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CENTRAL COOLING - COMPRESSIVE CHILLERS

by

J. E. Christian

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COMMUNITY
ENERGY
SYSTEMS

TECHNOLOGY EVALUATIONS

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Prepared by:

Oak Ridge National Laboratory
Operated by Union Carbide Corporation
for the U. S. Department of Energy

Prepared for:

Argonne National Laboratory
under Contract W-31-109-Eng-38
with the U. S. Department of Energy

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Printed in the United States of America
Available from
National Technical Information Service
U. S. Department of Commerce
5285 Port Royal Road
Springfield, Virginia 22161
Price: Printed Copy $6.00; Microfiche $3.00

ANL/CES/TE 78-2
Special Distribution

CENTRAL COOLING - COMPRESSIVE CHILLERS

by

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March 1978

Prepared for

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by

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Oak Ridge, Tennessee 37830
Operated by
Union Carbide Corporation
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U. S. Department of Energy

The four E's of the cover logo embody the goals of the Community Systems Program of the Department of Energy, DOE, namely:

- to conserve *Energy;*
- to preserve the *Environment;* and
- to achieve *Economy*
- in the design and operation of human settlements (*Ekistics).*

CONTENTS

LIST OF FIGURES

LIST OF TABLES

FOREWORD

The Community Systems Program of the Division of Buildings and Community Systems, Office of Energy Conservation, of the United States Department of Energy (DOE), is concerned with conserving energy and scarce fuels through new methods of satisfying the energy needs of American Communities. These programs are designed to develop innovative ways of combining current, emerging, and advanced technologies into Integrated Community Energy Systems (ICES) that could furnish any, or all, of the energy-using services of a community. The key goals of the Community System Program then, are to identify, evaluate, develop, demonstrate, and deploy energy systems and community designs that will optimally meet the needs of various communities.

The overall Community Systems effort is divided into three main areas: (a) Integrated Systems, (b) Community Planning & Design, and (c) Implementation Mechanisms. The *Integrated Systems* work is intended to develop the technology component and subsystem data base, system analysis methodology, and evaluations of various system conceptual designs which will help those interested in applying integrated systems to communities. Also included in this program is an active participation in demonstrations of ICES. The *Community Planning & Design* effort is designed to develop concepts, tools, and methodologies that relate urban form and energy utilization. This may then be used to optimize the design and operation of community energy systems. *Implementation Mechanisms* activities will provide data and develop strategies to accelerate the acceptance and implementation of community energy systems and energy-conserving community designs.

This report, prepared by Oak Ridge National Laboratory, is part of a series of Technology Evaluations of the performance and costs of components and subsystems which may be included in community energy systems and is part of the Integrated Systems effort. The reports are intended to provide sufficient data on current, emerging and advanced technologies so that they may be used by consulting engineers, architect/engineers, planners, developers, and others in the development of conceptual designs for community energy systems. Furthermore, sufficient detail is provided so that calculational models of each component may be devised for use in computer codes for the design of Integrated Systems. Another task of the Technology Evaluation activity is to devise calculational models which will provide part-load performance and costs of components suitable for use as subroutines in the computer codes being developed to analyze community energy systems. These will be published as supplements to the main Technology Evaluation reports.

It should be noted that an extensive data base already exists in technology evaluation studies completed by Oak Ridge National Laboratory (ORNL) for the Modular Integrated Utility System (MIUS) Program sponsored by the Department of Housing and Urban Development (HUD). These studies, however, were limited in that they were: (a) designed to characterize mainly off-the-shelf technologies up to 1973, (b) size limited to meet community limitations, (c) not designed to augment the development of computer subroutines, (d) intended for use as general information for city officials and keyed to residential communities, and (e) designed specifically for HUD-MIUS needs. The present documents are founded on the ORNL data base but are more technically oriented and are designed to be upgraded periodically to reflect changes in current, emerging, and advanced technologies. Furthermore, they will address the complete range of component sizes and their application to residential, commercial, light industrial, and institutional communities. The overall intent of these documents, however, is not to be a complete documentation of a given technology but will provide sufficient data for conceptual design application by a technically knowledgeable individual.

Data presentation is essentially in two forms. The main report includes a detailed description of the part-load performance, capital, operating and maintenance costs, availability, sizes, environmental effects, material and energy balances, and reliability of each component along with appropriate reference material for further study. Also included are concise data sheets which may be removed for filing in a notebook which will be supplied to interested individuals and organizations. The data sheets are colored and are perforated for ease of removal. Thus, the data sheets can be upgraded periodically while the report itself will be updated much less frequently.

Each document was reviewed by several inividuals from industry, research and development, utility, and consulting engineering organizations and the resulting reports will, hopefully, be of use to those individuals involved in community energy systems.

ABSTRACT

The purpose of this report is to provide representative cost and performance data in a concise, useable form for three types of compressive liquid packaged chillers: reciprocating, centrifugal, and screw. The data are presented in graphical form as well as in empirical equations. Reciprocating chillers are available from 2.5 to 240 tons with full-load COPs ranging from 2.85 to 3.87. Centrifugal chillers are available from 80 to 2,000 tons with full load COPs ranging from 4.1 to 4.9. Field-assembled centrifugal chillers have been installed with capacities up to 10,000 tons. Screw-type chillers are available from 100 to 750 tons with full load COPs ranging from 3.3 to 4.5.

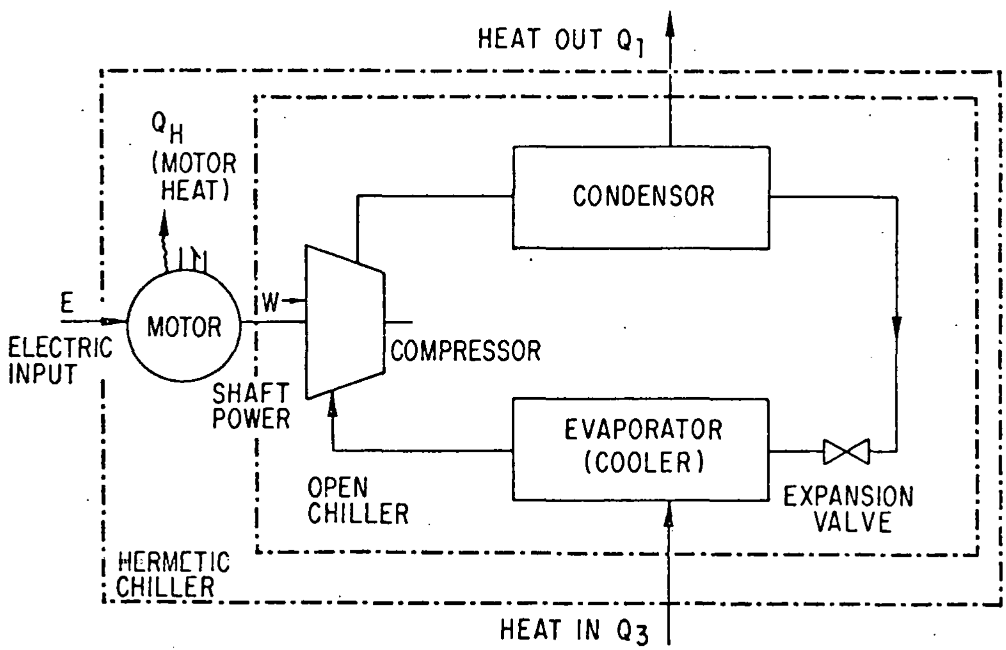
TECHNOLOGY EVALUATION SUMMARY SHEET OF

CENTRAL COOLING - COMPRESSIVE CHILLERS

By: J.E. Christian, ORNL March, 1978

INTEGRATED COMMUNITY ENERGY SYSTEMS

The following evaluation data are presented to help estimate the lifecycle cost* and performance of currently available compressive, packaged chillers. Three types of chillers are covered: reciprocating, centrifugal, and screw-type. The compressors are driven by either open or hermetic-type electric motors. Fig. DS-1 shows a schematic of the compression refrigeration system.



	OPEN CHILLER	HERMETIC CHILLER
COOLING CAPACITY	Q_3	Q_3
HEAT REJECTED	$Q_1 = Q_3 + W$	$Q_1 = Q_3 + W + Q_H$ $= Q_3 + E$

Fig. DS-1 Schematic of the Compression Refrigeration System

*Cost data throughout this evaluation are based on 1976$.

1 INTRODUCTION

All performance data on packaged chillers are based on manufacturers' data. The design operating conditions considered nominal are consistent with the Air Conditioning and Refrigeration Institute Standards (ARI) 590-76 and 550-77; 44°F leaving chilled water; 95°F leaving condenser water; 10°F temperature range across the evaporator and condenser; and 5 x 10^{-4} fouling factor.

The performance and cost data for representative compressive chillers are presented graphically and, as an aid to computer simulation, each graph is modeled empirically by an equation. The equations have been developed with aid of a computerized, unconstrained, unweighted, nonlinear least squares method. Figure DS-2 shows the major control variables and design parameters which affect the performance of the compressive chillers. The performance and cost equations are located in the main body of the compressive liquid chiller report.

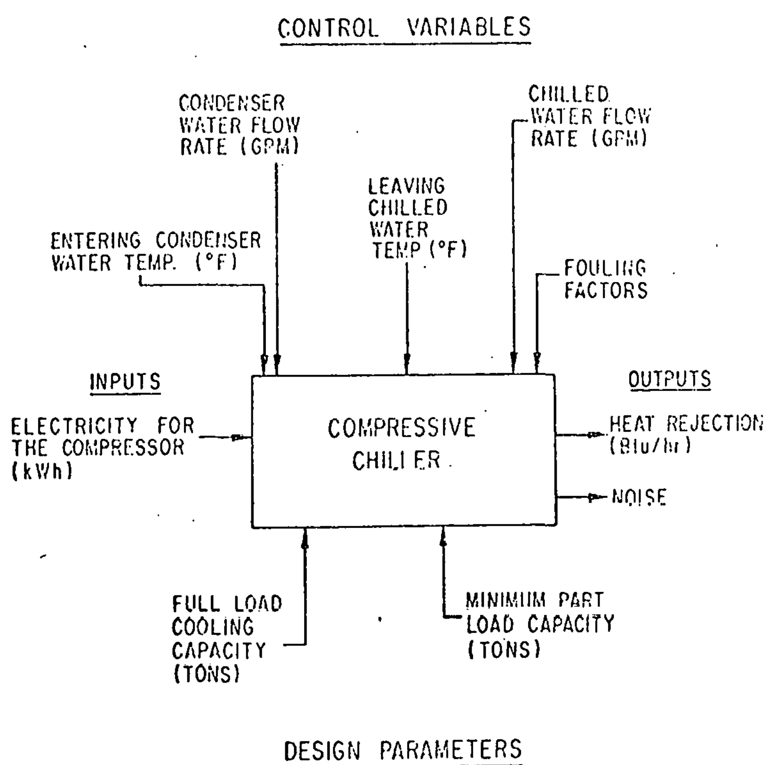


Fig. DS-2 Schematic of Compressive Chiller for Computer Simulation

A summary of the performance data on the three types of packaged chillers follows:

2 RECIPROCATING CHILLERS

2.1 SIZE RANGE (nominal cooling capacity)*

hermetic-water-cooled	10 to 185 tons
hermetic-air-cooled	2.5 to 165 tons
open	10 to 240 tons

2.2 MATERIAL AND ENERGY BALANCE

2.2.1 Full-Load COP

hermetic-water-cooled	3.65
hermetic-air-cooled	2.85
open-water-cooled	3.87
open-air-cooled	3.02

2.2.2 Full Load Performance as a Function of Condensing Temperature and Leaving Chilled Water Temperature

$$COP_R = -186.248 + \frac{36,772}{X} - \frac{1,249,240}{X^2} + 0.675(Y) + 0.00416667(Y^2) \quad \text{(Eq. DS-1)}$$

where:

COP_R = % of nominal COP

X = leaving condenser water temperature (°F) $(85° \leq X \leq 105°F)$

Y = leaving chilled water temperature (°F) $(40° \leq Y \leq 50°F)$

$$CAP_R = -46.2514 + \frac{24,593.8}{X} + \frac{904,779.0}{X^2} - 0.02235(Y) + 0.0441667(Y^2) \quad \text{(Eq. DS-2)}$$

where:

CAP_R = % of nominal cooling capacity

X = leaving condenser water temperature (°F) $(85° \leq X \leq 105°)$

Y = leaving chilled water temperature (°F) $(40 \leq Y \leq 50°F)$

*1 ton is equal to 12,000 Btu/h measured at the evaporator.

2.2.3 Part Load Performance

There are a number of different types of capacity control schemes available for reciprocating chillers. The equation shown below is for a single compressor chiller with suction valve - lift unloading.

$$COP_R = 56.14 + 0.58143\ (X) - 0.0014286\ (X^2) \qquad \text{(Eq. DS-3)}$$

where:

COP_R = % of nominal COP
X = % of full load ($25 \leq X \leq 100$)

2.2.4 Performance as a Function of the Fouling Factor

$$COP_R = 105. - 9000.\ (FF) \qquad \text{(Eq. DS-4)}$$

where:

$$(FF) = \text{fouling factor } (0.0005 \leq FF \leq 0.0025) \qquad \text{(Eq. DS-5)}$$

2.2.5 Chilled Water and Condenser Water Pressure Drop as a Function of Flow

The pressure drop for a representative reciprocating chiller with a chilled water flow of 2.4 gpm/ton and a condenser water flow of 3.0 gpm/ton varies from 7 to 19 ft of head. At water flows other than 2.4 gpm/ton through the evaporator and 3.0 gpm/ton through the condenser, the equation shown below can be used to estimate the corresponding pressure drop.

$$PD = 0.70899 + 0.113974\ (GPM) + 0.00887016\ (GPM)^2$$

where:

PD = % of nominal pressure drop

GPM = % of nominal evaporator or condenser water flow.

2.3 OPERATING REQUIREMENTS

The reciprocating units generally are run with exiting chilled-water temperatures in the range of 40°F to 50°F with a 10°F temperature range across the cooler. The chilled-water flow must be kept between 0.8 and 4 gpm/ton.

3 CENTRIFUGAL CHILLERS

3.1 SIZE RANGE (nominal cooling capacity)

Type	Capacity (tons)
hermetic	80 to 2000
open	90 to 1250
field-assembled, open	700 to 3000

3.2 MATERIAL AND ENERGY BALANCE

3.2.1 Full-Load Nominal COP

Type	COP
hermetic	4.1 to 4.65
open	4.2 to 4.9

3.2.2 Performance at Part Load and Various Condenser and Evaporator Water Temperatures

$$\begin{aligned}\text{\% of nominal power input} = {} & 916.347 + 0.532633(X) - 0.000559686(X^2) + 0.0000230630(X^3) \\ & - 32.7860\,(LCWT) + 0.378447\,(LCWT^2) - 0.00142857\,(LCWT^3) \\ & + \frac{1092.31}{LEWT} + \frac{2071.02}{LEWT^2} \qquad \text{(Eq. DS-6)}\end{aligned}$$

where:

X = % of full load

LCWT = leaving condenser water temperature (°F)

LEWT = leaving evaporator temperature (°F)

3.2.3 Performance as a Function of the Fouling Factor

$$COP_C = 104.0 - 7857.0\ (FF) \qquad \text{(Eq. DS-7)}$$

where:

COP_C = % of nominal COP

FF = fouling factor ($0.0005 \leq FF \leq 0.002$)

3.2.4 Chilled Water and Condenser Water Pressure Drop as a Function of Flow

The nominal pressure drop resulting from a 2-pass condenser and chiller varies between 12 and 25 ft of head. The equation in Section 2.2.5 can be used to estimate the corresponding pressure drop at flows varying from 2.4 gpm/ton for chilled water flow and 3.0 gpm/ton for condenser water flow.

3.3 OPERATING REQUIREMENTS

The centrifugal chiller reportedly operates stably down to 10% of the design load. The leaving chilled-water temperature is usually set at 44-45°F; however, if the humidity load is not excessive, 46-50°F might be satisfactory. The maximum operating range on most centrifugal chillers is between 60 and 80% of the design load. To reduce fouling, manufacturers suggest keeping the water flowrates above 3.3 fps.

4 SCREW-TYPE CHILLERS

4.1 AVAILABLE SIZE RANGE

Type	Capacity (tons)
hermetic	100 - 750 tons
open	500 - 750 tons

4.2.1 Full-Load Nominal COP

Type	COP
hermetic	3.3 - 4.3
open	3.5 - 4.5

4.2.2 Full Load Performance as a Function of Condensing Temperature and Leaving Chilled Water Temperature

$$COP_S = -417.618 + \frac{30581.9}{X} - \frac{670960.0}{X^2} + 10.1833(Y) - 0.0916667(Y^2) \quad \text{(Eq. DS-8)}$$

where:

COP_S = % of nominal COP

X = leaving condenser water temperature (°F)

Y = leaving chilled water temperature (°F)

The percent of nominal cooling capacity is determined as follows:

$$CAP_S = -74.6489 + \frac{10343.9}{X} - \frac{289606.0}{X^2} + 2.77500(Y) - 0.0125(Y^2) \quad \text{(Eq. DS-9)}$$

where:

CAP_S = % of nominal cooling capacity

X = leaving condenser water temperature (°F)

Y = leaving chilled water temperature (°F)

4.2.3 Performance at Part-Load

$$COP_S = 97.1 + 0.36X - 0.0033(X^2) \quad \text{(Eq. DS-10)}$$

where:

X = % of full load ($60 < X \leq 100$)

$$COP_S = 21.07 + 3.26X - 0.03X^2 \quad \text{(Eq. DS-11)}$$

where:

X = % of full load ($10 \leq X \leq 60$)

4.2.4 Performance as a Function of the Fouling Factor

(See Section 2.2.4.)

4.2.5 Chilled Water and Condenser Water Pressure Drops as a Function of Flow

The water pressure drop through the evaporator of one representative line of screw type chillers is 90-150 ft of head and about 160 ft of head through the condenser. For flows other than 2.4 gpm/ton through the evaporator and 3.0 gpm/ton through the condenser, use Eq. DS-12 to adjust the pressure drop.

$$PD = 0.70899 + 0.113974(GPM) + 0.00887016(GPM)^2 \qquad \text{(Eq. DS-12)}$$

where:

PD = % of nominal pressure drop

GPM = % of nominal evaporator or condenser water flow (GPM/ton)

4.3 OPERATING REQUIREMENTS

Screw-type compressor chillers operate down to 10% of the design-load capacity. However, a major stipulation is that the entering condenser water temperature must be at least 20°F greater than the leaving chilled-water temperature.

5 SAFETY REQUIREMENTS

Liquid chillers are covered by a number of safety codes, including those of:

1. *American National Standards Institute,*
2. *ASME Code for Unfired Pressure Vessels,*
3. *National Electrical Code,*
4. *Los Angeles Electrical Code,*
5. *Underwriters' Laboratories,* and
6. *Air Conditioning and Refrigeration Institute.*

6 ENVIRONMENTAL CONCERNS

Two major environmental concerns of central liquid chillers are the thermal discharge and noise. However, water is consumed indirectly by the chillers in wet cooling towers, and some water treatment may be necessary to remove the impurities in condenser water blowdown. A seperate ICES Technoloby Evaluation, titled Heat Rejection, discusses environmental impacts from various types of heat rejection equipment.

7 COST

7.1 F.O.B. COST

The equipment cost for reciprocating packaged chillers can be estimated by using Eq. DS-13, but the resulting cost does not include installation.

$$\$\begin{pmatrix}\text{reciprocating}\\\text{packaged chiller}\end{pmatrix} = 8000\left(\frac{\text{capacity (tons)}}{50}\right)^{.5} \qquad \text{(Eq. DS-13)}$$

The equipment cost for centrifugal and screw compressor packaged chillers can be estimated by using Eq. DS-14. Again, the installation cost is not included.

$$\$\begin{pmatrix}\text{centrifugal or screw}\\\text{compressor chiller}\end{pmatrix} = 54{,}000\left(\frac{\text{capacity (tons)}}{500}\right)^{.66} \qquad \text{(Eq.DS-14)}$$

7.2 INSTALLATION COST

Installation costs tend to be quite site-specific. An appropriate method of budget estimating the installation cost is presented in Sect. 7.2.

7.3 OPERATING AND MAINTENANCE (O&M) COSTS

O&M costs are a function of the size and can be estimated for reciprocating units between 10 and 185 tons by Eq. DS-15 and for centrifugal and screw-type chiller units between 105 and 2000 tons by Eq. DS-16.

$$\begin{matrix}\text{chiller}\\\text{O\&M cost}\end{matrix} = 1180\left(\frac{\text{capacity (tons)}}{50}\right)^{.77} \qquad \text{(Eq.DS-15)}$$

$$\begin{matrix}\text{chiller}\\\text{O\&M cost}\end{matrix} = 4800\left(\frac{\text{capacity (tons)}}{500}\right)^{.42} \qquad \text{(Eq.DS-16)}$$

TECHNOLOGY EVALUATION OF

CENTRAL COOLING - COMPRESSIVE CHILLERS

Prepared by J.E. Christian, ORNL

Date March, 1978

1 INTRODUCTION

1.1 SCOPE

The following technology evaluation contains performance and cost* information on the three major types of available compressive packaged water chillers: reciprocating, centrifugal, and screw-type, details of which will be discussed in Sections 2, 3, and 4, respectively. The 1975 *ASHRAE Guide and Data Book*[1] suggests the following as a rough guide for determining the types of compressors that generally are used in central space-cooling systems for air-conditioning applications:

Up to 80 tons**	-	reciprocating,
80 to 120 tons	-	reciprocating or centrifugal,
120 to 200 tons	-	screw, reciprocating, or centrifugal,
200 to 350 tons	-	screw or centrifugal, and
Above 350 tons	-	centrifugal.

Each of the three types of compressive chillers is available with either an open or a hermetic-type compressor. Open compressors are those in which the shaft extends through a seal in the compressor housing, symbolized by the inner dashed line in Fig. 1.1. Open compressors often are used because of their compatibility with steam turbine, gas turbine, gas engine, or variable motor drives.

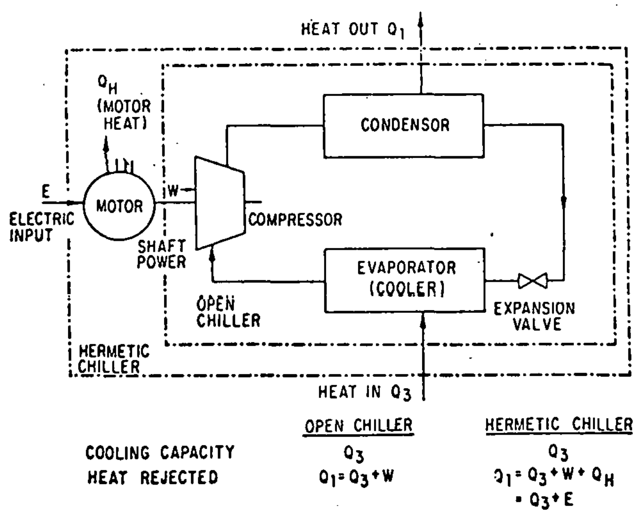


Fig. 1.1 Schematic of the Compression Refrigeration System

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*Cost data throughout this evaluation are based on 1976$.

**One ton is equivalent to 12,000 Btu of heat removed in one hour of steady-state operation.

Hermetic compressors, symbolized by the outer dashed line in Fig.1.1, are characterized by: (1) containment of the motor and compressor within the same pressure vessel, (2) integration of the motor shaft with the compressor shaft, and (3) contact between the motor and the refrigerant. The motor in a hermetic compressor is cooled by the refrigerant; thus, the heat caused by motor inefficiency is included in the total heat rejected by the refrigerant at the condenser. In a hermetic unit, the possibility of refrigerant leakage through a shaft seal is eliminated, and motor operating noise is subdued by the housing. Because forced-refrigerant cooling of the motor is very effective, smaller motors can be used, and the need for a heavy base to maintain motor/compressor shaft alignment is eliminated. Consequently, hermetic machines: (1) are less expensive than open machines, (2) have slightly greater power consumption (than an otherwise identical open model), and (3) operate more quietly. However, in the event of motor failure, the repair cost is higher for a hermetic unit.[2]

1.2 PROCESS DESCRIPTION

Figure 1.1 shows the basic components of the compression refrigeration cycle. In this system, the compressor pressurizes the vapor received from the evaporator; pressurized vapor flows from the compressor to the condenser where heat is rejected to transform the vapor into liquid refrigerant. This liquid flows from the condenser to a receiver (not shown in Fig. 1.1) and is then throttled through an expansion valve just before entering the evaporator. At the evaporator, heat from the return chilled water vaporizes the refrigerant. The vapor then flows to the suction side of the compressor, thereby completing the cycle.

1.3 PERFORMANCE AND COST FUNCTIONS

The cooling capacity and COP of compressive chillers varies as a function of operating conditions, such as cooling load and heat sink (condenser water) temperature. In the following sections, such performance functions are illustrated graphically, and, as an aid to computer simulation, each graph is modeled empirically by an equation. The equations have been developed with the aid of a computerized, unconstrained, unweighted, nonlinear least squares method. Figure 1.2 illustrates the major control variables and design parameters which affect the chiller performance.

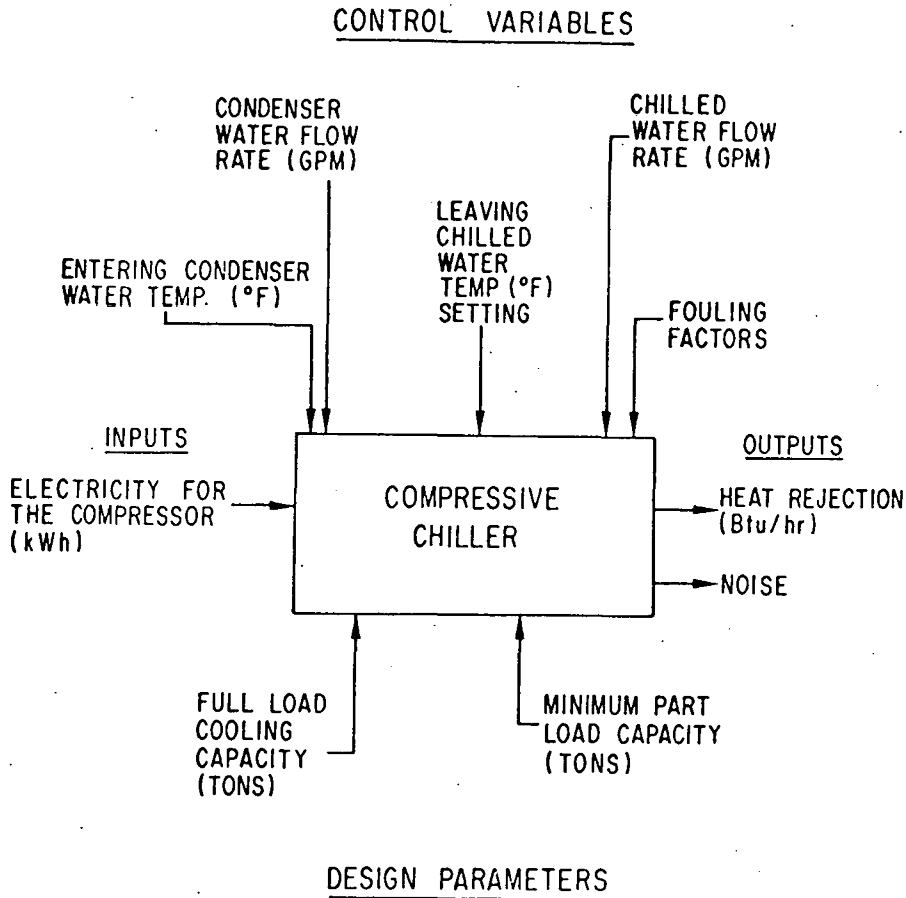


Fig. 1.2 Schematic of Compressive Chiller for Computer Simulation

For those functions with single independent variable dependents, a polynominal in the form of Eq. 1.1 is used.

$$Y= A + BX + CX^2 + DX^3 \qquad \text{(Eq. 1.1)}$$

The performance data is normalized to the nominal design conditions for reciprocating centrifugal and screw types chillers wherever possible. The dependent value (Y) is usually given as a percentage of the nominal condition. Thus, the absolute values of cooling capacity or COP can be found by multiplying Y/100 by the nominal values suggested in this report or provided by manufacturers.

2 RECIPROCATING PACKAGED LIQUID CHILLERS

2.1 DESCRIPTION

A reciprocating compressive chiller maintains a fairly constant capacity over a wide range of pressure ratios and therefore retains nearly full cooling capacity even at operating conditions with above-design wet-bulb ambient air temperatures. It is well suited for air-cooled condenser application. Three basic components make up a packaged liquid chiller. These are described below:

• *Compressors* of three types commonly are used in reciprocating chillers:

1. welded hermetic,
2. semihermetic, and
3. direct-drive open.

• *Condensers*. There are three types of condensing units; evaporative, air or water cooled. For smaller reciprocating chillers, air-cooled condensers are usually selected due to lower maintenance cost than both evaporative and water cooled condensers. The higher maintenance cost for evaporative and water cooled condensers results from makeup water fees, chemical water treatment, and more costly heat exchanger surface cleaning.

In reciprocating chiller applications, the higher maintenance cost of evaporative and water cooled condensers may be offset by reduction in electrical power costs. The air cooled condensers operate at temperatures approaching the ambient dry bulb temperature; whereas evaporative and water cooled condensers operate at lower temperatures approaching the ambient wet-bulb. As a result, air-cooled units operate at a higher pressure and therefore, require more power.

• *Evaporators (Coolers)* usually use direct expansion in which refrigerant evaporates while flowing inside tubes, and chilled liquid is cooled as it is guided several times over the outside of the tubes by shell side baffles.

2.1.1 Available Size Ranges

The capacity of a reciprocating packaged chiller is discussed in terms of the cooling output at design conditions as specified by *Air-Conditioning and Refrigeration Institute (ARI)* Standard 590-76 in nominal tons.* ARI Standard 590-76 specifies the following conditions which are taken to be the nominal design conditions:

- 44°F leaving chilled-water temperature;
- 2.4 gpm/ton, chilled-water flowrate;
- 95°F leaving condenser water temperature;
- 120°F leaving condenser air temperature for air-cooled units;
- 10°F temperature drop across both the evaporator and water-cooled condenser (temperature range); and
- 0.0005 fouling factor on the water side of both the evaporator and condenser.

The hermetic reciprocating packaged chillers are available in nominal capacities up to about 185 tons with a water-cooled condenser and 165 tons with a remote air-cooled condenser.

Open-drive type reciprocating packaged chillers are available in nominal capacities ranging from 50-240 tons. Completely contained pad-mounted, remote condensing unit, and remote condenser packages are all available for air-cooled chillers. Unitary and built-up chillers are available for water-cooled units; built-up systems are needed for evaporative-cooled units.

*1 ton is equivalent to 12,000 Btu of heat removed in one hour of steady-state operation.

2.1.2 Dimensions and Shipping Weight

Typical space requirements and shipping weights for a variety of reciprocating packaged chillers are shown in Table 2.1. The space requirements are shown with and without a minimum manufacturer-recommended service access area. The air-cooled reciprocating units which utilize a remote air-cooled condenser have approximately the same dimensions as the water-cooled units. However, the remote heat dissipating equipment for both the air- and water-cooled units is considered as separate from the packaged chiller unit itself, and therefore its space requirements are not reflected in Table 2.1.

Table 2.1 Reciprocating Packaged Chiller Dimensions and Shipping Weight

Nominal Capacity (tons)	Length x Width x Height Chiller only (ft)	Length x Width x Height Including Service Area (ft)	Weight (lb) Air-Cooled	Weight (lb) Water-Cooled
10	6.1 X 2 X 3	7 X 6 X 4	1010	1210
20*	10 X 5.2 X 3.7	14 X 8 X 4	3650	
50	9 X 3 X 4	15 X 6 X 5	2660	3215
95*	19.5 X 7.3 X 5.5	30 X 13 X 7	7710	
100	9.4 X 3 X 5	17 X 6 X 5	3385	4400
150	9.4 X 3 X 4.8	18 X 6 X 5	7010	8090
200	10.0 X 3.2 X 4.8	19 X 6.5 X 5	7880	8920

*Mounted as a package on the roof; includes the air-cooled condenser.

2.1.3 Electrical Requirements

Reciprocating chiller motors are available in standard ac voltages of 208, 230, 460, and 575 three-phase, 60-hz. The reciprocating chiller controls generally require 115 V-single phase, 60-hz.

2.1.4 Refrigerant Charge

The refrigerant charge used to provide the reference performance data is R-22, which is the most common refrigerant specified for reciprocating chillers. Adjustment factors for refrigerants 12 and 500 are available from most reciprocating chiller manufacturers.

2.2 MATERIAL AND ENERGY BALANCE

The two major inputs to a reciprocating compressor chiller are the electrical power required to operate the compressor motor, and the thermal energy withdrawn from the conditioned space. The major output is the thermal energy transferred to the condenser water or directly to the surrounding air in the case of air-cooled condensers.

The coefficient of performance (COP) is commonly used to compare the performance of refrigeration systems. The COP is defined as follows:

$$COP = \frac{Q_3}{E_{th}} \quad \text{(Eq.2.1)}$$

where:

Q_3 = useful refrigeration effect produced at stated conditions not accounting for the cooling capacity loss during distribution from the chiller to the conditioned space in Btus, and

E_{th} = heat equivalent of the total energy input rate required to operate the system, Btus.

2.2.1 Nominal Full-Load Performance

A survey of manufacturers' listed COP values of various size hermetic type reciprocating compressor packaged chillers at nominal full load operating conditions (as specified by ARI standard 590-76) is shown in Fig. 2.1.[3,4,5].

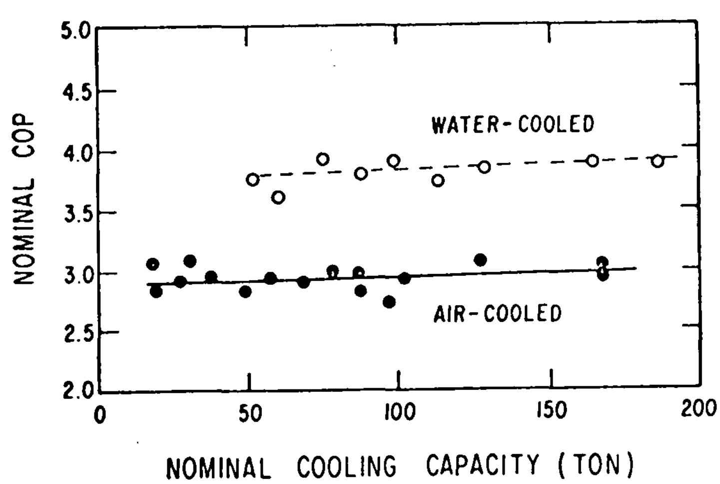


Fig. 2.1 Full-Load COP of Hermetic Type Reciprocating Packaged Chillers

the solid line through the lower series of data points in Fig. 2.1 indicates the full-load COP for air-cooled, hermetic type reciprocating chillers is about 2.85. The dashed line through the higher series of data points in Fig. 2.1 indicates that the full-load COP for hermetic type water-cooled reciprocating chillers is about 3.65. The COP values shown in Fig. 2.1 do not include the electrical input for heat rejection fans or chilled-water or condenser-water pumps. The water-cooled reciprocating packaged chillers averages about a 30% higher COP at full-load conditions than the same reciprocating chiller package coupled to an air-cooled condenser. However, by selecting a slightly larger air-cooled condenser, the COP difference between the two units could be reduced.

An open-drive reciprocating packaged chiller shows about a 5-8% higher COP since the compressor motor is cooled by the air surrounding the machine. However, the open-drive units cost more initially, and for this reason they have become less popular.[1]

2.2.2 Full-Load Performance at Various Condenser and Evaporator Water Temperatures

The full load COP and capacity of reciprocating chillers are dependent on both the condensing temperature and chilled water temperature. Figure 2.2 shows the relationship between the leaving condenser and leaving chilled water temperature values, and the COP and full load capacity values.

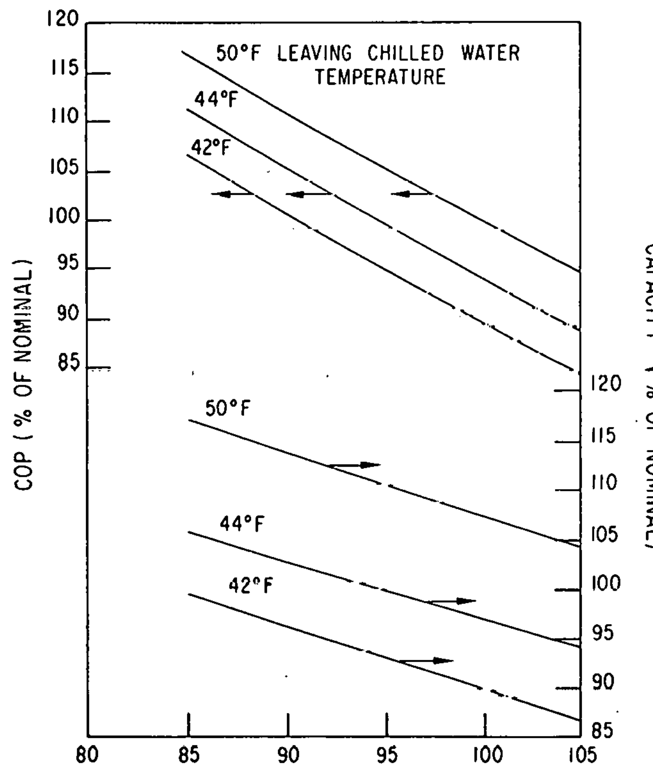


Fig. 2.2 Reciprocating Chiller Nominal COP and Full Load Capacity at Various Leaving Condenser and Leaving Chilled Water Temperatures

The values are representative of a number of different manufacturers' reciprocating chiller performance data.[3,4,5] Figure 2.2 can also be applied to air-cooled reciprocating chillers by using the leaving condenser water temperature as the ambient dry bulb temperature on the condenser.

The chilled water flow is fixed at ARI Standard 550-76-specified 2.4 gpm/ton, which is equal to a 10°F chilled water range. However, the same performance as shown in Fig. 2.2 can be assumed to result from various combinations of chilled water flow and range varying from 6 to 14°F. The cooling capacity at various leaving condensing and chilled water temperatures are related to the chilled water flow and range as shown below:

$$\text{Cooling Capacity (tons)} = \frac{\text{GPM x chilled water range}}{24} \qquad \text{(Eq. 2.2)}$$

Most manufacturers suggest that once the design GPM is established it should be maintained whenever the compressor is operating. Failure to do so may result in localized freezing.

The condenser water flow for the reciprocating chiller performance data shown in Fig. 2.2 varies within $\pm$ 5% of 3 gpm/ton of cooling capacity. The condenser water temperature is assumed to rise 10°F. Thus to figure the exact condenser water flow, the total heat rejection must first be estimated as shown in the following two equations.

$$\text{Heat Rejection (MBH)} = \text{Tons x 12 + kW x 3.413} \qquad \text{(Eq. 2.3)}$$

$$\text{GPM condenser water} = \frac{\text{MBH x 2}}{\text{Condenser water range}} \qquad \text{(Eq. 2.4)}$$

The fouling factor assumed in arriving at the curves shown in Fig. 2.2 is .0005.

The empirical equation developed to estimate the percent of nominal COP as a function of the leaving condenser water temperature and leaving chilled water temperature is shown in Eq. 2.5.

$$COP_R = -186.248 + \frac{36,772}{X} - \frac{1,249,240}{X^2} + 0.675(Y) + 0.00416667(Y^2) \qquad \text{(Eq. 2.5)}$$

where:

COP_R = percent of nominal cooling capacity

X = leaving condenser water temperature (°F)

Y = leaving chilled water temperature (°F)

The standard error for Eq. 2.5 is 0.676.

The empirical equation developed to estimate the percent of nominal cooling capacity as a function of the leaving condenser water temperature and leaving chilled water temperature is shown in Eq. 2.6:

$$CAP_R = -46.2514 + \frac{24,593.8}{X} - \frac{904,779}{X^2} - 2.235(Y) + 0.0441667\ (Y^2) \qquad \text{(Eq. 2.6)}$$

where:

CAP_R = percent of nominal cooling capacity

X = leaving condenser water temperature (°F)

Y = leaving chilled water temperature (°F)

The standard error for Eq. 2.6 is 0.927.

2.2.3 Part Load Performance

Figure 2.3 shows a variety of part load performance curves for reciprocating chillers with various capacity control alternative.[6,7] The curves shown are approximations to the actual part load performance since actual COP versus load is a step function not continuous as displayed in Fig. 2.3.

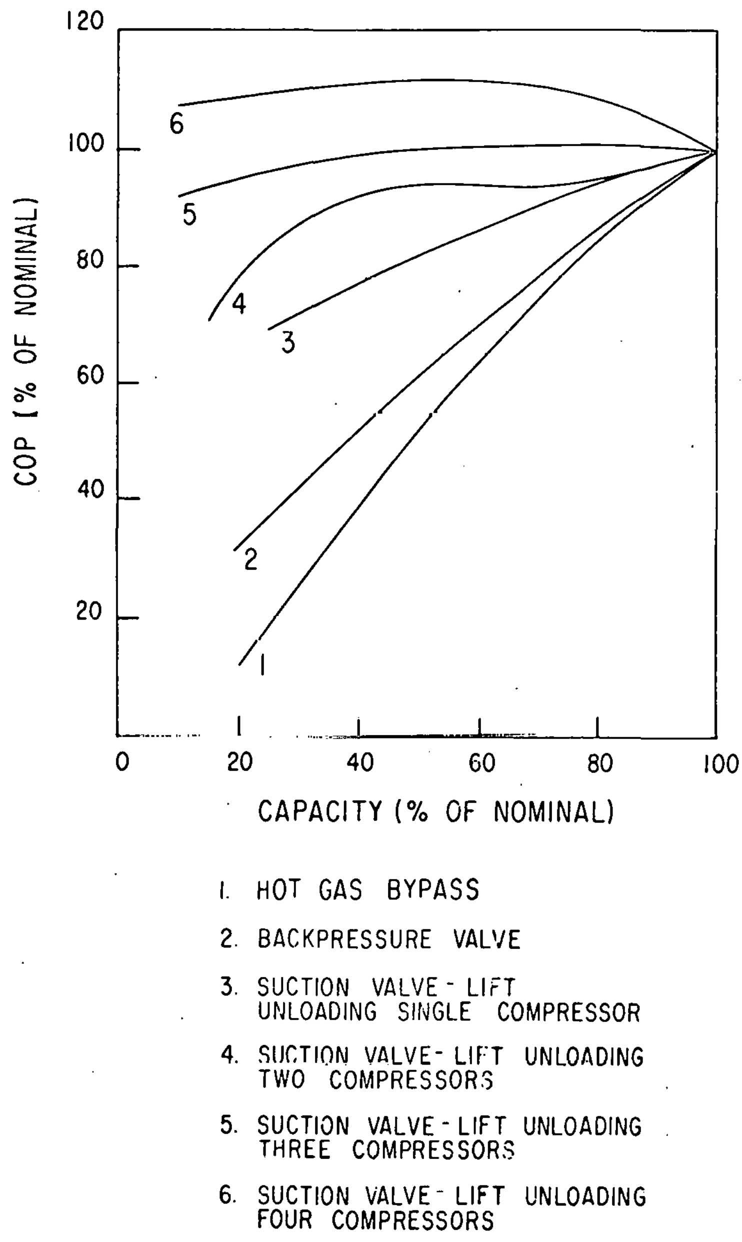


Fig. 2.3 Part-Load COP of a Variety of Reciprocating Package Chiller with One, Two, Three, and Four Compressors per Package

There are six curves shown in Fig. 2.3 each with a corresponding set of generalized equation coefficients listed in Table 2.2.

Table 2.2 Generalized Equation Coefficients for Various Reciprocating Chiller Capacity Control Alternatives - Nominal COP (Y) Versus Percentage of Nominal Capacity (X)

Capacity Control	Range of (X)	Coefficients A	B	C	D
1. Hot Gas Bypass	20 < x < 100	-2.83	1.181	-0.00153	
2. Back Pressure Valve	20 < x < 100	20.56	0.7144	0.0008	
3. Cylinder-head Bypass Single Compressor	25 < x < 100	56.14	0.58143	-0.0014286	
4. Cylinder-Head Bypass Two Compressors	15 < x < 100	37.5	2.75	-0.043125	0.00021875
5. Cylinder-Head Bypass Three Compressors	10 < x < 100	92.28	0.162857	-0.0008571	
6. Cylinder-Head Bypass Four Compressors	10 < x < 100	105.72	0.282143	-0.0033929	

A brief description of each method of capacity control is given below:

1. *Hot Gas Bypass* - Loads the compressor artificially by transferring heat to the suction gas. The hot gas is allowed to pass from the high pressure side of the vapor compressive cycle (bypassing the condenser) to the low side through a constant pressure valve. As the evaporator pressure tends to drop with lower cooling capacity demand the valve opens up, thus maintaining a constant suction pressure. The major disadvantage of this type of control as shown by curve number 1 in Fig. 2.3 is that the COP drops rapidly at part-load operating conditions.

   The major advantages of this control are (a) permits a constant speed compressor to operate at lower capacities with out cycling on and off, and (b) helps level the 8-10°F leaving chilled water temperature fluctuations resulting from other capacity controls.

2. *Back-Pressure Valve* - On multi-cylinder compressors, one or more cylinders can be made ineffective by allowing gas to pass through the inactive cylinders but not allowing compression to take place. There are still some pressure losses through the valves, cylinders, and connections, and because of these losses the COP drops as shown in Fig. 2.3 at part load operating conditions.

3. *Suction-Valve-Lift Unloading* - This unloading is accomplished by unseating the suction valves of certain cylinders in the compressor so that compression cannot take place. This is the most efficient method of capacity control since passage of the refrigerant vapor in and out of the cylinder through the suction valves without compression involves smaller losses than other methods.

4,5,&6. *Multiple Compressor Units Plus Suction Valve - Lift Unloading* - Thermostats or pressurestats may be used to both start and stop compressors, and load and unload cylinders in accordance with load demands.

This capacity control system has many advantages:

(a) Relatively high part load COP as shown by Fig. 2.3;

(b) Single-speed motors may be selected and operated continuously at their best efficiency;

(c) An individual reciprocating chiller package with multiple compressors has inherent standby equipment which allows part-load capacity if one of the machines breaks down.

(d) Compressors may be started in sequence to limit the current enrush if time delay devices are employed.

Multiple compressor units with two compressors are available starting with about 20 tons of nominal cooling capacity, with three compressors-70 tons and with four compressors-100 tons.

In pracice, a good control system should minimize the on-off cycling frequency while maintaining a satisfactory temperature control and a high COP at all partial load operations. During each compressor startup, oil leaves the crankcase at an accelerated rate. Therefore, all capacity control methods should avoid compressor running cycles of less than seven or eight minutes.[7]

To determine the reciprocating chiller COP at part load conditions and various leaving condenser and leaving chilled water temperatures, Fig. 2.2 can be used in conjunction with Fig. 2.3. First find the corresponding percent of nominal COP from Fig. 2.2 (Eq. 2.5) at the desired chilled water and condenser water temperatures. Then find the percent of nominal COP from Fig. 2.3 (Table 2.2) corresponding to the desired part load. The COP as a function of part load, leaving chilled water temperature and leaving condenser water temperature is:

$$COP = COP_{2.3} \times \frac{COP_{2.2}}{100} \quad \text{(Eq. 2.7)}$$

where:

COP = % of nominal COP

$COP_{2.3}$ = % of nominal COP from Fig. 2.3

$COP_{2.2}$ = % of nominal COP from Fig. 2.2

2.2.4 Performance as a Function of the Fouling Factor

The fouling factor is used to estimate the decrease in heat transfer capacity after the equipment has been in service for some time and dirt and scale deposits have increased the heat transfer surface resistances. Chiller manufacturers use a fouling factor of .0005 which is conventionally arrived at by assuming the water pumped through the heat exchangers is of good quality and is recirculated in a closed-loop system. The *ASHRAE Equipment Handbook*[1] recommends a fouling factor of .001 when the recirculated water is in an open system. The decrease in performance resulting from a higher fouling factor is by rule of thumb equivalent to raising the leaving condenser water temperature 2.5°F for every .0005 increase in the fouling factor.

The use of untreated condenser water results in higher fouling factors in some cases as indicated by Table 2.3.[8]

Table 2.3 Fouling Factors for Heat Transfer Surfaces of Various Types of Untreated Circulating Water*

Water	Fouling Factor
Sea water	.005
Brackish water	.001
Cooling tower and artificial spray pond:	
Treated makeup	.001
Untreated	.003
City of well water (such as Great Lakes)	.001
River water:	
Minimum	.001
Mississippi	.002
Engine jacket	.001
Distilled	.0005

*Temperature of the heating medium less than 240°F, temperature of water less than 125°F, water velocity greater than 3fps.

Figure 2.4 shows the estimated percentage reduction of the nominal COP at fouling factors from .0005 to .0025 for either the evaporator or condenser.

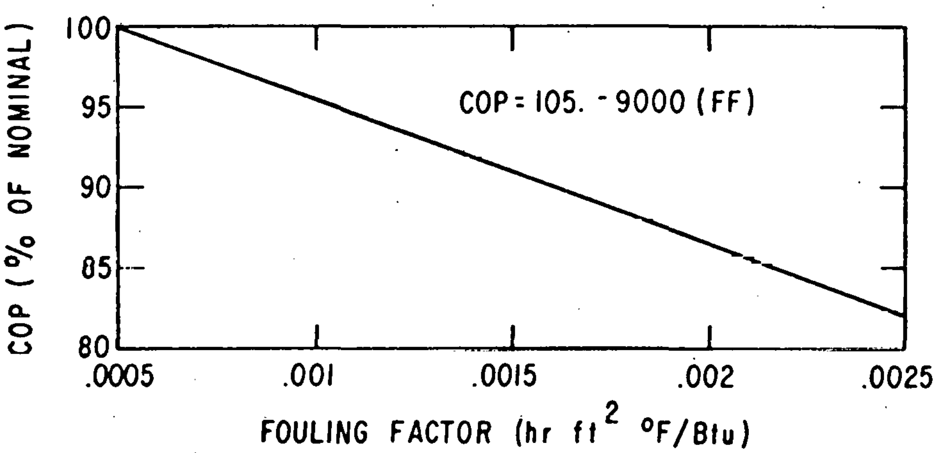


Fig. 2.4 Reduction of a Reciprocating Compressor Package Chiller COP as a Function of the Condenser or Evaporator Assumed Fouling Factor

If both the condenser and the evaporator have fouling factors above .0005 the overall reduction in COP can be estimated by first finding the reduction factor due to fouling of one of the components then using Fig. 2.4 again to find the reduction factor due to the fouling of the other component and multiplying both factors and the nominal COP together and dividing by 10,000 to obtain the reduced COP.

To obtain the rated performance of a packaged chiller assuming a .0005 fouling factor when you have water quality which you know will probably give you a fouling factor of .001, many engineers simply specify a larger heat exchanger surface. The reduction in the nominal COP can be mathematically represented by Eq. 2.8.

$$COP = 105.0 - 9000\ (FF) \qquad \text{(Eq. 2.8)}$$

where:

COP = % of nominal COP, and

FF = fouling factor ($0.0005 \leq FF \leq 0.0025$).

2.2.5 Chilled Water and Condenser Water Pressure Drop as a Function of Flow

The water pressure drop through reciprocating chiller evaporators and condensers varies from machine to machine. However, for rough estimates pump sizes, Fig. 2.5 is provided to guide the designer in approximating the pressure drop through standard size evaporators and condensers with varying water flowrates.[3,4,5]

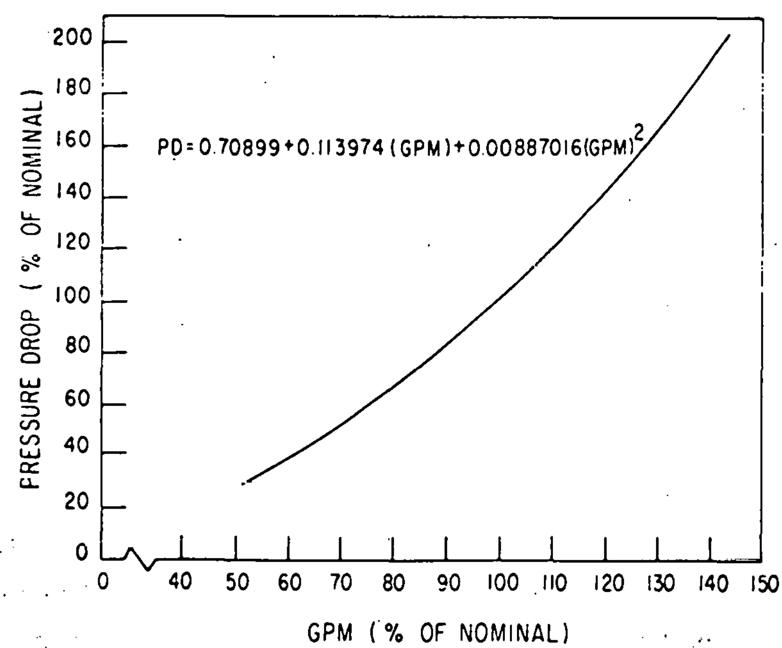


Fig. 2.5 Representative Evaporator and Condenser Water Pressure Drop Versus Flow

Figure 2.5 is a graph with the percent of nominal flow along the vertical axis and percent of nominal pressure drop along the horizontal axis. The nominal flow through the evaporator is 2.4 gpm/ton. The nominal flow through the condenser is 3.0 gpm/ton. - The corresponding pressure drop for the nominal flows varies from 7 to 19 ft of head.

2.2.6 Auxiliary Electrical Inputs

A few reciprocating package chillers include the air-cooled condenser within the preassembled package. These units are most commonly installed on the roof or on concrete pads near the building. Fig. 2.6 shows an estimated electrical power demand of the electric fan motors operating at full load.

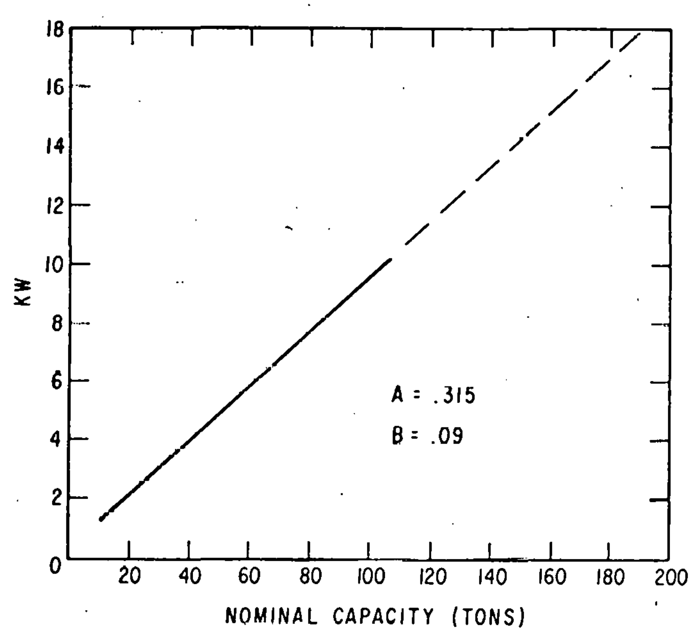


Fig. 2.6 Auxiliary Electric Power Required by the Fans on Air-Cooled Reciprocating Packaged Chillers

The curve was derived by converting the specified required fan motor brake horsepower into kW and then assuming the fan motors were 70% efficient. The electric power requirement of water cooling towers and remote air-cooled condensers is not estimated in this section but will be addressed in a seperate ICES evaluation on heat disposal technology.

2.3 OPERATING REQUIREMENTS

2.3.1 Minimum and Maximum Operating Conditions

Generally the minimum allowable exiting chilled water temperature is set at 40°F and the maximum at 50°F. The minimum chilled water flowrate is .8 gpm/ton and the maximum flowrate is around 4 gpm/ton. The nominal ARI Standard 590-76 chilled water flowrate specified is 2.4 gpm/ton.

2.3.2 Multiple-Unit Application

The reciprocating package chillers can be used in series or parallel operation and with constant and variable water flow. For air-conditioning purposes, paralleling units are more common than series application. The kinds of control logic with multiple units of various capacities are numerous.[6]

2.4 MAINTENANCE AND RELIABILITY

2.4.1 Maintenance Requirements

- *General.* The periodic inspection and maintenance checks should be carried out by a qualified refrigeration service mechanic. It is most important that all controls are set correctly to protect the equipment against improper operation.

When the unit is to be out of service for a prolonged period of time, it should be completely drained of water if it happens to be located where a freezing temperature can be encountered for even a short period of time.

- *Chiller and Water-Cooled Condenser Cleaning.* Depending on the water quality, the water sides of shell and tube units should be cleaned with chemicals to improve the heat transfer abilities. The scale which builds up on the heat-transfer surface is composed, in most cases, of magnesium and calcium carbonates, sulphates, and other impurities.

The suggested method for cleaning is to circulate the cleaning agent through the unit, preferably using a special pump and tank circuit set up for reverse flushing. This is followed by a fresh water flushing and finally circulation of a neutralizing solution.

- *Air-Cooled Condensers.* The face of the condenser should be cleaned at least once a month during operation, and if condensers pick up dirt very quickly, it is suggested that they be cleaned more frequently. If a condenser is allowed to get too dirty, the machine will run a high head pressure and will not give satisfactory performance.

The most costly breakdown of a reciprocating chiller is to have a burnout of a hermetic compressor motor. In most cases, the cause of a hermetic motor burnout is impurities in the refrigerant system. The impurities may be in the form of dirt, scale, or moisture left in the system at the time of installation.

Field servicing of a system after a hermetic motor burnout is generally a time-consuming and expensive operation. Not only must the compressor be repaired, but the entire system must be thoroughly cleaned of all harmful contaminants left by the burnout. Repeated burnouts generally indicate inadequate system cleanout after the previous failure.

2.4.2 Economic Life

The economic life of a reciprocating packaged chiller is estimated to be 13 years[4] for smaller units and about 20 years for units greater than 15 tons.[1]

2.4.3 Reliability

All reciprocating packaged chillers are factory assembled and are tested before shipment. Some reciprocating packaged chillers have 2 or 3 separate refrigerant circuits, each with its own compressor, evaporator, and condenser. This permits an element of standby capacity if a component in one of the refrigerant circuits prematurely fails.

3 CENTRIFUGAL PACKAGED LIQUID CHILLERS

3.1 DESCRIPTION

The centrifugal chiller unit shown in Fig. 3.1 consists basically of a centrifugal compressor, an evaporator, and a condenser. The compressor uses centrifugal force to raise the pressure of a continuous flow of refrigerant gas from the evaporator pressure to the condenser pressure. The evaporator is usually a shell-and-tube heat exchanger with the refrigerant in the shell side. The condenser is usually a shell-and-tube type which uses water as a means of condensing, although, air-cooled or evaporative condensers are becoming more popular in areas with water availability limitations or water quality problems. The compressor in Fig. 3.1 is of the hermetic type in which gas flows through the electric motor winding to the suction side of the compressor impellers. This section will cover both hermetic and open type centrifugal packaged chillers.

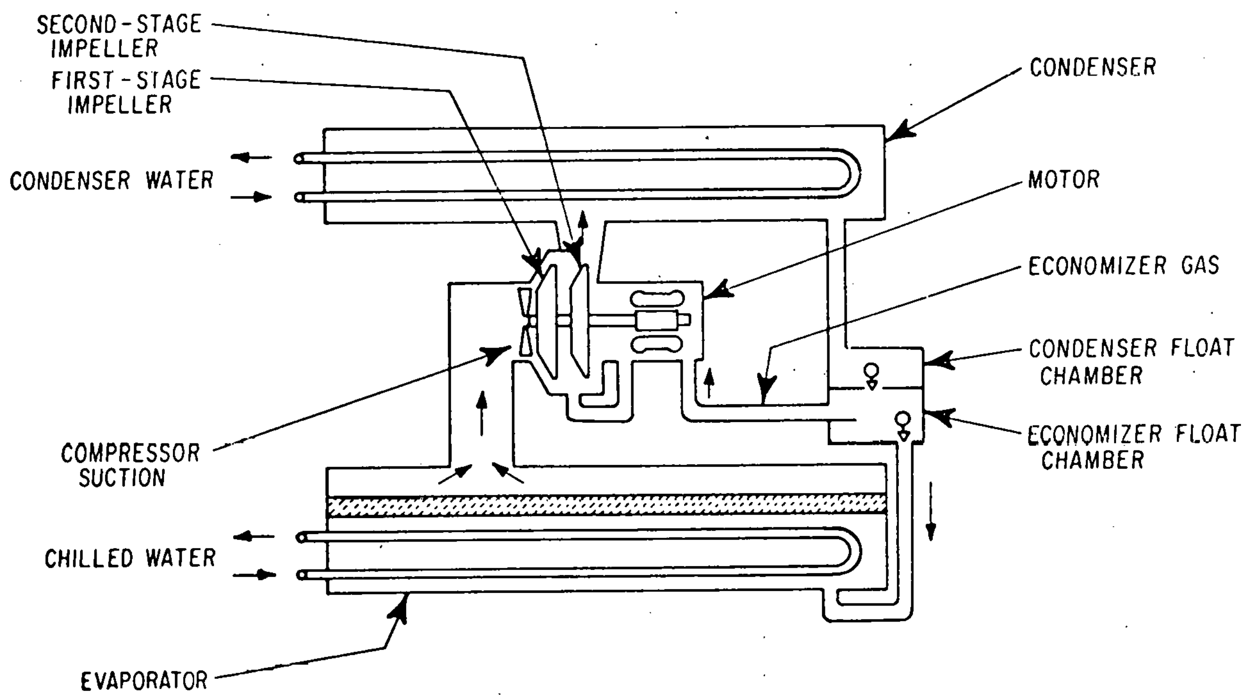


Fig. 3.1 Basic Components of Centrifugal Refrigeration System

The hermetic and some smaller open-type centrifugal chillers are one piece, factory assembled units, shipped ready for field-charging, and connection to water and electrical sources.

Large, open-type centrifugal chillers are shipped disassembled for field erection, piping, and wiring, consisting of either single- or double-stage centrifugal compressors suitable for electric motor, steam turbine, gas engine, diesel engine, or gas turbine drive, a shell-and-tube refrigerant condenser, a flash-type intercooler, a drum-type purge unit, a refrigerant transfer unit and pumpout receiver, a pneumatic control center, and associated interconnecting refrigerant.

3.1.1 Manufacturers and Available Size Ranges

The size ranges of chillers are classified in nominal tons. A nominal ton, being refrigerating capacity equivalent to 12,000 Btu/h, measured at operating conditions consistent with ARI Standard 550-77:

44°F leaving chilled-water temperature,

2.4 gpm/ton chilled-water flowrate,

95°F leaving condenser water temperature,

3.0 gpm/ton condenser water flowrate,

.0005 fouling factor on the water side of both the evaporator and condenser.

The hermetic centrifugal, packaged chillers currently are available in nominal capacities ranging from 80 to 2000 tons.

Open-drive centrifugal packaged chillers are available in nominal capacities ranging from 90 to 1250 tons, and large field-assembled, open-drive centrifugal chillers are available in sizes ranging from 700 to 3000 tons.

According to one manufacturer, multi-stage compressor models extend the range to 10,000 tons, although the models described in this section cover centrifugal chillers up to 2000 tons. Table 3.1 shows a partial list of major manufacturers of centrifugal compression chiller equipment and their available size range.

Table 3.1 Partial Listing of Major Centrifugal Compression Chiller Manufacturers

Company	Nominal Capacity Range (tons)
Single-Stage hermetic packaged chillers	
Carrier	90 - 450
Westinghouse	161 - 550
York	90 - 1250
Single-Stage, open-type packaged unit	
York	90 - 1250
Two-Stage hermetic packaged chillers	
Trane	80 - 1290
Carrier	450 - 2000
Two-Stage open-type field assembled	
Carrier	100 - 10000
York	700 - 8500
Trane	1800 - 5000

3.1.2 Electrical Requirements

The electrical inputs suitable for centrifugal compressor motors are 200-225, (4 conductor), 230 (220-240, 3 conductor), 460 (440-480), 575 (550-600), 2400 (2300-2500), and 4160 (4000-4300) volts for 60 Hz, 3-phase power; available in full or pqart winding motors for starting. Motors will operate satisfactorily at 10% below the minimum and at 10% above the maximum system voltage.

3.1.3 Dimensions and Weights

Typical space requirements and shipping weights of a variety of centrifugal packaged chillers are shown in Table 3.2. The dimensions are shown with and without service access area. To estimate the operating weight, multiply the shipping weight by 1.05.

Table 3.2 Typical Centrifugal Chiller Dimensions and Weights

Nominal Capacity (tons)	Length x Width x Height Chiller Only (ft)	Length x Width x Height Including Service Area (ft)	Shipping Weight (lb)
90	14.0 X 3.3 X 5.5	25.0 X 8.5 X 6.0	6,726
200	13.7 X 4.6 X 6.5	29.0 X 9.0 X 7.0	11,030
400	13.7 X 5.0 X 8.3	29.0 X 9.5 X 8.0	17,024
800	18.0 X 8.0 X 8.5	32.0 X 11.0 X 10.0	28,353
1200	19.5 X 10.0 X 10.0	32.0 X 11.0 X 11.0	39,454
1600	20.0 X 12.0 X 11.0	33.0 X 14.0 X 12.0	50,996
2000	20.0 X 14.4 X 10.0	35.0 X 16.0 X 12.0	62,160

3.1.4 Refrigerant Charge

Refrigerants normally used in the hermetic and open-type centrifugal chillers are R-11, R-12, R-22, and R-500. Refrigerant 11 frequently is used at low and moderate capacities because it helps to maintain optimum efficiency of pressure vessels designed for 15 psig design working pressure for typical water-cooled applications. R-12, R-22, and R-500 are popular for a wide range of capacities because of favorable compressor size for water chilling applications. For more information on chiller refrigerants see chapters 14 and 31 of Ref. 10, and Chapter 14 of Ref 7.

3.2 MATERIAL AND ENERGY BALANCE

The two major inputs to the centrifugal chiller are the thermal energy received from the space-conditioned buildings and the electrical input to the compressor motor.

The major output of concern is the thermal energy rejected to a heat sink. The actual performance and cost data on the heat dissipating equipment will not be considered in this technology evaluation.

A simple energy balance of a hermetic chiller with a refrigerant cooled compressor motor shows that the output to a heat sink must be equal to: (1) the sum of the thermal energy inputs from the space-conditioned buildings, and (2) the thermal equivalent of the electrical power required to operate the hermetic type centrifugal compressor motor. For an open-type compressor motor, the thermal discharge is equivalent to the heat removed from the conditioned building spaces plus the work of compression.

The Coefficient of Performance (COP) is commonly used to compare the performance of refrigeration systems. The COP may be defined as follows:

$$COP = \frac{Q_3}{E_{th}} \qquad \text{(Eq. 3.1)}$$

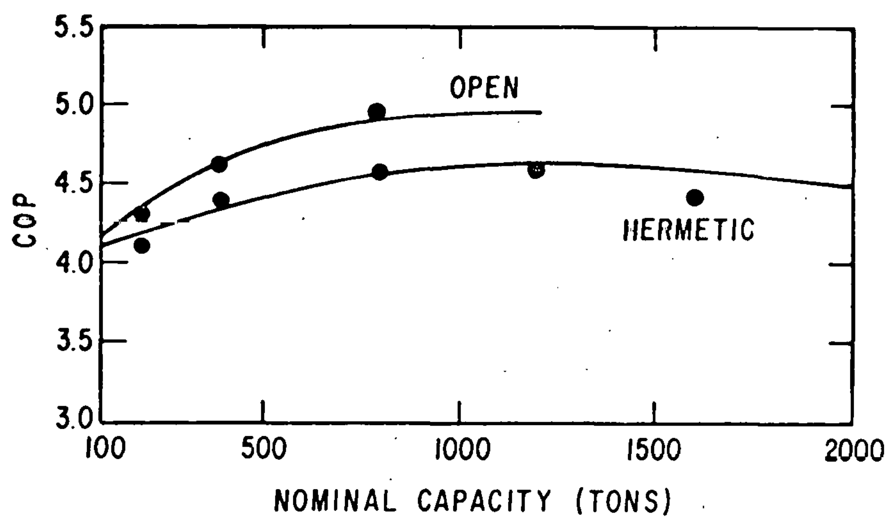
where:

Q_3 = useful refrigeration effect produced at stated conditions, and

E_{th} = heat equivalent of the total energy input rate required to operate the compressor.

3.2.1 Full-Load Performance

Manufacturer listed COP values for a variety of hermetic centrifugal chillers from 90 to 2000 tons and open-type from 90-1250 tons at standard rating conditions, as specified by ARI Standard 550-77 are shown in Fig. 3.2. An open-drive motor is air-cooled and does not require refrigerant as a cooling medium. Elimination of refrigerant motor winding cooling improves the overall COP by 3-8% as shown by the upper curve in Fig. 3.2. The empirical equations shown in Fig. 3.2 represent the COP vs capacity curves.



$$COP\ (HERMETIC) = 4.0 + 0.001\ (TON) - 0.0000004\ (TON)^2$$

$$COP\ (OPEN) = 4.0 + 0.0019\ (TON) - 0.000001\ (TON)^2$$

Fig. 3.2 COP of Various Size Centrifugal Compression Chillers

3.2.2 Performance at Part Load and Various Condenser and Evaporator Water Temperatures

Most centrifugal chillers can be operated down to 10% of the design load. Figure 3.3 provides a variety of part load curves versus power input at different leaving condenser and leaving evaporator water temperatures.[11] The nominal design point in which the power input equals 100% conforms with ARI Standard 550-77.

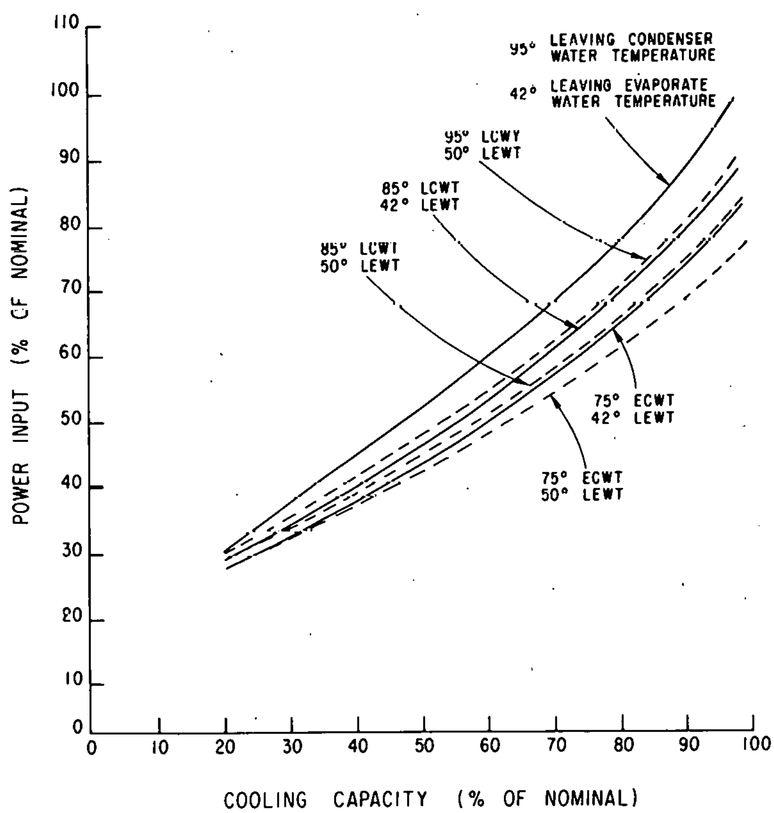


Fig. 3.2 COP of Various Condenser and Evaporator Water

The chiller performance data presented in Fig. 3.3 are based on a 2-pass evaporator and condenser arrangement with a 10°F temperature drop.

Generally, the lowest unit operating cost results from the use of the maximum number of passes possible, and provides the longest practical water travel to increase heat transfer efficiency since the refrigerant temperatures are closer to the leaving water temperatures. Examination of various size centrifugal chillers from one manufacturer indicates about a 5% improvement in COP at full-load conditions for a chiller with 3-pass condenser and cooler opposed to the same chiller with only two passes. The nominal water pressure drop resulting from a 2-pass condenser and chiller varies between 12 and 25 ft of water.[12] Figure 2.5 can be used to estimate the pressure drop through a 2-pass condenser for a centrifugal chiller as the flow is varied. The water pressure drop increases and associated pumping horsepower increases with an additional number of passes. On multi-stage compressors, economics can be designed into the centrifugal units to improve the COP without adding any additional evaporator or condenser surface. An economizer can improve the COP by as much as 6% on some multi-stage compressors; this is the same improvement obtained by adding 15-30% more surface to the evaporator and condenser.[7]

The effect of increased unit capacity and efficiency on kW input vs water pumping horsepowers and condenser water cost should be economically balanced to arrive at the final pass selection for the lowest overall cost of a specific chiller installation.

Figure 3.3 shows that the power requirement is reduced as the chilled water leaving temperature is allowed to rise. The curves shown in Fig. 3.3 are based on manufacturers' data for a 200-ton hermetic unit with a 42°F design exiting chilled water temperature.

Equation 3.2 is based on the data presented in Fig. 3.3 and is capable of estimating the percent of nominal power input to the centrifugal chiller as a function of the part load capacity needed from 10 to 100% of full-load, the leaving chilled water temperature from 40° to 50°F, and the leaving condenser water temperature from 75° to 100°F. The full-load potential capacity of the centrifugal chiller is assumed to be equal to the design capacity within the operating range discussed above.

$$\begin{aligned}\text{\% of nominal power input} = {} & 916.347 + 0.532633(X) - 0.000559686(X^2) \\ & + 0.00002306(X^3) - 32.7860\,(LCWT) + 0.378447\,(LCWT^2) \\ & - 0.00142857\,(LCWT^3) + \frac{1092.31}{LCWT} + \frac{2071.02^*}{LCWT^2} \qquad \text{(Eq. 3.2)}\end{aligned}$$

where:

X = % of full load,

LCWT = leaving condenser water temperature (°F), and

LEWT = leaving evaporator temperature (°F).

The standard error of Eq. 3.2 is 2.30.

3.2.3 Performance as a Function of the Fouling Factor

The performance data on centrifugal chillers is based on an assumed fouling factor of .0005. With good quality water used for both the condensed water and circulating chiller water, .0005 is reasonable. However, if poor water conditions exist which cannot be corrected by proper water treatment, the COP of any centrifugal chiller will be reduced by increased corrosion, depositing heat resistant scale, sedimentation and organic material growth within the condenser and cooler inhibiting heat transfer between the refrigerant and the circulating water.

Figure 3.4 shows the estimated percentage reduction of the nominal COP at fouling factors between .0005 and .002 for either the cooler or condenser. Table 2.3 lists suggested fouling factors for various types of untreated circulating water.

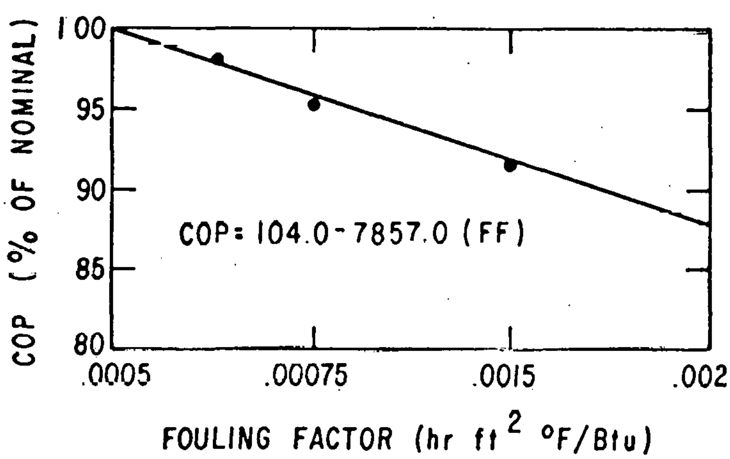


Fig. 3.4 Recution of Centrifugal Chiller COP as a Function of Condenser or Cooler Fouling Factor

If both the condenser and the cooler have fouling factors above .0005 the overall reduction in COP can be estimated by first finding the reduction caused by a higher rate of fouling of one of the components, and then multiplying the reduced COP a second time by the reduction factor due to a higher fouling factor of the other component. For instance, assuming a condenser fouling factor of .001 and a cooler fouling factor of .00075, the resulting COP can be estimated as shown below:

(.98) (.95) (nominal COP)

3.3 OPERATING REQUIREMENTS

3.3.1 Capacity Control

Capacity control is normally provided automatically by sensing exiting chilled liquid temperature and adjusting the compressor capacity control devices, such as the variable inlet guide vanes or the suction damper. If the temperature of the chilled water continued to decrease after the capacity control has reached its minimum position, a low-temperature control will stop the compressor and then restart the compressor when a rise in temperature indicates the need for cooling.

3.3.2 Minimum and Maximum Operating Conditions

The centrifugal chiller can be operated down to 10-20% of the nominal rated capacity successfully. Operating variables, such as chilled water temperature, condenser water temperature, and water flows, can be adjusted to increase the cooling capacity anywhere from 100-150% of the nominal rated capacity. The large variability in the maximum attainable capacity is caused by different combinations of condenser, cooler and compressor sizes.

The maximum number of condenser and cooler water passes should be used, without producing excessive water pressure drop. Sometimes a slight reduction in the condenser water flow (and slightly higher exiting water temperature) will allow a better selection (smaller model with a higher COP) than will the choice of fewer water passes when a rigid pressure drop limit exists.

The entering condenser water temperature should be allowed to drop as far as possible but must be maintained equal to or higher than the return chilled water temperatures. The manufacturer suggested chilled water temperature ranges from 40°F to about 50°F.

A power demand limiter found on most centrifugal chiller machines can be used to limit current draw during periods of high electrical demand charges. This control can be set from about 40 to 100 percent of full-load amperes. Whenever power consumption is limited, cooling capacity is correspondingly reduced.

The chilled water leaving the evaporator must be of a sufficiently low temperature to handle the dehumidification required by the latent load in the building. The widely accepted 44 to 45°F leaving chilled water temperature may prove unnecessarily low in a large number of installations. A comparison of the lifecycle operating cost of centrifugal unit generally will indicate that unless the latent load is quite severe or the distribution of the chilled water to the condition space excessive, 46° to 50°F chilled water temperatures seem economically justified.[13,14]

3.3.3 Multiple-Unit Operation

Two or more units can be applied with the chilled and condenser water flows in series through the unit. Assuming two units of equal size, each will reduce in capacity as the load decreases to about 40% of the total capacity, at which point one of the units will be shut down by the sequence control. If more than two units are installed, it appears that each machine should operate as near to 70% of full load as possible to obtain the highest operating COP. Most centrifugal machines use the least energy per ton at around 60 to 80 percent of the maximum rated capacity.[13]

3.4 MAINTENANCE AND RELIABILITY

3.4.1 Maintenance Requirements

Cleaning of inside tube surfaces will be required at various intervals depending on the water condition Condenser tubes will only need cleaning annually if proper water treatment is maintained. Cooler tubes will need less frequent cleaning if the chilled water circuit is a closed loop.

During inspection, seasonal shutdowns, or planned maintenance, refrigerant can be pumped out of the unit into a receiver for storage during this period preventing the loss of costly refrigerant. Where more than one centrifugal chiller is required, only one refrigerant transfer unit and storage system is necessary.

To reduce fouling, a minimum water velocity of about 3.3 fps is recommended in coolers and condensers. Proper water treatment and regular tube cleaning are recommended for all liquid chillers so that power consumption and operating problems are kept to a minimum.[1]

A good indicator of potential operating problems is the amount of purging required by the centrifugal machine. The purge unite evacuates air and water from the refrigerant loop and recovers and returns the refrigerant mixed with air. A machine may be perfectly air-tight, yet develop a water leak that is detected only by operation of the purge system. If water is allowed to remain in the machine, serious damage to tubes and other internal parts can occur.

3.4.2 Economic Life

The suggested economic life of a large centrifugal chiller varies from 15[2] to 25 years.[9,15]

3.4.3 Reliability

Manufacturers indicate some 400 quality assurance tests are performed on each unit prior to shipment. Some centrifugal machines available have qualified for the ASME "N" stamp, symbolic of the compliance with current regulations regarding the use of equipment in a nuclear power plant.

4 SCREW-TYPE PACKAGED LIQUID CHILLERS

4.1 DESCRIPTION

The helical screw-compressor chiller is a positive displacement machine that has nearly constant flow performance and provides incremental capacity modulation similar to the centrifugal compressor units. The capacity control mechanism for screw machines is unique in that a working slide varies the compression ratio rather than the compressor speed, the suction gas inlet, or the number of working cylinders.

- *Compressor* consists of essentially two cylindrical, helically grooved rotors, a male (lobes) and a female (gullies), in a stationary housing with inlet and outlet gas ports. The flow of gas in the rotors is both radial and axial.

Compression is obtained by direct volume reduction with pure rotary motion. Three phases in the principle of operation can be distinguished - suction, compression, and discharge.

- *Suction:* As a lobe of the male rotor begins to unmesh from an interlobe space in the female rotor, a void is created and gas is drawn in through the inlet port.

- *Compression:* Further rotation starts meshing of another male lobe with the female interlobe space on the suction end and progressively compresses the gas. Thus, the occupied volume of the trapped gas within the interlobe space is decreased and the gas pressure consequently increased.

- *Discharge:* At a point determined by the designed built-in volume ratio, the discharge port is uncovered and the compressed gas is discharged by further meshing of the lobe and interlobe space.

The screw compressors used in liquid chillers are oil injected to provide a reduced operating noise, lower operating speed, increased thermal and volumetric efficiencies, and smaller condensers when a portion of the total heat rejection is accomplished by an oil cooler.

• *The evaporator* may be a flooded design or a direct-expansion design. There is no particular cost advantage of one design over the other up to the 300-400 ton range where the flooded cooler becomes more economical. Coolers used in screw-type chillers differ from other liquid coolers in that they are designed for higher maximum working pressure. The condenser may be included as part of the liquid chilling package when water-cooled, or may be remote. Rooftop, air-cooled liquid chilling packages are also available. When remote air-cooled or evaporative-cooled condensers are applied to liquid chilling packages, a liquid receiver generally replaces the water-cooled condenser on the package structure. Water-cooled condensers are shell-and-tube type, cleanable, with replaceable externally finned copper tubes.

• *Oil separators* are required by most screw chillers because the oil is cooled by injection directly into the compressor.

4.1.1 Manufacturers and Available Size Ranges

The screw compressor packaged chillers are classified according to nominal tons of cooling capacity. The operating conditions which the nominal tonnage is estimated agrees with ARI Standard 590-76; 95°F leaving condenser water temperature; 44°F exiting chilled-water temperature; and 0.0005 fouling factor.

The hermetic screw compressor packaged chillers are available in nominal capacities ranging from 100-750 tons. The open-type screw compressor packaged chillers are available in nominal capacities ranging from 500-750 tons. The major manufacturers of screw compressor chiller packages for building space conditioning are Dunham Bush, York, and Carrier.

4.1.2 Dimensions and Weight

Typical space requirements and shipping weights of various screw compressor packaged chillers are shown in Table 4.1.[16] The dimensions shown are with and without service access space for routine inspection and periodic tube removal for cleaning. The approximate operating weight can be obtained by multiplying the shipping weight shown in Table 4.1 by a factor of 1.025. The physical data shown in Table 4.1 apply to both hermetic and open-type compressor chiller units.

Table 4.1 Screw Compressor Packaged Chiller Dimensions and Shipping Weights

Nominal Capacity (tons)	Length X Width X Height Chiller Only (ft)	Length X Width X Height Indlucing Service Area (ft)	Shipping Weight
105*	27.0 X 7.5 X 5.6	27.0 X 14.0 X 7.0	13,000
120	12.4 X 3.0 X 4.9	29.0 X 8.0 X 7.0	6,950
350	14.1 X 3.6 X 6.5	35.0 X 9.0 X 9.0	13,700
580	17.0 X 4.9 X 9.0	37.0 X 12.0 X 11.0	26,500
750	17.0 X 5.4 X 9.0	37.0 X 12.0 X 11.0	29,700

*Rooftop model heat dissipation equipment included in the package.

4.1.3 Electrical Requirements

The electrical inputs available for the screw compressor packaged chiller are 208 (200-208, 4-conductor), 230 (220-240, 3-conductor), 460 (440-480), 505 (550-600), volts for 60 Hz, 3-phase power and 400 (280-415) volts for 50 Hz, 3-phase. The control package requires 115 volts, 3-phase, 60 Hz.

4.1.4 Refrigerant Charge

Manufacturers' literature on screw compressor packaged chillers indicates that R-22 is commonly used in space conditioning systems.[16]

The ASHRAE 1975 equipment volume mentions that R-22 and R-717 are popular because the compressor size required is small. R-12 is not used often because a larger compressor is required and, on air-cooled applications, oil dilution can be a problem at high condensing temperature.

4.2 MATERIAL AND ENERGY BALANCE

The major energy inputs to the screw compressor chiller are the thermal energy from the space inside the buildings being cooled and the electrical requirements of the compressor motor. Neither the thermal energy picked up in the liquid coolant transfer from the building being cooled to the chiller nor the electrical input to the oil pump motor, chilled-water pump, condenser water pump and controls are considered in this technology evaluation.

The major energy output concern is the thermal energy rejected to a heat sink. Because most packaged screw-type compressor chillers do not include a cooling tower or other form of final heat dissipating equipment, most of the cost data presented in this report do not reflect the additional cost of this piece of equipment.

4.2.1 Full-Load Performance

The coefficient of performance is used to compare the performance of screw compressor chillers to other refrigeration systems. The COP is defined as follows:

$$COP = \frac{Q_3}{E_{th}} \qquad \text{(Eq.4.1)}$$

where:

Q_3 = refrigeration effect produced at stated conditions, and
E_{th} = heat equivalent of the total energy input rate required to operate the compressor.

The operating conditions used to display a nominal COP vs. chiller capacity curve are those stated by ARI Standard 590; 44° exiting chilled-water temperature; 95° leaving condenser-water temperature; .0005 fouling factor; 10°F water temperature difference through both chiller and condenser; and normal 2-pass chiller and condenser. The nominal COP values for several hermetic and open-type screw compressor chillers are shown in Fig. 4.1.

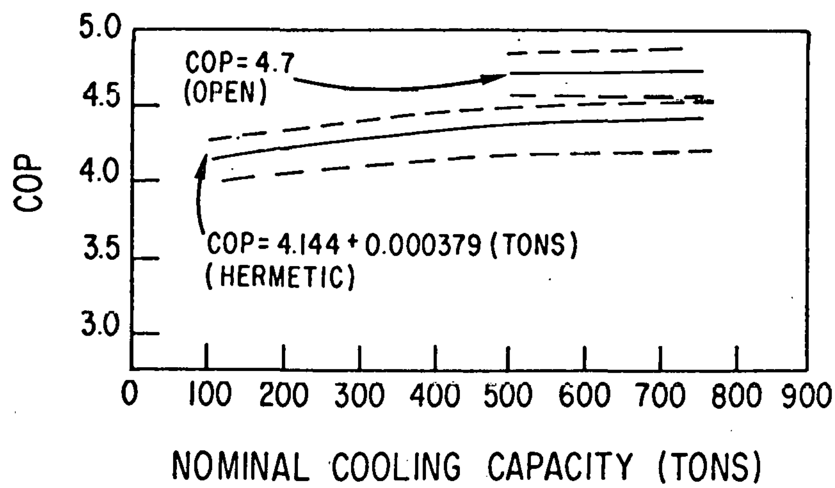


Fig. 4.1 COP of Various Size Screw Compressor Packaged Chillers (Hermetic)

4.2.2 Full-Load Performance at Various Condenser and Evaporator Water Temperatures

The full-load COP and capacity of screw type chillers are dependent on both the condensing temperature and chilled water temperature. Figure 4.2 shows the relationship between the leaving condenser and leaving chilled water temperature values the COP, and the full-load capacity.[16] The values are representative of most screw type chillers.

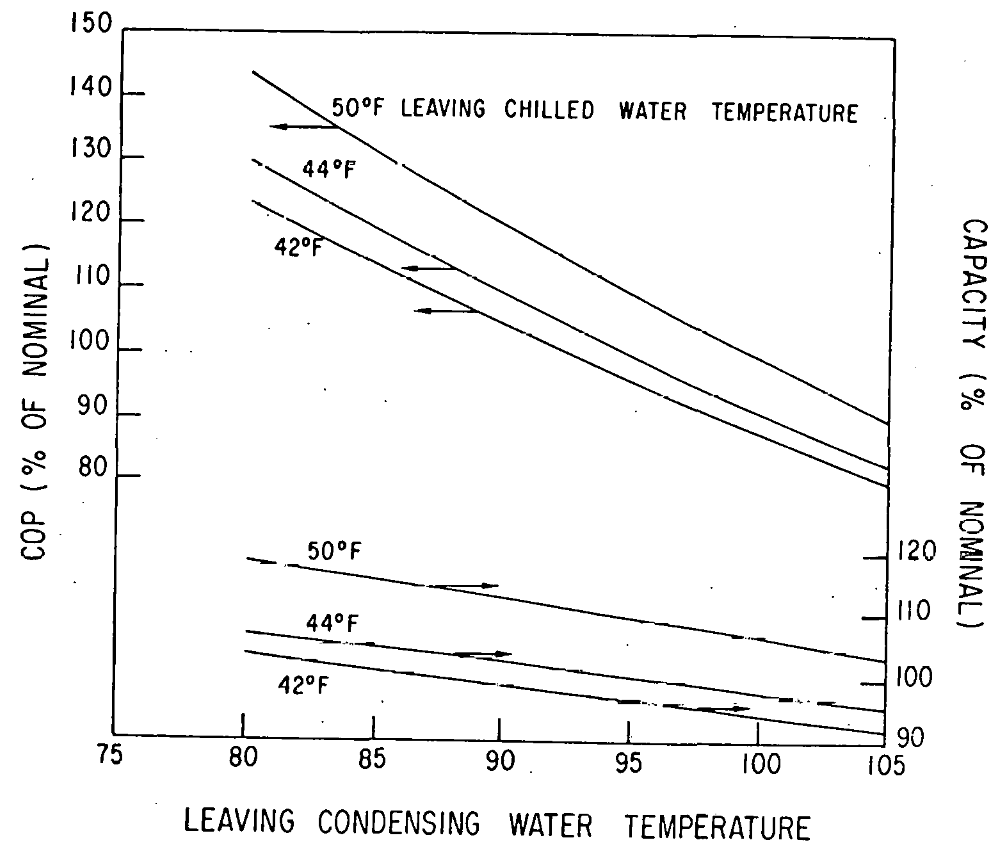


Fig. 4.2 Effect on COP of Varying the Condenser Water Temperature from Nominal Full-Load Conditions

The top three lines in Fig. 4.2 show the change in the full load screw compressor chiller COP as the condenser water temperature changes from 80° to 105°F and chilled water varies between 42° and 50°F. A reduction in the condenser water temperature at constant 3 gpm/ton flow of 5°F results in an 8-10% improvement in the COP. An increase in chilled-water temperature at a constant 2.4 gpm/ton flow results in a 6-11% improvement in the COP.

The lower lines represents the maximum available capacity at various condenser water temperatures from 80° to 105°F. Since the screw compressor is a positive displacement compressor, it does not surge. Since it has no clearance volume in the compression chamber, it will pump high volumetric flows at very high heads. Because of this, screw compressor chillers suffer less capacity reduction at above design condensing temperatures than other types of compressive chillers.

The chilled-water flow is assumed to be fixed at 2.4 gpm/ton, which corresponds to a 10°F chilled water range. The cooling capacity at various leaving condensing temperatures from 80-105°F and leaving chilled water temperatures from 40 to 50°F are related to the chilled water flow and range as shown below:

$$\text{Cooling Capacity (tons)} = \frac{\text{GPM X Chilled water range}}{24} \qquad \text{(Eq. 4.2)}$$

The condenser water flow for the screw compressor chiller performance data shown in fig. 4.2 varies around 3 gpm/ton. The condenser water temperature is assumed to rise 10°F. Thus, to estimate the condenser water flow, the total heat rejection must first be estimated as shown in the following two equations.

$$\text{Heat Rejection (MBH)} = \frac{\text{MBH x 2}}{\text{Condenser Water Range}} \qquad \text{(Eq. 4.3)}$$

The fouling factor assumed in arriving at the curves in Fig. 4.2 is 0.0005.

The empirical equation developed to estimate the percent of nominal COP as a function of the leaving condenser water temperature and leaving chilled water temperature is shown in Eq. 4.4.

$$\text{COP} = -417.618 + \frac{30581.9}{X} - \frac{670960.}{X^2} + 10.1833\,(Y) - 0.0916667\,(Y^2) \qquad \text{(Eq. 4.4)}$$

where:

COP = % of nominal COP

X = leaving condenser water temperature (°F)

Y = leaving chilled water temperature (°F)

The standard error for Eq. 4.4 is 1.85.

The empirical equation developed to estimate the percent of nominal cooling capacity as a function of the leaving condenser water and chilled water temperatures is shown in Eq. 4.5.

$$CAP = -74.6489 + \frac{10343.9}{X} - \frac{289606}{X^2} + 2.77500\,(Y) - 0.0125\,(Y^2) \qquad \text{(Eq. 4.5)}$$

where:

CAP = % of nominal cooling capacity

X = leaving condenser water temperature (°F)

Y = leaving chilled water temperature (°F)

The standard error for Eq. 4.5 is 0.91.

4.2.3 Performance at Part Load

The COP curve shown in Fig. 4.1 is for full-load, 60 hz operation. An estimated COP for part-load operation is shown in Fig. 4.3.

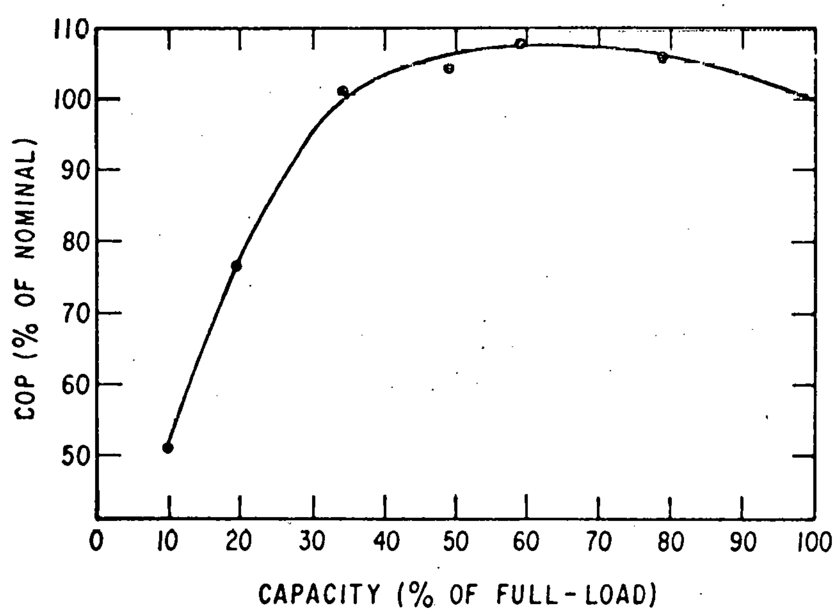


Fig. 4.3 Part-Load Performance Curve (COP Vs % of Full-Load Capacity)

The part-load COP remains at or above the nominal COP down to 35% of full-load capacity. Below 35%, the COP drops relatively quickly. At 20% capacity, the COP falls to about 80% of the nominal value. The machines operate stably down to 10% of full load; however, the COP at that level is only half the nominal rated COP value.

The part-load COP curve can be estimated by substituting coefficients A, B, and C shown in Table 4.2 into Eq. 1.1.

Table 4.2 Generalized Equation Coefficients - Percent of Nominal COP (Y) Vs Percent of Full Capacity (X)

	Coefficients		
Range of X	A	B	C
$10 \leq X \leq 60$	21.07	3.26	-.03
$60 < X \leq 100$	97.1	.36	-.0033

In actual operation, it is rare for full-cooling load to occur; therefore, the total cooling equipment usually is allowed to operate at part-load. A 300-ton screw compressor chiller performance at actual part-load operating conditions has been measured.[14] COP measurements at various exiting chilled-water temperatures were recorded, and an increase of .17 in the COP for each degree rise in leaving chilled water temperature was found. Figure 4.2, which is based on manufacturers' data for full-load condition, shows a change of between 0.05 and 0.12 in the COP for each degree rise in leaving chilled-water temperature.

4.2.4 Performance as a Function of the Fouling Factor

See section 2.2.4.

4.2.5 Chilled Water and Condenser Water Pressure Drop as a Function of Water Flow

For a chilled water flow of 2.4 gpm/ton the pressure drop specified by one manufacturer's screw compressor type chillers varies from 90-150 ft of head, and for a condenser water flow of 3.0 gpm/ton the pressure drop through the condenser is around 160 ft of head.[16] Figure 2.5 can be used to approximate the pressure drop through the evaporator and condenser at different water flows.

4.3 OPERATING REQUIREMENTS

4.3.1 Capacity Control

Usually the leaving chilled-water temperature is sensed for capacity control. The screw compressor has a hydraulically-actuated sliding valve arrangement which controls the working length of the rotors and provides this type of machine with a stepless capacity modulation from 100% down to 10% with complete system stability. A good description of how the capacity is controlled can be found in Ref. 17.

4.3.2 Minimum and Maximum Operating Conditions

The condenser and cooler water temperatures are limited by one major requirement: the entering condenser water temperature must be at least 20°F greater than the leaving chilled-water temperature. For instance, if the exiting chilled-water temperature is 50°F, the lowest allowable entering condenser temperature must be 70°F.

Table 4.3 shows the maximum and minimum water flows recommended by the manufacturer for the more commonly selected 2-pass cooler and condenser arrangement. The nominal flowrate as specified in ARI Standard 550 is 2.4 gpm/ton through the cooler and 3.0 gpm/ton through the condenser.

Table 4.3 Condenser and Evaporator Maximum and Minimum Allowable Water Flow Temperatures

	% of Nominal
Cooler	
Minimum	30 - 44
Maximum	116 - 173
Condenser	
Minimum	2.7 - 29
Maximum	105 - 111

The large variation shown for the cooler minimum and maximum water flowrates reflects the availability of three different size coolers for most screw compressor packaged chiller units.

Various size condensers also are available for each unit, but manufacturers' data only provide performance on one size.

4.4 MAINTENANCE AND RELIABILITY

4.4.1 Maintenance Requirements

The screw compressor has oil injection cooling which is essential to efficient and quiet performance. An important element in the oil-injected screw compressor is the lubricating system's oil separator, which must be maintained at optimum efficiency to ensure maximum compressor output.[17]

The oil filter pressure drop should be monitored carefully and the elements changed periodically. Because the oil system and refrigeration system merge at the compressor, much of the loose dirt and fine contaminants in the system eventually collect in the oil sump where they are removed by the oil filter.[1]

Certain maintenance should be performed annually, or on a regularly scheduled shutdown. These include checking and calibrating all operation and safety controls, tightening all electrical connections, inspecting power contacts in starters, dielectric checking of hermetic and open motors, and checking the alignment of open motors.

Based on 6000 operating hours per year, a reasonable inspection or changeout timetable is shown below:[1]

Shaft seals	1.5 - 4 years	Replace
Hydraulic cylinder seals	1.5 - 4 years	Replace
Thrust bearings	4 - 6 years	Check pre-load and/or replace
Shaft bearings	7 - 10 years	Inspect

It is strongly recommended by the manufacturer that a water treatment specialist be consulted for additive systems to counteract or prevent damages caused by dissolved or suspended materials on the heat-transfer surfaces and in the water-distribution piping.

4.4.2 ECONOMIC LIFE

The economic life of a screw compressor packaged chiller is assumed to be about the same as for a centrifugal chiller, i.e., about 15 to 25 years.

4.4.3 Reliability

Because the screw compressor does not have the operating pressure and temperature limitations of centrifugal compressors, liquid slugging does not harm the machine.

The application of screw compressors for space conditioning is a relatively new development occurring within the last 5-6 years; however, the screw compressor has been used in the refrigeration field for about 13 years.

5 SAFETY REQUIREMENTS

All packaged chillers must conform to the following codes where applicable:

1. *American National Standards Institute* (ANS B9.1)
2. *ASME Code for Unfired Pressure Vessels*
3. *National Electrical Code*
4. *Los Angeles Electrical Code*
5. *Underwriters' Laboratories*
6. *Air Conditioning and Refrigeration Institute*
7. *Southern Standard Mechanical Code* (Chapter IV, Section 401-406)

An example of the type of restriction placed on packaged chillers would be as follows:

Compressors should not be located in any "hazardous location,"and no portion of any refrigeration system should be installed in a location where it would be subject to damage from an external source.

Also, packaged chillers should have an adequate monitoring system to protect the safety of equipment, such as safety controls, to stop the compressor during loss of oil pressure or excessive temperature or during an overload of the motor.

6 ENVIRONMENTAL EFFECTS

Two major environmental effects are generated by the installation and operation of a packaged chiller:

- The thermal discharge is the sum of the internal heat removed from the occupied spaces plus the thermal equivalent of the electrical power input for hermetic units. If the electric motor heat loss is recovered on an open-type unit, the heat recovered from the motor windings can be subtracted from the total thermal discharges.

- Noise can be minimized by constructing the mechanical equipment room or building in an acoustically sound manner, such that the noise level at occupied locations is below noise standards. Machines should not be located near windows or between structures where normal operating sounds may be objectionable.

A standard for sound measurement and rating has been issued by the Air Conditioning and Refrigeration Institute in *ARI Standard 575-73.*

Special attention should be given to the mounting pads where mechanical vibrations might be transferred to occupied building spaces. Most manufacturers offer a more shock-absorbing mounting bracket as auxiliary equipment. Another environmental impact mitigating measure which can be taken is to operate the chillers with the lowest possible condenser-water temperature to reduce both power consumption and the noise.

7 COST CONSIDERATIONS

7.1 ESTIMATED F.O.B. CAPITAL COSTS

Figure 7.1 shows the estimated equipment cost of several types of compressive packaged chillers. The lower curve (1) shows the equipment cost for water-cooled packaged reciprocating water chillers, ranging in size from 10 - 185 tons.[9] The cost curve shown does not include the costs of equipment for heat dissipation to the environment or installation.

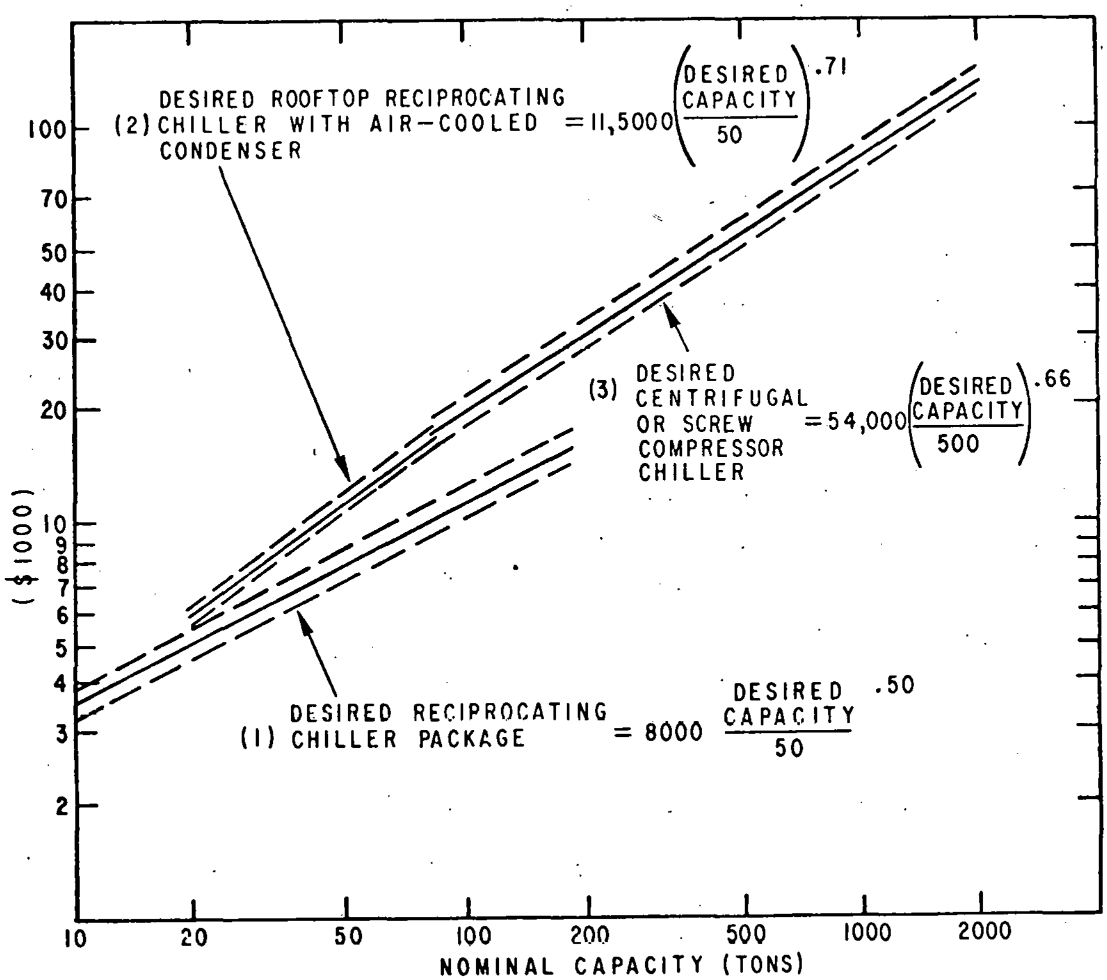


Fig. 7.1 F.O.B. Equipment Cost (1976 dollars)*

*Installation cost not included.

An air-cooled, packaged, reciprocating water chiller conventionally has the same size equipment as the corresponding water-cooled unit less the water-cooled condenser. To arrive at the estimated air-cooled reciprocating chiller cost, multiply the water-cooled reciprocating chiller unit cost shown in Fig. 7.1 by a factor of .87. The air-cooled reciprocating packaged chiller cost arrived at in the above mentioned manner does not include the air-cooled condenser. Examination of the air-cooled condenser cost for several different size air-cooled reciprocating compressor water chillers indicates that the cost of an air-cooled condenser is between 55 to 60% of the equipment cost for the air-cooled reciprocating package chiller itself.

Figure 7.1 (2) shows the equipment cost for air-cooled rooftop reciprocating water chillers from 20 to 85 tons.[9] Although the cost includes that of heat dissipating equipment, it does not include that of installation.

The upper right curve shown in Fig. 7.1 shows the estimated F.O.B. capital cost for hermetic type centrifugal compressor packaged chillers from 80 to 2000 tons.[9] The cost does not include heat dissipating equipment, such as cooling towers, or installation. An examination of the cost curve for centrifugal compressor chillers with open-drive electric compressor motors (3) reveals that they are between 15% to 18% higher than hermetic-type chillers.

For preliminary cost estimating purposes, the centrifugal compressor chiller equipment cost curve can also be used for estimating the screw compressor chiller costs.

7.2 ESTIMATED INSTALLATION COSTS

Figure 7.2 shows the number of man-hours needed to align and erect in place a compressive chiller varying in cooling capacity from 10 to 2000 tons.

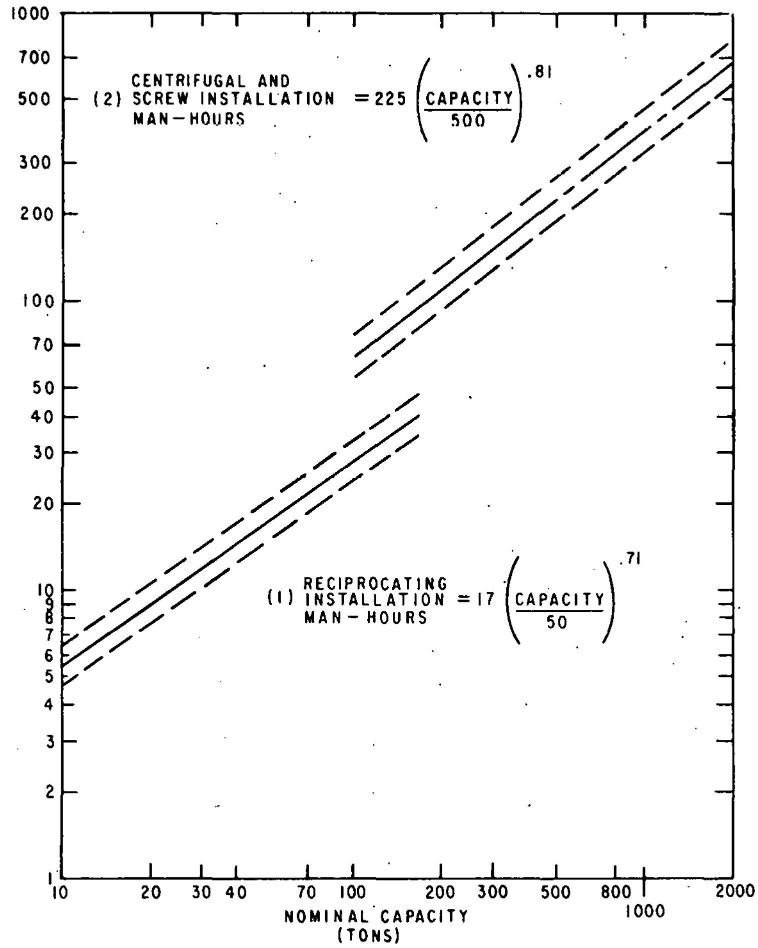


Fig. 7.2 Labor Requirement for Aligning and Erecting Packaged Chillers

The lower curve shows the estimated manhours for a reciprocating packaged chiller from 10 to 185 tons. The labor involved includes installation up to piping and wiring. Although piping and wiring vary considerably from job to job, the data shown in Figs. 7.3 and 7.4 might be of assistance in estimating the total installation cost.[9]

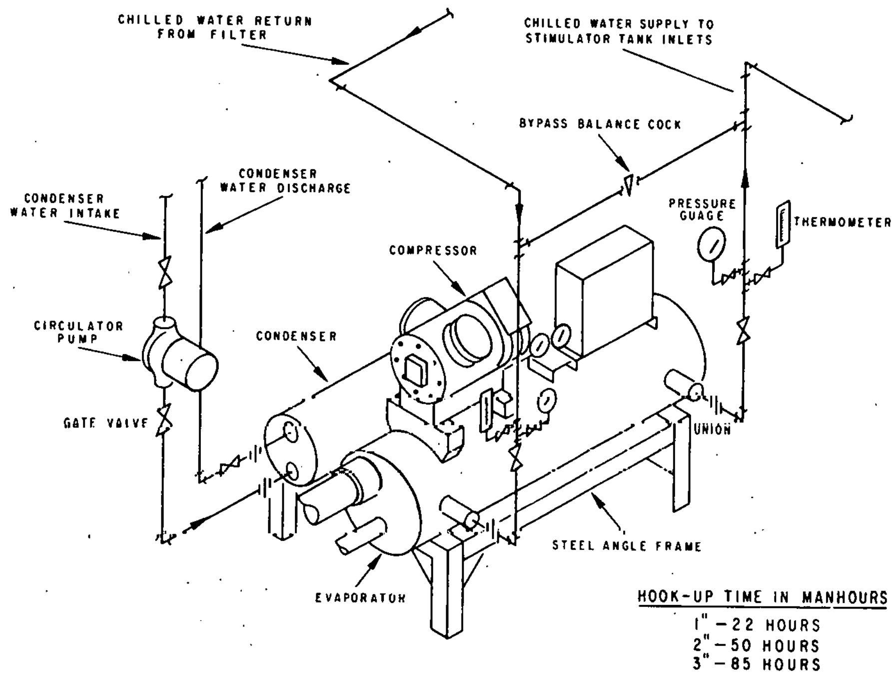


Fig. 7.3 Water-Cooled Water Chiller Piping[4]

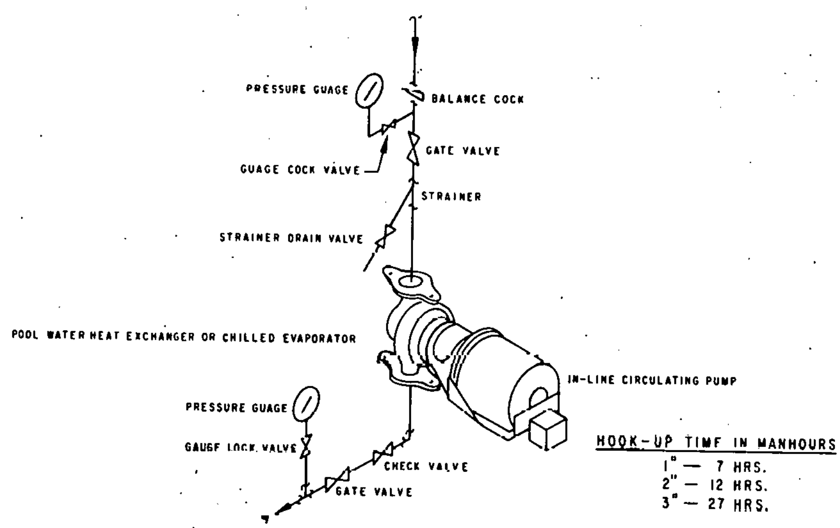


Fig. 7.4 Water Chiller/Pump Hook-Up[4]

Two mechanical cost estimating manuals that provide detailed cost information for specific job estimating are: *Ottaviano's National Mechanical Estimator*[9] and Richardson's *Process Plant Construction Estimating Standards.*[18]

The labor rate (1976) for installing and servicing mechanical equipment in the Knoxville, Tennessee area was about $12/hr. Overhead and profits about doubles the total installation cost.

7.3 ESTIMATED OPERATING AND MAINTENANCE (O&M) COST

Figure 7.5 shows the estimated full-service maintenance contract cost for a water-cooled compressive chiller -- reciprocating, centrifugal, or screw-type.

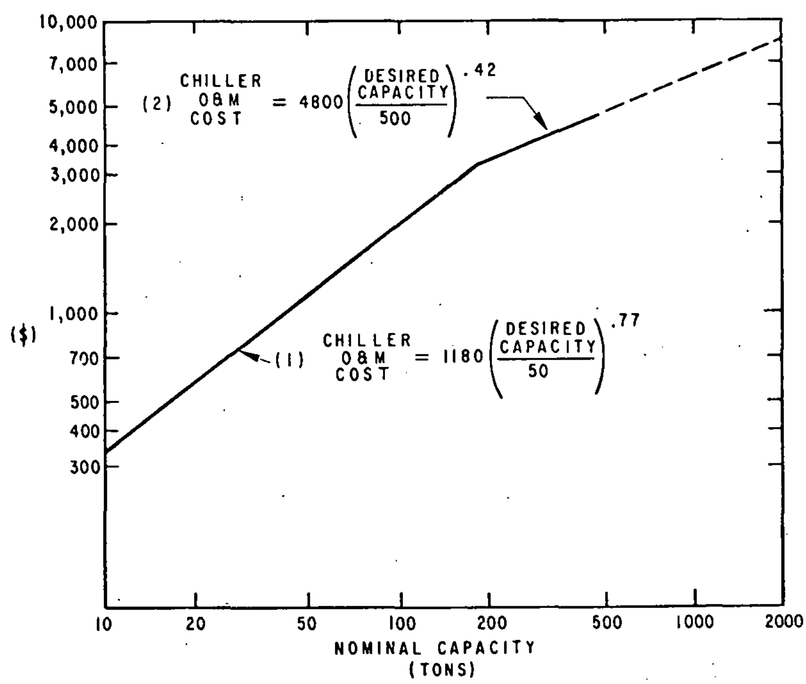


Fig. 7.5 Chiller O&M Cost

The full service maintenance contract normally is written for a one-year period. During this time, the air-conditioning company contracts to keep in good repair and operating efficiently, the equipment covered by the contract. The annual fee shown in Fig. 7.5 is charged for the overall expenses, labor and parts. With this type of contract, the air-conditioning company assumes the risk of full equipment failure. The service contract cost reflects a 35-40% mark-up which is actually the insurance premium needed to guarantee the equipment. The cost to replace the compressor times 1/8 to 1/9 represents about 35-40% of the fee charged by the air-conditioning contractor who realizes a "fair profit" in today's market[9].

Equations 7.1 and 7.2, reproduced in Fig. 7.5 can be used to represent mathematically the operation and maintenance cost curve.

Reciprocating Units between 10 and 185 tons

$$\text{Chiller O\&M cost} = 1180\left(\frac{\text{capacity}}{50}\right)^{.77} \qquad \text{(Eq. 7.1)}$$

Centrifugal and Screw-Type Uniits between 105 and 2000 tons

$$\text{Chiller O\&M cost} = 4800\left(\frac{\text{capacity}}{500}\right)^{.42} \qquad \text{(Eq. 7.1)}$$

The correction factors shown in Table 7.1 are useful in adjusting the O&M costs to various conditions.

Table 7.1 O&M Cost Correction Factors

Age factor	
1 to 5 yrs	1.00
6 to 8 yrs	1.10
9 to 11 yrs	1.20
Inspections per year	
4	0.80
6	1.00
7	1.10
8	1.25
9	1.40
Direct labor cost	
$ 7.00/hr	0.65
$ 9.00/hr	0.84
$12.00/hr	1.00
$14.00/hr	1.03
$16.00/hr	1.16
More than one unit located in same general area	
1st unit	1.00
2nd unit	0.80
3rd unit	0.80
4th unit	0.70
5th unit	0.65
>5 unit	0.65

8 INTEGRATION INTO AN ICES

The peak cooling load which is of relatively short duration imposes a substantial demand charge on the energy consumption of an electric motor-driven compressor. Use of water storage tanks - that can be filled with chilled water at night and then emptied during the daytime when the load reaches higher levels - should be closely evaluated because even the relatively high investment cost of tanks can be returned in a relatively short period through the use of more even compressor chiller loading and also smaller capacity machines.

As discussed in the introduction of this evaluation, all three types of chillers are available with both hermetic and open-drive electric motors. It appears advisable to consider the open drive in total energy plants because the heat released by open motors can possibly be utilized rather than reducing the cooling capacity and COP.

9 STATUS OF DEVELOPMENT AND POTENTIAL FOR IMPROVEMENT

Individual, prepackaged refrigerating units for industrial processing and space-conditioning have been available for years.[19] Most of the recent improvements have been to reduce the onsite installation cost by factory-assembling as much of the piping and system controls as possible.

Two areas not largely incorporated into the use of packaged chillers are thermal storage and heat recovery from the open-type compressor motors.

Use of thermal storage would reduce the required peak capacity and allow the refrigeration machine to operate at steady-state conditions more hours near the unit's optimal operating conditions. Excessive on and off cycling of the compressor reduces the COP from the obtainable steady-state COP. The storage medium could be water or the refrigerant itself stored at an intermediate point in the refrigerant loop.

The second packaged chiller modification offering potential performance improvement is the utilization of the waste heat from the electric motor. A hermetic-type compressor performance is penalized because of the heat added from the electric-motor windings. Presently available open-type compressor motors generally are allowed to heat up the surrounding machine room and eventually to be vented to the atmosphere.

Another potential improvement to the packaged chiller could be to provide automatic condenser tube cleaning devices to hold down the fouling factor.[20] To each condenser tube is added two baskets and a polypropylene bristled brush. One basket is mounted at each end of a condenser tube, to house the brush at the downstream end of the tube. The brush is propelled through the tube by reversing the direction of condenser flow. The flow reversal is accomplished by means of an automatic, four-way valve incorporated into the condenser-water piping system.

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