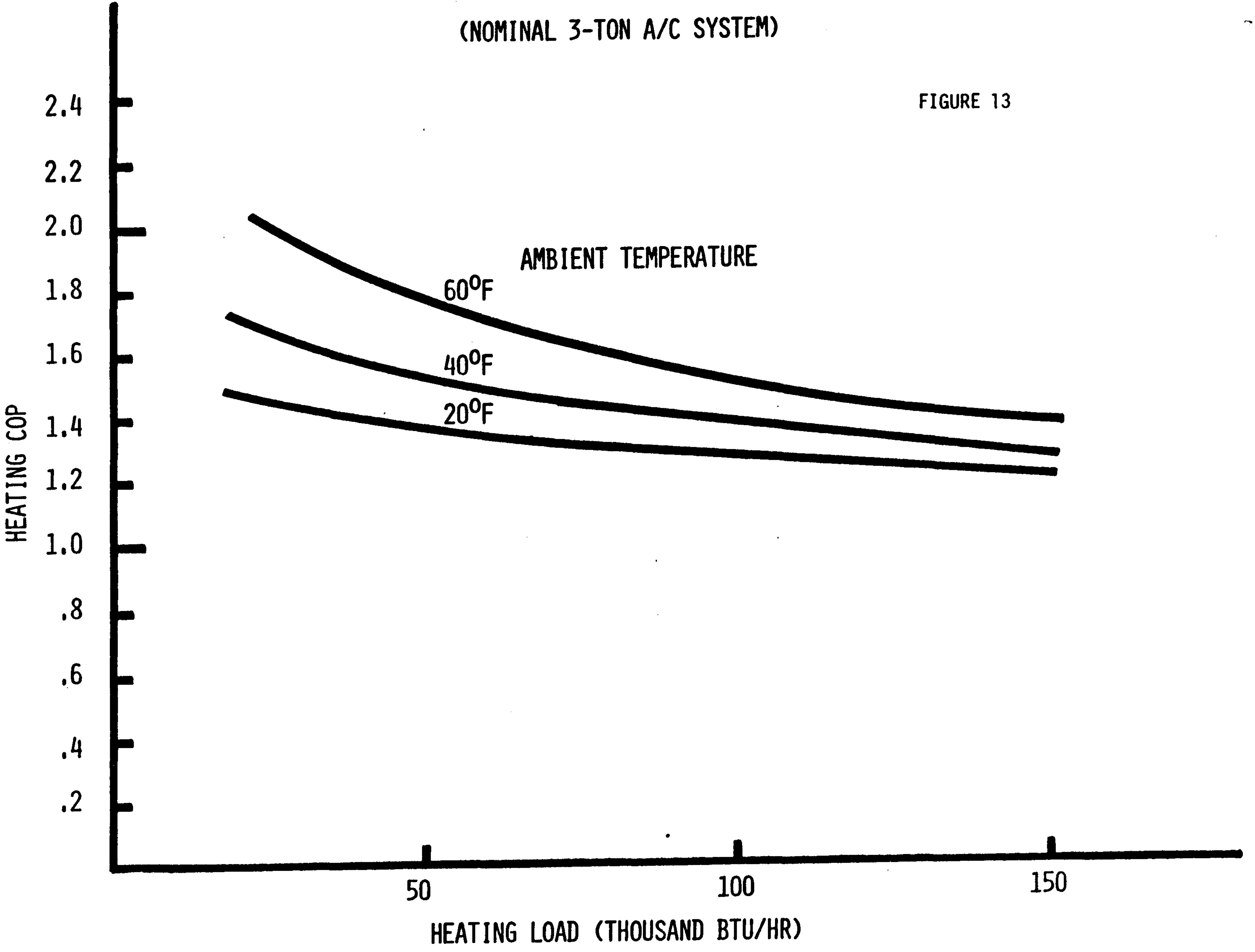


FIGURE 12

HEAT PUMP SYSTEM PERFORMANCE MAP
(NOMINAL 3-TON A/C SYSTEM)

FIGURE 13



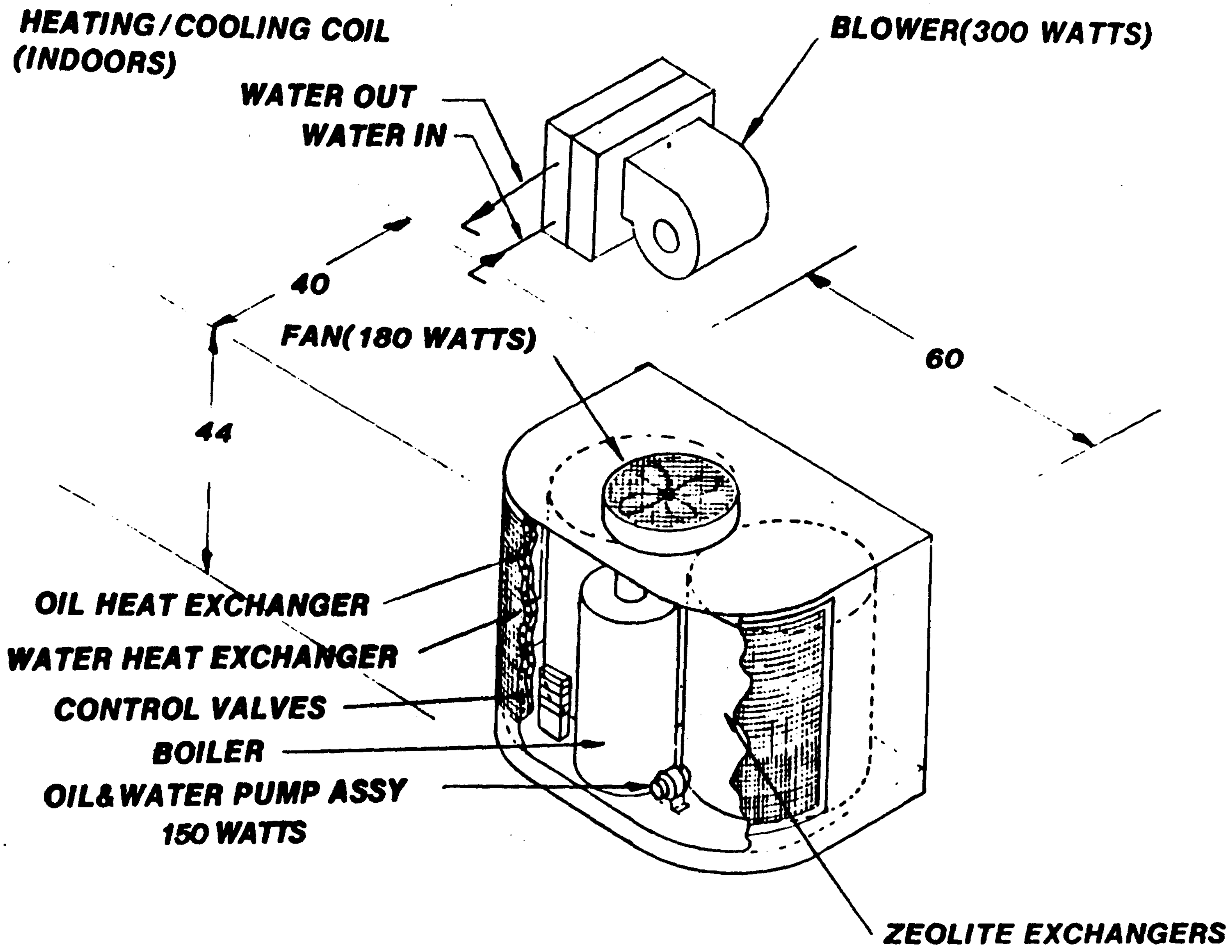
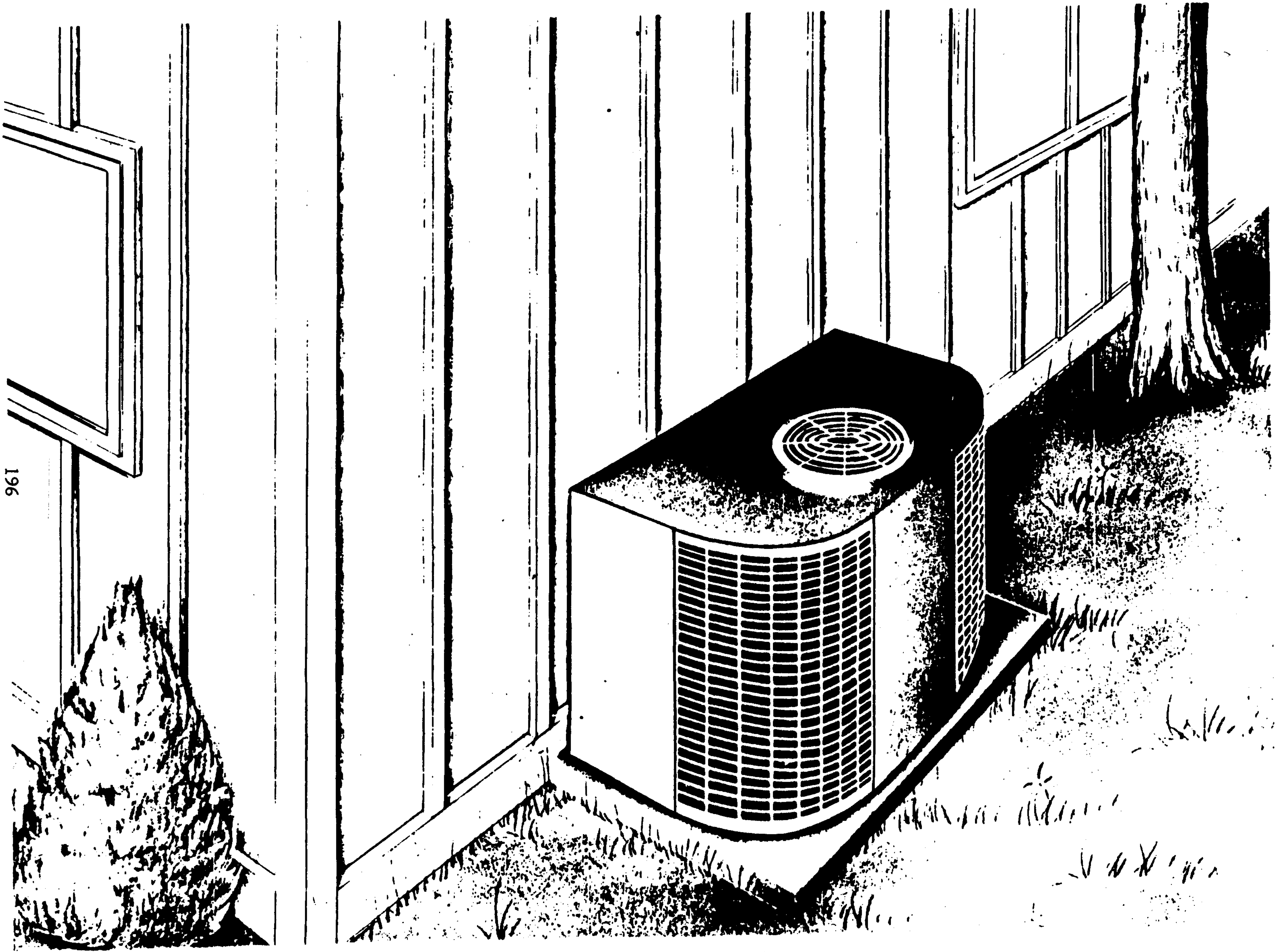


FIGURE 14 ZEOPOWER 3 TON HEAT PUMP (800 LBS TOTAL WEIGHT)



[illegible]