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(54) Title: ENERGY MANAGEMENT SYSTEM FOR REFRIGERATION SYSTEMS

(57) Abstract

A system and method for managing energy in large refrigeration systems and the like by continuously monitoring operating parameters and controlling to optimize the refrigeration system elements. A multiplicity of remote sensors (142) are disposed at the appropriate points in the refrigeration system to produce analog electrical signals representative of various temperatures such as wet bulb temperature, dry bulb temperature, condensing temperature, evaporator air temperature, evaporator refrigerant temperature, and similar temperatures, and various pressures such as head pressure, booster suction pressure, intermediate suction pressure, and the like. A signal processor (102, 104, 106) is provided to receive signals from the sensors, to condition the analog signals, convert to digital signals and to feed a digital computer (100) which has a memory (114) for storing system design parameters and refrigerant characteristics. The digital computer calculates the instantaneous system operating parameters from the digital signals, compares these parameters to the design parameters, and operates control relays (108) to load and unload compressors to maintain optimum system pressures, to energize and deenergize fans and pumps in accordance with the system requirements, and to produce alarms (161, 162) when non-condensible gases and/or fouled condensers are indicated in the refrigeration system. In systems subject to frost build up on the evaporator, defrost cycles are also controlled by the computer to occur only when necessary.

ENERGY MANAGEMENT SYSTEM FOR REFRIGERATION SYSTEMS

Technical Field

The present invention relates to an integrated energy management system for refrigeration systems and more particularly to a method and system for automati-
5 cally controlling certain functions of a refrigeration system to minimize the energy costs and to provide the user with continuous data on parameters of the system.

Background Art

Over the recent decades, the use of refrigeration
10 systems and air conditioning systems has grown to the point that such systems represent a major consumer of electrical energy in the United States as well as in many of the major industrial nations of the world. The designs for such systems developed during a period
15 when electrical energy was extremely cheap. Efficiency often was relegated to a secondary factor in such designs with the initial cost being a primary factor. However, in recent years, the shortages of fossil fuels for producing electrical energy and the
20 continuing increase in demand for electrical energy has led to significant improvements in the design to minimize input power, both in the interest of conservation of energy and the reduction of operating costs.
25 As is well known, the price of electricity has continued to spiral upward to the point that operating costs may dominate over capital investment costs. This situation is particularly severe with respect to

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large industrial refrigeration systems, such as those
having capacities from 100 tons (90 metric tons) to
1,000 tons (909 metric tons) and greater. To
illustrate, a 1,000 ton (909 metric ton) refrigeration
5   system, designed to the state of the art, may have an
annual operating cost of over $400,000.00 for power at
5¢ per kilowatt hour. This number assumes that the
system is operating at its design parameters and that
all of the components are operating at their maximum
10   efficiencies.

     What may not be generally realized is the penalty
that the user must pay for naturally occuring
deterioration of the elements of the system and for
the changes in the operating environment which cause
15   the operating parameters to depart from the original
design parameters. To illustrate this problem, assume
an ammonia compressive type refrigeration system
having an evaporative condenser and the following
design parameters:

20   Average Load   1000 Tons
                                         (909 metric tons)

     Suction refrigerant   20°F (-6.6°C)
     temperature

     Wet bulb temperature   78°F (25.5°C)

25   Condensing temperature   96.3°F (35.7°C)

     Compressor motor efficiency 0.92

     Condenser fan and water   0.85
     pump motor efficiency

     Fan and water pump   10% of
30   horsepower   tonnage load

     When this system is operating at its design para-
meters and at maximum efficiency, with a power cost of
5¢ per kilowatt hour, the cost per ton of cooling may
be calculated to be 4.5¢ per ton per hour which gives
35   an annual operating cost of $394,200.00.

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Two common problems that arise that will decrease the efficiency and therefore increase the operating costs are: build up of non-condensible gases in the system, and film fouling in the evaporative condensers. As is well known, either of these problems reduces the efficiency of the system. For example, a fouled condenser will result in an increase in condensing temperature and head pressure. Assuming that this change is from 35.7°C to 38.3°C, it can be calculated that the costs per ton per hour will increase to 4.78¢ and the yearly cost to operate will increase to $418,728.00, which represents wasted power totalling $24,528.00. In this same system, for power costs of 7¢ per kilowatt hour, the wasted power would cost $34,339.00 if the conditions were allowed to continue for one year.

Similarly, if non-condensibles build up in the system, the head pressure will increase for a given load. Assuming that the normal pressure for 35.7°C condensing temperature to 13 kg/cm^2 increases to 14 kg/cm^2 due to non-condensible gases, the efficiency is decreased about 8% causing a yearly increase in power cost of $31,536.00.

It may be noted that in the example given, a condensing temperature of 35.7°C will occur for ammonia at a head pressure of 13 kg/cm^2 under normal conditions. When the condensing temperature rose to 38.3°C as assumed due to a fouled condenser, the head pressure would also go up proportionally. However, most systems in current operation provide the operator only with measurements of the head pressure and many plant operators depend upon such pressure readings to determine when certain preventive maintenance is

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required such as purging the system of non-condensible
gases and cleaning of condensers. As will be shown in
more detail later, to determine when condensers are
fouled requires knowledge of the wet bulb temperature
5   at the condenser and the condensing temperature of the
refrigerant. Similarly, determination of the presence
of non-condensible gases requires knowledge of the
head pressure and the condensing temperature. As is
well known, the head pressure and the condensing
10   temperature can increase in synchronism in a system
operating at its maximum efficiency due to an
increased load. Similarly, when operating on a less
than maximum load, the head pressure and condensing
temperature will decrease proportionately. Thus, a
15   head pressure reading alone cannot indicate a need for
purging or for maintenance of the condenser.
Consequently, many large systems may operate for an
extended period of time with these problems remaining
undetected and therefore waste power.
20   Another source of wasted energy in present
systems stems from the problems of controlling the
fans and water pumps associated with the condensers.
For example, when the relative humidity is much lower
than that utilized as a design parameter, it may not
25   be necessary to operate the fans since natural
evaporation may be sufficient. The important
parameter here is the difference in temperature
between the wet bulb reading and the condensing
temperature referred to as ΔT_C. For the design
30   example given above with a wet bulb temperature of
25.5°C and a condensing temperature of 35.7°C, the ΔT_C
is equal to -7.7°C. Under conditions of much lower
temperature than the average value for the location of

the system utilized at design, the wet bulb
temperature will go down. However, the control
criteria should be ΔT and not an absolute measurement
of pressure or humidity, since a higher condensing
5   temperature will also increase ΔT_C, but the wet bulb
temperature remains constant. Similarly, if the load
drops on the refrigeration system the cost to operate
the compressors will drop proportionately but the
condenser pumps and fans will remain operating at full
10   load. Again, power may be wasted from this source.

When the outside air temperature is very low and
at periods in which the water may freeze, the water
pump motors may be cut off, saving energy. Again, the
wet bulb and dry bulb temperatures must be known to
15   determine when this can be done.

If a user of this type of refrigeration system
can keep the system free of non-condensible gases,
maintain the condenser at maximum efficiency, and
control the running of the water pump motors and fan
20   motors, the cost per ton per hour of the system can be
prevented from increasing above its value predicted
from the design and can in many instances be
significantly reduced below that by taking advantage
of favorable changes in ambient conditions and during
25   periods of reduced load. These various problems have
been recognized by those of skill in this art. For
example, in the following U.S. Patents, the head
pressure has been used as an indicator of efficiency
and apparatus provided sensitive to a measurement of
30   the head pressure to effect certain system controls:
Vogel, et al., U.S. Patent No. 4,193,781; Seely, U.S.
Patent No. 3,481,152; and Wood, U.S. Patent No.
3,196,629. Vogel has elected to prevent the system

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head pressure from going below a minimum for a
particular operating mode. Wood is concerned with a
refrigeration system using an air cooled condenser to
maintain a desired head pressure to prevent the
5   capacity of the system from being impared during low
outside ambient temperature conditions. Seely
utilizes a plurality of condensers, switching the
condensers on and off to maintain a desired head
pressure. Therefore, these patents fall far short of
10 the requirements delineated above. In U.S. Patent No.
4,085,594 to Mayer, a control responsive to the
temperature of the condensed refrigerant is disclosed
to vary the speed of cooling tower fans so that the
minimum power is required at all times. While
15 possibly advantageous in a tower which simply cools
the water, it does not take into account the relative
humidity as is desirable in an evaporative condenser.
McAshan, U.S. Patent No. 3,707,851 measures the
ambient air temperature in a refrigeration condenser
20 and the temperature of the liquid refrigerant to
produce an alarm when a malfunction in the system,
fouled condenser, or the like occurs. Carroway, in
U.S. Patent No. 2,847,831, is concerned with
controlling the operation of systems, including
25 evaporative condensers, and recognizes the conditions
under which the fan and/or pumps can be shut down.
Various mechanical type pressure and temperature
measuring elements are utilized, but he chooses to
base his control on atmospheric air temperature rather
30 than wet bulb temperature.
     Automatic purge mechanisms are known in the art
which will detect the presence of non-condensible
gases in the system and will automatically vent these

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to the atmosphere. For example, see U.S. Patents to
Indis, et al, No. 3,013,404 and to Kieme, No.
2,598,799. However, automatic purging in an ammonia
system which results in loss of refrigerant causes
5   safety problems. It is thus preferable to require
purging to be done manually under controlled
conditions. When non-condensibles are due to leaks,
the automatic purgers may not operate fast enough and
therefore are not satisfactory. Other patents have
10   provided various display and diagnostic aids to assist
the operator in locating malfunctions in a system.
These include Anderson, et al, U.S. Patent 4,038,061
and Wills, U.S. Patent 4,146,085. Schulze, Sr., in
U.S. Patent 4,186,563, discloses a display of energy
15   use. Kayan, in U.S. Patent 3,082,951, discloses
methods of calculating the performance of a
refrigeration system for system measurements.

Although some of the energy efficiency factors
discussed above have been attacked individually by
20   workers in this art, there has been no known modern
and efficient energy management system available which
can provide certain automatic controls responsive to
measured values of specific system parameters, and
which also provides readouts to the operator or alarms
25   alerting the operator to perform certain manual
maintenance actions.

Disclosure of the Invention

My invention is an energy management system for
large refrigeration systems which generally have one
30   or more evaporative condensers. This system comprises
four basic elements. First, a plurality of sensors is

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provided which sense various system pressures and
system temperatures at the appropriate points. The
system is adaptable to being installed in an existing
refrigeration system, and may, of course, be an
5   integral part of a new installation. The pressure and
temperature sensors are of the electrical type and I
prefer types which can be utilized with relatively
long leads to permit location of each sensor at a
convenient location.

10   Second, the sensing elements are connected to a
microcomputer system which includes signal processing
circuits and analog-to-digital converters which
produce digital output signals from the various analog
parameter measurements from the sensors.

15   Third, a programmable memory system is provided
and is programmed with the various design parameters
of the refrigeration system into which the energy
management system is installed. The computer is
programmed to scan the various sensor inputs and to
20   store the readings of each input. The computer
calculates a number of measures of system performance
by various algorithms and the measures are compared to
the various stored parameters. Numerical readouts are
then provided for various parameters which are of
25   significance to the operator.

          Fourth, the computer includes an interface unit
which is connected into the refrigeration system such
that control of certain components of the
refrigeration system may be effected as required by
30   the system conditions. A printer is included which
will periodically print out status reports for the
system. Alarm signals are provided which, if certain
faults occur in the refrigeration system which would

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decrease its efficiency, alerts the operator to take
maintenance actions.

Discussing my invention as applied to a two-stage
refrigeration system having a set of main compressors,
5   a set of evaporative condensers, an intercooler,
evaporators, and booster compressors, the specific
measurement and control action will be described. The
evaporative condensers utilize water pumps which serve
to spray water over the condensing coils to maintain a
10 film of water on the coils by which latent heat is
absorbed during evaporation. The unevaporated water
is collected at the base of the condenser and
recirculated by the pump. Fans are provided to cause
a flow of air through the condensers to carry away the
15 heated vapor and to thus eventually transfer the
removed heat to the atmosphere. I provide two
temperature sensors associated with the condensers.
One sensor responds to the temperature of the outside
ambient air referred to as the dry bulb temperature,
20 and the second sensor is installed in the water flow
path and measures what is referred to as the wet bulb
temperature. Thus, the computer is provided with the
necessary input measurements to be able to calculate
the relative humidity from the wet and dry bulb
25 readings. In a system having multiple condensers, a
wet bulb sensor would be provided for each condenser.

A key system parameter is the condensing tempera-
ture and accordingly I provide a temperature sensor
immersed in the liquid refrigerant flowing from the
30 condensers to the receiver. An evaporator is in the
area to be cooled by the refrigeration system and I
provide temperature sensors to monitor the air
temperature in the evaporator and the temperature of
the vaporized refrigerant at that point.

The compressor discharge or head temperature is measured in my system between the condensers and the receiver. This location takes into account the pressure drop across the condensers. The input
5   pressure or compressor suction is measured by a pressure transducer installed between the intercooler and the main compressors. The pressure between the evaporator and the booster compressor, termed the booster suction, is measured by a transducer installed
10 at that point.

The computer includes a display board which provides a display of the following:

1. Wet bulb temperature, T_{wb}.

2. Ambient temperature, T_{db}.

15 3. Percent humidity.

4. Percent system efficiency.

5. Compressor suction T_s.

6. Compressor discharge or head pressure, T_h.

7. Cost per ton per hour.

20 8. Fouled condenser alarm.

9. Non-condensible gases alarm.

10. Status of the system control outputs.

The system control status indicators show which of the various controlled devices are on or off.
25 Typical of such status indications are:

1. Condensers and indicating

   a) Fans off or on,

   b) Pumps off or on.

2. Main compressor loading and indicating com-
30   pressors on or off line.

3. Booster compressor loading and indicating compressors on or off line.

4. Defrost system off or on.

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Temperature and pressure sensor data from the refrigeration system are applied to conditioning circuitry which filters, amplifies and scales the measured parameters. The microcomputer is under soft-
5 ware control which causes it to scan the input sensors one at a time. After selecting a sensor signal for its input, the computer receives a binary number equivalent to the value of the data at that time from the A/D converter. After a scan of all the input
10 sensors is accomplished, the stored digitized data have conversion algorithms applied thereto and the various calculations performed. The parameters to be displayed are converted to an equivalent binary coded decimal format for operation of the front panel
15 displays. The interface unit includes solid state control relays which are activated or deactivated according to results of analysis of the data. The computer may also be programmed to produce trend analysis, and a summary of the data trend and control
20 information may be printed out on the printer at selected intervals.

When the evaporative condensers of the system are mounted outside, there is danger of damage to the water pumps and associated elements due to freezing.
25 The computer will operate the appropriate control relays to cut the water pumps off when the wet bulb temperatures are found to be freezing or below. Similarly, during cold ambient conditions, water is not necessary for operation of the condenser as it
30 will then operate as an air cooled condenser. Thus, by monitoring the ambient air temperature, the head

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pressure and the refrigerant pressure, the computer
may determine when water pumps can be deenergized to
save energy. The computer also utilizes the signals
from the wet and dry bulb sensors to perform the
5   relative humidity calculation for display.

As previously discussed, in a refrigeration
system in which all of the components are in good
operating condition, the condensing temperatures and
the head pressure will have the values found in the
10   condensing temperature versus head pressure charts for
the particular refrigerant in use. When the load on
the refrigeration system changes, the condensing
temperature and the head pressure will change;
however, such changes will follow the relationships
15   indicated in the tables. Any deviation between the
normal condensing temperature and the head pressure
will indicate a problem in the system and will
indicate a reduction in efficiency. The programmable
memory in the computer will have the characteristics
20   of the refrigerant in use and the design wet bulb and
dry bulb temperatures appropriately stored. The
computer will then compare the condensing temperature
and the head pressure at each measurement time. If a
deviation is noted from the stored values and the wet
25   bulb temperature has not changed to a new standard
value, an appropriate alarm will be energized. For
example, if the head pressure and the condensing
temperature are excessive, coupled with an increase in
wet bulb temperature, this condition indicates
30   condenser fouling and the condenser fouling alarm can
be set. I prefer a relative low threshold such that
the condensers can be cleaned before excessive fouling
can take place. Thus, it is not necessary for the

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operator to wait until serious fouling has occurred before he knows to accomplish this preventive maintenance.

Similarly, if the head pressure increases while 5 the condensing temperature remains normal, the non-condensible gas alarm will be energized. Again, any desired threshold for this alarm may be programmed into the computer. When this alarm occurs, the operator may then bleed or purge the system. As is 10 well known, air and oil vapor are the most common non-condensible gases encountered and generally enter the refrigerant lines from the compressors.

As the computer calculates the relative humidity from the wet bulb and dry bulb data, it effectively 15 calculates the efficiency gained from the evaporative effect of the water compared to the energy required by the fans. When this is less than the fan energy, the computer operates the control relays to turn the fans off. The fan may also be deenergized when the 20 relative humidity is so low that air flow from the fans is not necessary. The efficiency of the condenser is determined by the ΔT_C or difference between the wet bulb temperature and the condensing temperature. ΔT_C will change with the variations in 25 load on the refrigeration system in a known manner when compared with the head pressure. When such changes do not track, as in a case in which ΔT_C is reduced, yet the head pressure remains normal and the wet bulb remains normal but an increase in the 30 condensing temperature occurs, condenser fouling is indicated.

An overall efficiency of the refrigeration system may be determined by comparing the head pressure to

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the suction pressure. From stored data in the
programmable memory, such as cost of power and the
power consumption of the compressors and motors, the
computer calculates the cost to operate per ton per
5   hour and displays this information for the operator to
thereby give an overall system efficiency indication.

Another important function of my control system
is to minimize the cost of operation of the fans in
the evaporator that move air from the controlled space
10 across the evaporator coils and the suction or inter-
mediate refrigerant pressures. In prior art systems,
fouling of the evaporator coils due to ice or frost
build-up is controlled by a timing cycle which
periodically bypasses the refrigerant around the
15 evaporator and defrosts by directing hot gas from the
receiver through the evaporator. This method is very
inefficient since it may attempt to defrost when not
needed and delay defrosting when actually required.
Advantageously, I utilize the measurements of
20 evaporator air temperature and evaporator refrigerant
temperature to automatically defrost whenever
sufficient frost build-up occurs to reduce cooling
efficiency. The difference between the refrigerant
temperature T_r and the air temperature T_a (referred to
25 as ΔT_e) is utilized to effect defrost control. When
ΔT_e becomes greater than the design value for the
maximum cooling load, fouled evaporator coils are
indicated and a defrost cycle is instituted.

The measurements of ΔT_e also yields significant
30 power savings during light load conditions, such as at
night and on weekends when the cooled space is not
entered and the materials being cooled have attained
equilibrium. This condition results in a ΔT_e lower

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than maximum load. Thus, evaporator fans may cut off and on during periods of low $\triangle T_e$ with a resultant savings in power. For example, in a cold storage system having a -34°C T_a requirement for a -40°C T_r

5   (5.5°C $\triangle T_e$), and 100 HP evaporator fans, assume that the fans are cut off for $\triangle T_e$=2.2°C and will be shut down one-half of the time during a weekend. For power costs of five cents per kwh, a savings of about $8,660.00 would be realized.

10   For convenience, the following is a list of symbology used herein:

<u>Pressures</u>

P_h main compressor discharge pressure at receiver input.

15   P_{ms}, P_s main compressor suction pressure.

P_{bs} booster compressor suction pressure.

<u>Temperatures</u>

T_{db} dry bulb or outside ambient

20   temperature.

T_{wb} wet bulb or evaporative temperature in condensers.

T_c condensing temperature of refrigerant at receiver input.

25   $\triangle T_e$ $T_a - T_r$

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$$\triangle T_c \qquad\qquad T_c - T_{wb}$$

T_r refrigerant temperature at
evaporator input.

T_a air flowing across evaporator

5   temperature.

It is therefore a principle object of my
invention to provide an energy management system which
can be connected to refrigeration systems and the like
which will control the operation thereof continuously

10 and automatically to optimize the performance and to
minimize the operating costs.

It is another object of my invention to provide
an energy management system for large refrigeration
systems which will provide the operator with constant

15 monitoring of all parameters defined in the design of
the refrigeration system as well as efficiency and
costs to operate.

It is yet another object of my invention to
provide an energy management system for large

20 refrigeration systems which will provide visual and
audible alarms for certain conditions which reduce the
efficiency of the refrigeration system.

It is still another object of my invention to
provide an energy management system having a

25 multiplicity of temperature and pressure transducers
that can be installed at key points in the
refrigeration system and an electronic system for
receiving electrical signals from the transducers for
monitoring all of the key operating parameters.

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It is a further object of my invention to provide an energy management system in which an electronic control system evaluates key system parameters, performs continuous calculations of factors defining

5   the operating efficiency of the system, and automatically controls various components in the refrigeration system so as to optimize the operation of the refrigeration system with respect to operating costs.

10 It is yet a further object of my invention to disclose a method of monitoring and controlling large refrigeration systems to minimize the operation costs thereof.

These and other objects and advantages of my

15 invention will become apparent from the following detailed description and the drawings.

Brief Description of the Drawings

Figure 1 is a simplified schematic diagram of a single stage refrigeration system showing points of

20 parameter measurement;

Figure 2 is a graph of ammonia pressure versus condensing temperatures;

Figure 3 is a functional block diagram of the energy management system connected to a two stage

25 refrigeration system;

Figure 4 is a functional diagram of a defrost system for the system of Figure 2;

Figure 5 is a block diagram of the computer board;

30 Figure 6 is a block diagram of the analog signal input board;

Figure 7 is a block diagram of the display and alarm board;

Figure 8 is a block diagram of the output relay circuit board; and

.5   Figure 9 is a functional flow chart of the computer operation of the energy management system of the invention.

Detailed Description of the Preferred Embodiment

Referring to Figure 1 of the drawings, a
10 schematic diagram of a single stage refrigeration system having an evaporative type condenser is shown. The system consists basically of the evaporative condenser 10 having a coil 12 for receiving the refrigerant in its gaseous state under pressure from
15 compressor 26 via line 11. Water is drawn from a sump via line 15 by pump 14 and introduced at the top of the condenser via line 16. the water is generally sprayed over the coils 12 such that a film of water on the outside of the coils will be evaporated removing
20 heat to condense the gaseous refrigerant to its liquid state. A fan 16 operates to move air across the coils to carry away the latent heat produced. The condensed refrigerant is collected in receiver 18 via line 13 from which it flows under pressure via line 17 to
25 expansion valve 22. It is, of course, to be understood that some refrigeration systems of this type utilize other means for producing expansion of the liquid refrigerant. The liquid refrigerant expands in the evaporator 20 as it changes to its
30 gaseous state and absorbs heat from the ambient air moving through the evaporator. From the evaporator

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coil 24, the gaseous refrigerant is drawn via line 19 into compressor 26 which compresses the gas and causes it to flow back to the condenser.

My invention comprises an energy management
5   system which may be connected to a refrigeration system such as that shown in Figure 1. My system will monitor and measure certain critical parameters of the refrigeration system such as at the points indicated in Figure 1. For example, a measurement of the wet
10 bulb temperature T_{wb} is provided by a sensor bulb installed in the condenser such that the water spray produces a film on the sensor bulb which will be subject to the evaporative effect of the air moving through the condenser. Therefore, the wet bulb sensor
15 will be at a temperature lower than ambient due to the heat lost in vaporization of the water and will be indicative of the operation of the condensing coil 12. A temperature sensor outside the condenser measures the ambient air temperature referred to as the dry
20 bulb temperature T_{db}. As is well known, the relative humidity in the condenser may be determined from the difference between the wet bulb temperature and the dry bulb temperature. The humidity, of course, will affect the efficiency of the condenser since a very
25 high humidity will result in much slower evaporation of the water and a very low humidity will result in rapid evaporation of the water. It is pertinent to note that a custom refrigeration system is designed for a particular geographical region in which it is to
30 be used. The design will take into account the average relative humidity of the region but which will vary from season to season. Thus, the condenser must be of sufficient size to operate properly at the

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expected extremes. Many refrigeration systems will
use multiple condensers such that the amount of
condensing surface can be increased at times of very
high humidity and decreased at times of low relative
5 humidity by cutting condensing units in and out.

Two important parameters that provide information
as to the efficiency of the refrigeration system are
the compressor discharge pressure or head pressure P_h,
and the temperature of the condensed refrigerant which
10 flows into receiver 18, T_c. Measurements of these
parameters taken on line 13 from condenser 10 to
receiver 18 will therefore provide important operating
information. An additional pressure reading which,
when combined with the head pressure P_h, will indicate
15 the efficiency of the compressor is the suction
pressure P_S at the input to the compressor 26 measured
on line 19.

My system may also require measurements of the
refrigerant temperature in the evaporator coil 24 and
20 the temperature of the air flow through the evaporator
T_a. Although I have shown in Figure 1 the main
parameter measurements required for operation of my
energy management system, certain other measurements
may be made for special purposes as will be discussed
25 more fully below.

Before describing the preferred embodiment of my
invention, it is pertinent to discuss briefly the
significance of certain variations in the pressure and
temperature readings of the refrigerant in a system
30 such as shown in Figure 1. For very large systems, it
is common to use ammonia as a refrigerant. Turning
now to Figure 2, a graph 5 is shown which defines the
condensing temperature T_c of ammonia for various head

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pressures P_h.   When a refrigeration system is
designed, the designer will select an operating point
such as A in Figure 2, which will be the condensing
temperature and head pressure at the designed load of
5   the refrigerator for a selected ambient temperature
and relative humidity.   In this example, the head
pressure will be 13 kg/cm^2 and the condensing
temperature will be approximately 35.7°C.   As
conditions in the refrigeration system change from the
10 selected   design   ambient   temperature,   relative
humidity, and refrigeration load, the head pressure
and condensing temperature will change accordingly.
As long as the refrigeration system is working
properly, at maximum efficiency, and within the
15 capabilities of its condensers, these parameters will
change according to graph 5.   In other words, if
conditions were such that the head pressure dropped to
11.25 kg/cm^2 as at D, the condensing temperature would
drop to about 30°C.   As previously mentioned, most
20 refrigeration   systems   provide   a   readout   to   the
operator of a head pressure P_h.   Changes of operating
conditions from the design point A of a system will
always result in changes in the head pressure.
Therefore, without additional information the operator
25 cannot interpret whether the change in head pressure
is normal or abnormal.
        As is well known, certain abnormal conditions
will cause the head pressure and the condensing
temperature to change from the design point to sets of
30 values which are inconsistent with the original design
parameters.   For example, if condenser coil 12 become
coated with mineral deposit from the water, with dirt
or other materials brought in by the air flow, or for

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any other reason and all other system parameters are at design value, such fouled condensers will reduce the efficiency of the condenser with the result that the condensing temperature T_C will rise along with a
5   change in head pressure P_h as indicated at B in Figure 2. Here, the head pressure is now about 13 kg/cm and T_C is now about 38.3°C. An operator depending entirely on monitoring head pressure would have no indication of any abnormal conditions since the head
10   pressure may increase with increased load or ambient or wet bulb temperature. It may also be noted that the condensing temperature T_C will, under the fouled condenser conditions, follow graph 5 along with the head pressure.
15   When non-condensible gases such as air and oil vapor build up in the refrigeration lines, the compressor must do more work to produce the same cooling capacity. Assuming the initial conditions at point A on curve 5, P_h can rise to, for example, about
20   14 kg/cm^2 as at C while the condensing temperature remains at 35.5°C. Again, the operator does not know whether this higher-than-design head pressure is due to the abnormal condition of non-condensibles or to a change in relative humidity or other ambient
25   conditions. As the load on the refrigeration system changes within the limits of the system capacity, P_h will vary along graph 6 while the condensing temperature T_C would vary in accordance with graph 5. It has been found that for each two pounds of head
30   pressure increase due to non-condensibles, a typical system will lose one percent in efficiency. Therefore, the rise indicated at point C would represent about 7 1/2 percent loss in efficiency.
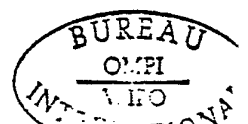
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Similarly, a 0.94°C rise in condensing temperature
will also produce about a three percent loss in
efficiency. As may be readily recognized at this
point, it is necessary to have knowledge of both the
5   head pressure P_h and the condensing temperature T_c to
determine if there are any abnormalities in the system
operation. It is also necessary to know the wet bulb
and dry bulb temperatures T_{wb} and T_{db} to determine if
changes from the standard operating conditions such as
10 at point A of Figure 2 are due to change in ambient
conditions or to change in load conditions, or to
combinations thereof. Normally, the suction pressure
P_s will remain relatively constant as the other
parameters change.

15   In Figure 3, I have shown a functional block
diagram of a large two stage. refrigeration system
having a set of sensors installed therein connected to
a functional block diagram of the energy management
system of my invention. The refrigeration system is
20 shown having two evaporative condensers 30 and 32
illustrated along with connections for additional
condensers. Although modern custom refrigeration
designs may now utilize a large single condenser, it
is much more common to find a multiple condenser
25 system in present use. For large systems, ammonia is
most commonly used as the refrigerant with multiple
condensers connected in parallel. Each condenser
feeds the liquid refrigerant to a common receiver
through traps such that the gas pressure and liquid
30 levels equalize among the condensers. A condenser may
be cut out by turning off the fans and pumps to that
condenser. With systems using freon, control of
condensers may be accomplished by remote control

valves 42 as indicated functionally in Figure 3.
Control of the condensers as the loads change
minimizes operating costs. Such systems are
particularly suited for operation by my energy

5   management system since the additional condensers can
be controlled automatically when desired. A receiver
70 common to all the condensers is shown which
connects via line 71 through intercooler 87 to
evaporator 76.

10   A two stage system as illustrated may be used
when very low temperatures are required. For example,
the room controlled by evaporator 76 may require a
-34°C temperature. Intercooler 87 is utilized to
subcool the refrigerant from receiver 70 down to

15 -7.8°C, for example. Thus, the refrigerant has a
greater capacity for heat absorption in the evaporator
76. For this two stage system, a booster compressor
84 is required which draws the refrigerant from
evaporator 76 and may typically raise its pressure

20 from 23 cm of water vacuum to 1.75 kg/cm^2 at its
output on line 85. The temperature of the refrigerant
at the input to booster compressor 84 may be on the
order of -40°C for this example and 43.30°C at the
output. Intercooler 87 operates to remove much of the

25 superheat picked up from the booster compressor such
that the refrigerant entering the main compressor may
be at about -11°C at 1.75 kg/cm^2. The main compressor
88 then raises the pressure of the refrigerant to, for
example, 13 kg/cm^2 in a typical system. The

30 temperature of the gaseous refrigerant may be in the
range from 42.2°C to 140°C. The compressed gas flows
via line 89 to the coil 34 in condenser 30, and to
condenser 32 and additional condensers if valves 42

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are open. For purposes of discussion, consider that
valves 42 are closed and that only condenser 30 is
operational. Water from sump 50 is pumped via pump 36
to spray system 48 to maintain condenser coil 34

5   coated with a film of water on its surfaces. A fan
system 62 driven by motor 60 is shown schematically
and it is to be understood that each condenser may
have multiple fans. Air from fans 62 flows through
coils 34 carrying the evaporated water and its latent

10 heat away from the coils, normally to the outside air.
Pump 36 is driven by motor 38. I have indicated main
compressors 88 representing a number of paralleled
compressors which are controlled to cut in and out
with variation in load.

15   To   illustrate   the   operation   of   my   energy
management system, I have shown a number of sensors
disposed at the appropriate points in the exemplary
refrigeration system. It is to be understood that the
management system will accommodate additional sensors

20 which may be necessary when multiple compressors
and/or multiple condensers are used.   Referring to
condenser 30 as an example, I dispose a temperature
sensor 46 adjacent to condenser coil 34 such that it
is maintained with a film of water on the sensor bulb

25 from water spray 48. Thus, the temperature registered
by sensor 46 depends upon the relative humidity in the
condenser and provides a measure of the wet bulb
temperature $T_{wb(1)}$. Similarly, condenser 32 includes
a wet bulb sensor 66 producing $T_{wb(2)}$.   Another

30 temperature sensor 68 is mounted outside of condenser
30 and 32 to obtain a measure of the ambient air
temperature noted as T_{db} or dry bulb temperature.
Each of these sensors, which are electrical, connects

to multiplexer 102 connected to signal conditioning
circuits 104 which provide circuits that develop an
analog voltage proportional to the temperature of each
of the connected temperature sensors. Amplifiers are
5 included to produce the desired range of analog
voltages as will be described below.

Sensor 74 is installed in line 35 from the
condensers to receiver 70 in contact with the liquid
refrigerant flowing therethrough. Its output is
10 connected to signal multiplex circuit 102 via lead 75
and produces a measurement of condenser temperature
T_c. Temperature sensor 80 is disposed in the
evaporator 76 so as to be in contact with the
refrigerant flowing through the evaporator. This
15 sensor measures temperature T_r and is connected to
signal multiplex circuit 102 by lead 77. Temperature
sensor 78 is installed in the air stream flowing
through evaporator 76 to measure the air temperature
T_a being controlled by the refrigeration system.
20 Sensor 78 is connected to signal multiplex circuit 102
via leads 79.

For this illustrated refrigeration system,
pressure measurements on this system are obtained by a
set of electrical pressure transducers. Transducer
25 72, connected to the output line 35 from the
condensers, measures the head pressure P_h and is
connected to signal multiplex circuit 102 via leads
73. Transducer 86 measures the suction pressure P_{ms}
at the input of main compressor 88 and transducer 82
30 measures the suction pressure P_{bs} at the input of
booster compressor 84. As may be noted, additional
inputs 116 to signal multiplex circuit 102 may connect

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to sensors and transducers in additional condensers and compressors, not shown.

As may now be understood, the signals from the temperature sensors and pressure transducers are
5 multiplexed and conditioned by circuits 102 and 104 to provide a number of analog voltages each proportional to one of the measured system parameters. Conditioning circuits element 102 is connected via bus 103 to multiplexer 104 which is controlled by
10 microcomputer 100 to scan each of the parameter voltages in a preselected sequence. When a voltage is sensed by circuits 104, it is transferred to analog-to-digital convertor 106 which produces a binary number proportional to the original voltage amplitude.
15 The output from from A/D converter 106 is connected to microcomputer 100 by bus 105. The microcomputer 100 also is connected via bus 113 to parameter memory 114. All of the pertinent system data for the refrigeration system to be controlled is stored in parameter memory
20 114 including: the head pressure/condenser temperature relationship for the particular refrigerant used, which is commonly ammonia; the design parameters of wet bulb and dry bulb readings; maximum system capacity; design head pressure and
25 condensing temperature; power consumption of all fan and pump motors in the system; power consumption of main compressors as a function of head pressure and other parameters; cost of power; and other required preselected data such as values of thresholds for
30 alarms and the like.

The microcomputer is programmed to continuously calculate various measures of the operation of the refrigeration system as will be described in more

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detail below. As will be seen, a number of conditions
may exist in which maximum efficiency and therefore
minimum energy will be obtained when various pumps and
fans are shut down. Therefore, relay control board
5 108 is provided connected to microcomputer by bus 107.
As may be noted from the diagram, relay control board
108 is connected via leads 39 to pump motor 38 for
condenser 30 and pump motor 40 via leads 41 for
condenser 32. Although not· shown, additional
10 condensers .would have their respective pump motors
connected to relay control circuits as indicated at
118. Leads 61 and 63 control fan motors 60 and 64
respectively.

       Relay control circuits 108 are controlled by
15 microcomputer 100 via bus 107 to turn the connected
motors off and on. As will be discussed in more
detail hereinafter, microcomputer 100, in monitoring
the various system operating parameters, will as
appropriate determine that system efficiency can be
20 improved by: cutting condensers in and out by means
of control valves 42 or other control valves for
additional condensers not shown; by cutting fan motors
such as 60 and 64 off and on in various condensers;
and by cutting water pumps such as 38 and 40 off and
25 on in. the various condensers. These control
operations which will occur automatically in
accordance with my invention will result in changes in
certain of the measured system parameters to a more
optimum value as determined by the microcomputer from
30 the information stored in parameter memory 114 and the
data being received from the refrigeration system. As
may now be recognized, my information management
system and the refrigeration system cooperate in a

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closed loop control mode to more nearly optimize
operation in accordance with existing conditions of
refrigeration system load, condition of the various
components, and ambient temperature and humidity
5   conditions.

It is to be understood that the main compressors
88 will include well known mechanisms for controlling
the refrigerant volume in accordance with the
refrigeration load. However, as will be obvious to
10 those of skill in the art, my energy management system
can also monitor the demand on the main compressors 88
and provide the necessary control function. Thus, in
a custom designed refrigeration system incorporating
my energy management system as an integral part
15 thereof, more precise control of the compressors can
be provided. For example, leads 92 from relay control
circuits 108 control the main compressor loading and
leads 94 to booster compressor 84 permits control of
intermediate pressure at the input to main compressors
20 88.

Associated with microcomputer 100 via bus 111 are
display panel 110 and printer 112. Display panel 110
provides digital readouts for each function monitored
by the various sensors and transducers. This permits
25 the operator to determine at a glance the operating
condition of the refrigeration system. Additional
readouts, such as the cost per ton per hour for
operation of the refrigeration system, are provided
with the readout determined by the microcomputer from
30 stored information such as the cost per kilowatt hour
for electrical power and calculations based on the
readings obtained from the sensors. Display panel 110
also provides visible alarms which will indicate

abnormal conditions in this refrigeration system such
as fouled condensers and presence of non-condensible
gases in the refrigerant lines. The visual alarms may
have parallel audible alarms to alert the operator
5   when the problem is noted. Microcomputer 100 will
include selected thresholds which, if exceeded, will
initiate the alarms. Microcomputer 100 controls
printer 112 to print out reports at periodic
intervals, which can be selected by the operator, of
10   system status, system history, and trends of system
operation. Such statistical information, not
available in prior art systems, can alert the users of
the refrigeration system to significant changes in the
use of the refrigeration system which might indicate
15   necessity for increasing capacity, adding additional
condensers, or due to a reduction in refrigeration
capacity needed, a reduction in size of main motors
and the like which could permit smaller motors to be
substituted with a significant saving in operating
20   costs.

Although not shown in the refrigeration system of
Figure 3, a defrosting system must be provided for
evaporator 76 since ice and frost will build up on the
coils and reduce the cooling efficiency. In prior art
25   control systems for refrigeration systems, it is
common to cycle a defrost system on periodically; for
example, the evaporator is operative for 20 minutes,
and is then switched to the defrost mode for 2
minutes. However, when a controlled space is operated
30   under high moisture infiltration conditions and under
high load, the evaporator can become fouled with frost
quickly. The defrost cycle may be too long under such
conditions. Conversely, at night and during weekends

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during which moisture infiltration is minimum, the cycle may be too short.

Advantageously, my energy management system monitors the evaporator conditions and automatically
5 controls defrosting to occur only when needed. Turning now to Figure 4, a typical defrost arrangement is shown illustrating my control system based on the difference ΔT_e between refrigerant temperature T_r and the temperature T_a of the air flowing across the
10 evaporator rather than on a time cycle. In accordance with the refrigeration system design, a maximum ΔT_e is selected based on the maximum load. For illustration, a $\Delta T_{e(d)}$ of 8°C will be assumed from T_a = -40°C and T_r = -31.6°C. Thus, under full load conditions, the
15 refrigerant from expansion valve 179 will be flowing through coil 76 at -40°C as measured by temperature sensor 80 and the air flow from fan 175 will have a temperature of -31.6°C as measured by sensor 78. As frost builds up on coil 76, the heat transfer
20 efficiency of evaporator coil 76 will be reduced and the air temperature will rise. For an amount of increase determined experimentally, the defrost system will be activated.

The flow of liquid refrigerant from receiver 70
25 to evaporator 76 is cut off by solenoid valve 177 via lead 184 from the relay control board of the energy management system. Solenoid valve 176 is opened via lead 186 which permits hot gaseous refrigerant from receiver 70 to flow through coil 76 to quickly melt
30 collected frost. Pilot valve 180 is activated by the hot gas to open valve 182. Fan 175 is turned off during defrost. After defrosting for a predetermined

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period, the energy management system returns valve 176
to off, valve 177 on, and fan 175 on. Pilot valve 180
will return to off as well as spring valve 182.

The value of $\triangle T_e$ is also utilized to decrease
5   system operating costs under light loads, such as at
night and on weekends. For example, if the load is
reduced such that $\triangle T_e$ drops to 2.2°C, evaporator fan
175 can be cut off. For a large refrigeration system,
the evaporator fan motors may be 100 Hp. If a 50% fan
10 duty factor on each weekend can be achieved, the
reduction in heat load due to fan motors and the
reduction in power consumption for a 5¢/kwhr power
cost, a savings of over $8,000.00 per year will
result.

15   Details of the computer controls of my energy
management system will now be given with reference to
Figures 5 through 8 which illustrates the elements
thereof, referred to herein as "boards". Figure 5 is
a block diagram of the CPU board 100 which utilizes an
20 8-bit Z-80 microprossor 125. System clock 128
produces the basic system bit rate. A real time clock
130 is provided to permit real time preprogramming of
the computer system when desired. Random access
memory (RAM) 126 supplies 16K bytes of memory.
25 Programmable read-only memory (ROM) 127 stores the
operational program for CPU 125 as well as the various
calculation algorithms and similar data. Parameter
memory 114 includes parameter sense logic 120 and
parameter patch panel 121. Parameter patch panel 121
30 is comprised of a multiplicity of 10 x 10 matrices
connected to BCD encoders which may be, for example,
74LS147 integrated circuits. Matrices may be utilized
to store installation-specific parameters such as:

system capacity; power drain of the drive motors in
the refrigeration system; cost of electrical power;
and refrigeration system design parameters such as
ambient temperature, wet bulb temperature, head
5 pressure, and condensing temperature. The patch panel
121 may be manually programmed at installation and
changed when required due to modification of the
refrigeration system. Thus, parameter memory 114
supplies CPU 125 with the necessary design data for
10 comparison to operating data for monitoring of the
performance of the refrigeration system.

The computer board 100 receives its input from
the refrigeration system via analog interface 136 and
bus 105 from the analog board shown in Figure 6 in
15 block diagram form. CPU 125 and analog control logic
circuits 131 control the multiplexing of the input
sensors 142 and the gain of amplifier 141 for each
selected sensor. A/D converter 106 in the analog
board of Figure 6 produces binary coded signals as
20 inputs to CPU 125 which utilizes the instantaneous
pressure and temperature readings to compute the
various system operating parameters.

Computer control board 108, shown in block
diagram form in Figure 8, includes a set of solid
25 state relays 170 connected to the refrigeration system
to appropriately control each unit in the system as
required by computer board 100 so as to maintain
maximum operating efficiency of the refrigeration
system at all times. A set of conventional
30 electromagnetic relays 172 is provided having their
contacts in parallel with the contacts of solid state
relays 170. These relays are manually operable by a
key-lock switch to override the solid state relays and

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also will automatically close in case of loss of power
to the computer. Actuation of the manual override
relays 172 will completely remove the control of the
refrigeration system from the energy management system
5 and therefore permit continuing operation of the
refrigeration system in case of failure of the energy
management system. CPU 125, in its continuous
monitoring of the various pressure and temperature
information obtained from sensors 142, will operate
10 solid state relays 170 through relay control logic 132
and relay interface 137 to energize and deenergize the
appropriate elements in the refrigeration system as
indicated by the instantaneous conditions of the
refrigeration system to thereby optimize efficiency
15 and minimize operating costs.

Computer board 100 drives display board 110 shown
in block diagram form in Figure 7 via display
interface 139 and bus 111. All alphanumeric displays
may be 8-digit LED 7-segment types. The temperature
20 displays are controlled by display memory and driver
143 which includes a multiplexer and circuitry to
sequentially energize each of the seven temperature
displays shown to minimize power consumption. Values
of wet bulb temperature, dry bulb temperature, and
25 condensing temperature are indicated by displays 150,
151, and 152, respectively. The conditions in the
evaporator are indicated by the air temperature and
refrigerant temperature displays 154 and 155,
respectively. CPU 125 continuously calculates the
30 difference between the wet bulb temperature and the
condensing temperature, ΔT_c, which is shown on display
152. Similarly, ΔT_e is calculated, which is the
difference between temperature T_a and temperature T_r,

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and is indicated by display 156. Display memory and driver 144 multiplexes readings of head pressure, main suction, and booster suction as indicated by displays 157, 158 and 159, respectively.

5   Display memory and driver 165 operates refrigeration system condition displays 166, 167, and 168 which may be LED bar-type displays. Display 166 provides the operator with the instantaneous cost per ton per hour; display 167 indicates relative humidity; and

10   display 168 indicates the efficiency of the refrigeration system.

Display board 110 also provides system alarms which may be of both visual and audible types. Alarm driver 160 will supply alarm signals to visual alarm

15   display 161 and audible alarm display 162. The status of each of the solid state relays in the relay board Figure 8 is indicated by a relay monitor display 164 operated via display driver 163 to indicate to the operator which elements of the refrigeration system

20   are energized. Decoder 134 in computer board 100 of Figure 4 produces the necessary coding from the output of CPU 125 to operate the displays of Figure 7 via interface 139 and bus 111. In addition to instructing display board 110, CPU 125 controls a data logging

25   printer 112 which may be, for example, an alphanumeric thermal printer to periodically log refrigeration system operating conditions and the various control operations effected. Printer control logic 133 generates the necessary handshake procedures to

30   control printer 112 and is interfaced to the printer via a bit parallel, serial character interface 138.

Turning now to Figure 6, sensors 142 are disposed at the appropriate point in the refrigeration system

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and connect to an analog multiplexer 102 and signal
conditioning circuits 104. Multiplexer 102 may
comprise two dual 8-channel analog multiplexers
providing 16 channels of differential inputs.
5 Multiplexers 102 may be types MPC8D and are under the
control of the computer board 100 to sequentially
select sensors at a scan rate of approximately ten
samples per second. Thus, each sensor is sequentially
connected by multiplexer 102 to signal conditioning
10 amplifier 104. Under computer control, amplifier 104
will have its gain changed to scale the input
parameter to fall within preselected limits.
Amplifier 104, which may be a type 3606A, provides low
drift, differential amplification and filtering to
15 ensure high common mode and normal mode noise
rejection as well as RFI/EMI immunity in harsh
environments. The amplified signal from the sensor is
directed to A/D converter 106 which may be a 13-bit
quad slope unit, type AD7550. A/D converter 106
20 converts the input signal to an equivalent digital
representation between maximum counts of -4095 and
+4095. The digital signals then flow to the computer
board 100 via interface 136.

As previously mentioned, deviation of operating
25 parameters of the refrigeration system from design
values indicates a drop in efficiency of approximately
one percent for each 0.14 kg/cm^2 of high side pressure
deviation or three percent loss for 1°F temperature
deviation. Therefore, I prefer precision sensors that
30 produce accurate parameter measurements. Pressure
transducers such as 72, 82, and 86 of Figure 3 are
preferably Beta pressure cells available from the BLH
Company. These units produce a change in resistance

with a change in pressure and have a rated output of 3 mV/V with an accuracy of $\pm 0.5\%$. Excitation power to the pressure transducers includes provisions to correct for transducer cable resistance variations.

5 The temperature transducers are preferably current devices, such as the model AC 2626 available from Analog Devices Company. This unit produces an output current linearly proportional to absolute temperature and has an output of 1 ua/deg C. Thus, the

10 temperature inputs are insensitive to the length of the connecting cable. These sensors have an accuracy of $\pm 0.5°C$.

A flow diagram of the various operations and calculations performed by the computer of my energy

15 management system is presented in Figure 9. Generally, input data stored in parameter memory 114 are indicated in the left hand column of the diagram and the various system output displays and controls are shown in the right hand column. The following are

20 typical calculations performed by the computer from the readings of the various sensors and typical controls and displays which result from the calculations.

1. Relative Humidity

25 Inputs from wet bulb sensor 46 and dry bulb sensor 68 are processed and compared with psychrometric tables stored in the memory to produce a reading of percent relative humidity which is displayed.

30 2. System Efficiency

When the refrigeration system is operating such that the head pressure P_h and the condensing temperature T_c are exactly in accordance with the

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ammonia pressure-condensing temperature tables and
that the difference between the wet bulb temperature
T_{wb} and condensing temperature T_C is equal to the
design value $\Delta T_C(d)$, then 100% efficiency is being
5   achieved.   To   indicate   a   departure   from   100%
efficiency, the values of P_h and T_C are compared with
the ammonia tables stored in memory to determine if
there is a deviation.   If a deviation is found, the
percentage deviation is calculated and from that a
10   percent   efficiency   is   determined   and   displayed.
However,   if   condensers   are   fouled,   the   system
efficiency will be reduced yet the P_h/T_C relationship
would show no deviation.   Therefore, it is necessary
to calculate ΔT_C from wet bulb temperature T_{wb} and
15   condensing temperature T_C.   After ΔT_C is found, it is
compared to the design value $\Delta T_C(d)$ which is stored in
memory.   If ΔT_C is greater, as indicated by the "yes"
from the decision step, the increase in ΔT_C is then
utilized to calculate the percent efficiency which is
20   displayed.   The change in efficiency due to a change
in ΔT_C is determined with reference to the ammonia
tables.   It has been found empirically that a change
in pressure of one psi (7×10^{-2} kg/cm^2) will result
in a change in efficiency of approximately 0.5%.   It
25   may be noted at a typical operating point that a
change in pressure of one psi is equivalent to an
approximately .75°C change in T_C.   This empirical
information is thus stored in the computer memory to
permit a percentage efficiency to be calculated.
30   It may be noted that since a reduction in
efficiency due to an increase in ΔT_C is caused by
fouled condensers, detection of this condition also
energizes the fouled condenser alarm.   It should also

be noted that fouled condensers could occur simultaneously with a build-up of non-condensibles in the refrigerant. If the calculation of ΔT_C shows no increase, and there is no deviation in P_h/T_C then
5   neither fouled condensers nor non-condensibles are present, indicating 100% efficiency which will be displayed. However, when a deviation of P_h/T_C is found and the decrease in efficiency noted on the display, this also triggers the non-condensible alarm
10   as indicated. The value of ΔT_C determined from the above calculation is displayed to provide the operator with a constant indication of this important parameter.

3. Condenser Controls

15   As previously discussed, $\Delta T_C(d)$ is selected at design of the refrigeration system for full load operation. When the system is operated at less than full load, ΔT_C will decrease due to a reduction in P_h/T_C. Thus, the value of ΔT_C calculated as above is
20   compared to $\Delta T_C(d)$ to determine if it is less than designed value. If so, this is an indication of reduced load. Generally, the compressors in the system will have internal controls which will reduce the capacity of the compressors proportional to the
25   reduction in load so as to reduce the input power to the compressors. This also has the effect of reducing the cost per ton per hour to operate the compressors. However, the condenser fans and pumps, which may draw 10% of the full load compressor input power, will
30   continue to run and their power drain will become a larger percentage of the total input power drain. As an example, assume a refrigeration system having a maximum design value of 852 tons (775 metric tons), a

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$T_C(d)$ of -7.7°C, evaporative refrigerant temperature
of T_r of -6.6°C and a head pressure P_h of 13 kg/cm^2.
When working at full load, the compressors will draw
approximately 946 horsepower which, at 5¢ per kilowatt
5   hour, will have a cost of 4.5¢ per ton or $38.34 per
hour. Assume the condenser pumps and fans require 104
horsepower, their cost to operate will be
approximately $4.57 per hour. Assume now that the
load of the system is reduced from 852 tons (775
10 metric tons) to 213 tons (194 metric tons). At this
point, the compressor power required will drop to 191
horsepower or 3.65¢ per ton per hour for a total of
$7.77 per hour. However, it will be recognized that
the reduction in drain on the compressors does not
15 affect the power required by the condenser pumps and
fans and their operating costs remain at $4.57 per
hour.

The compressors operating at 213 tons (194 metric
tons) output will result in a drop in head pressure P_h
20 to 10.27 kg/cm^2, dropping the condensing temperature
T_C proportionately and for the same condenser
efficiency will result in a ΔT_C of 2.2°C. Dropping
the load further to 106 tons (96 metric tons) will
reduce the compressor operating costs to $3.73 an
25 hour, P_h to 9.9 kg/cm^2 and ΔT_C to 1.6°C with the
condenser fans and pumps still costing $4.57 per hour
to run. I have found empirically that a reduction in
load which causes ΔT_C to drop to 1.7°C will permit the
condenser pumps and fans to be turned off with the
30 resultant rise in T_C well within the capacity of the
condensers. Thus, for lightly loaded conditions, such
as may occur during weekends or during long periods in
which the control space is at equilibrium, significant

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savings can result from shutting down the fans and
pumps.   It should also be noted that changes in
relative humidity and ambient air temperature can
produce a reduction in ΔT_c for a given load and in
5   such conditions turning off condenser pumps and fans
will result in economies.   For systems having multiple
condensers,   my   energy   management   system   will
automatically cut out a condenser by turning off fans
and pumps progressively as ΔT_c drops.   For very
10   lightly loaded conditions, as in the example above,
all of the fans and pumps may be turned off.

As indicated in Figure 9, the lower ΔT_c limit
which may be, for example 1.7°C, is stored as $\Delta T_c(1)$.
If the comprarison of ΔT_c to $\Delta T_c(d)$ shows a smaller
15   value of ΔT_c, this value is then compared to $\Delta T_c(1)$
and if lower, the condenser fans and condenser pumps
are turned off.   As will be recognized, this is a
closed loop system and the calculations and controls
described will tend to be cyclic within the limits and
20   delays programmed into the computer.   Thus, an optimum
savings in power can be achieved without degrading the
refrigeration system performance.

4.   Cost per ton per hour

It is of interest to the operator to know the
25   cost per ton per hour to operate his compressors.   The
cost per ton per hour for the compressors operating at
maximum   design   load   is   determined   for   a   given
electrical   power   rate   and   stored   in   memory.
Similarly, the design head pressure $P_h(d)$ for maximum
30   load is stored.   Measurements of head pressure and
suction pressure are utilized to determine ΔP, the
difference between these two pressure readings.   When
a drop in load on the refrigeration system occurs,

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this results in a reduction of operating costs and is
reflected by a drop in $\triangle$P. This proportional drop in
cost, taken with the stored maximum load cost,
provides a reduced cost per ton per hour figure.
5   However, when there is a fouled condenser or non-
condensibles, the overall system efficiency is reduced
and therefore this number is adjusted upward if
necessary. The final cost per ton per hour is read
out on the display. It should be noted that the
10 pressure reading P_S for this calculation will be the
booster suction pressure in a two-stage system and the
main compressor suction in a single state system.

   5. Optimum Intermediate Pressure Control

   In a two-stage refrigeration system such as
15 illustrated in Figure 3, it may be noted that the
intermediate pressure, which is referred to here as
the main suction pressure P_{ms}, will depend upon the
loading of booster compressors 84 and main compressors
88. In prior art refrigeration systems, this value is
20 preselected for maximum load conditions. the optimum
intermediate suction pressure P_{ms} is calculated by the
following equation:

$$P_{ms} = P_{bs} \sqrt{\frac{P_r}{P_{bs}}} \, ;$$

25 in which all pressure readings are in psi absolute.
As may be recognized, when a prior art system operates
at less than maximum load, the head pressure P_h will
drop with the result that the intermediate suction
pressure is no longer optimum. This increases the
30 cost to operate for lighter-than-maximum loads.
Advantageously, my energy management system can
continuously and automatically control the loading of

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the booster compressors and main compressors to always maintain the optimum intermediate pressure as the system loads vary from time to time. A design value of the booster suction pressure $P_{bs}(d)$ is selected
5   during design and the value stored in the computer memory. The value of booster suction pressure measured by transducer 82 is continually compared to the design value to operate the booster compressors control so as to maintain booster suction at the
10 design value by loading or unloading booster suction at the design valve by loading or unloading booster compressors 84. The measured value of the booster pressure P_{bs} from transducer 82 and of head pressure P_h from transducer 72 is used by the computer to
15 calculate P_{ms} as expressed in the equation above. Suction pressure P_{ms} is monitored by transducer 86 and the main compressors control varies the loading of main compressors 88 to hold the intermediate pressure P_{ms} at the value calculated for the measured head
20 pressure P_h. Therefore, a minimum operating cost for the compressors is obtained at all values of system loading.

       6. Defrost Cycle.

           As discussed in detail above, defrosting of
25 evaporator 76 is controlled by the value of $\triangle T_e$ rather than by a timing device as in prior art refrigeration systems. Accordingly, the evaporator air temperature T_a and refrigerant temperature T_r are utilized to calculate $\triangle T_e$. The value of $\triangle T_e$ at any given time is
30 compared to the maximum design $\triangle T_e(d)$ to determine if the observed value is greater. If so, this indicates fouling of the evaporator coils from frost. Therefore, the hot gas defrost control initiates a

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defrost cycle and the evaporator fan is turned off.
As may be recognized, a close tolerance can be set on
the amount ΔT_e is allowed to increase above its
design value to initiate defrosting, and the
5 evaporator can be defrosted when even a small amount
of frost builds up. Thus, the defrost cycle may be
very short and is only initiated when required. The
same calculation of ΔT_e may be used to determine when
the evaporator is very lightly loaded such that the
10 evaporator fans can be shut down. A low threshold of
$\Delta T_e(d)$ is selected empirically such that when ΔT_e
drops to that value the evaporator fans are disabled.
As ΔT_e then begins to increase due to the lack of fans
to the point at which it exceeds the lower threshold,
15 the fans will be reenergized. The lower ΔT_e threshold
may be selected to produce about a 50% duty cycle for
the evaporator fans under light load conditions. The
value of ΔT_e calculated is displayed to inform the
operator of the condition of the evaporator.
20   Although my invention has been disclosed with
reference to a one-stage and two-stage refrigeration
system, it is equally applicable to other systems and
applications involving compressors, condensers and
evaporators, such as large air conditioning systems,
25 distilling systems, systems employing cooling towers
and others in which various controls depend upon the
interrelationship of various operating parameters. It
will also be obvious to those of skill in the art to
make many modifications to the specific means
30 illustrated without departing from the spirit or scope
of my invention.

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I CLAIM:

1. An energy management system for a single
stage refrigeration system having evaporative type
condensers comprising:

5   A.   a plurality of electrical temperature
sensors (46, 66, 74) attached to said refrigeration
system, said sensors producing electrical signals
representative of operating parameters of said
refrigeration system including

10   a. wet bulb temperature,
                      b. dry bulb temperature,
                      c. condensing temperature, and
                      d. ambient temperature;
             B.   a plurality of electrical pressure

15 sensors (72, 86) attached to said refrigeration
system, said pressure sensors producing electrical
signals representative of operating parameters of said
refrigeration system including
                      a. head pressure, and

20   b. suction pressure;
             C.   signal processing circuits (102, 104,
106) connected to said temperature sensors and said
pressure sensors for receiving said respective
electrical signals therefrom for production a sequence

25 of binary digital signals representative of said
electrical signals;

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D.   computer means (100) connected to said
signal processing circuits for receiving said sequence
of binary digital signals, said computer means having
memory means (114) for storing system design
5   parameters, said computer means programmed to
periodically calculate other operating parameters of
said refrigeration system including

        a.   relative humidity,

        b.   efficienty of said refrigeration
10   system,

        c.   temperature difference between said
        condensing temperature and said wet
        bulb temperature,

        d.   difference between said head pres-
15   sure and said suction pressure, and

        e.   the cost to operate said refrigera-
        tion system;

E.   control relays (108) connected to said
computer means for controlling, in said refrigeration
20 system,

        a.   condenser fans in response to
        calculated value of said difference
        between said condensing temperature
        and said wet bulb temperature,

25   b.   condenser pumps responsive to the
        calculated value of said difference
        between said condensing temperature
        and said wet bulb temperature, and

        c.   compressor loading responsive to
30   calculated value of said difference
        between said head pressure and said
        suction pressure; and

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     F.   first   alarm means (160,   161,   162) responsive to a deviation of said head pressure relative to said condensing temperature to indicate presence of non-condensible gases in said
5 refrigeration system, and second alarm means (160, 161, 162) responsive to said difference between said condensing temperature and said wet bulb temperature to indicate fouled condensers in said refrigeration system.

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2. An energy management system for a double stage refrigeration system having evaporative type condensers, booster compressors, and main compressors, comprising:

5   A. a plurality of electrical temperature sensors (46, 66, 74, 76, 80) attached to said refrigeration system, said sensors producing electrical signals representative of the following operating parameters of said refrigeration system,

10   a.   wet bulb temperature,
          b.   dry bulb temperature,
          c.   condensing temperature,
          d.   evaporator air temperature,
          e.   evaporator refrigerant temperature,
15   and
          f.   ambient temperature;

B. a plurality of electrical pressure sensors (72, 82, 86) attached to said refrigeration system, said pressure sensors producing electrical
20 signals representative of the following operating parameters of said refrigeration system

          a.   head pressure,
          b.   intermediate suction pressure, and
          c.   booster suction pressure;

25   C. signal processing circuits (102, 104, 106) connected to said temperature sensors and said pressure sensors for receiving said respective electrical signals therefrom for producing a sequence of binary digital signals representative of said
30 respective electrical signals;

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D.   computer means (100) connected to said
signal processing circuits for receiving said sequence
of binary digital signals, said computer means having
memory means (114) for storing system design
5 parameters, said computer means programmed to
periodically calculate the operating parameters of
said refrigeration system

            a.   relative humidity,

            b.   efficiency of said refrigeration
10   system,

            c.   temperature difference between said
                  condensing temperature and said wet
                  bulb temperature,

            d.   difference between said head
15   pressure and said booster suction
                  pressure,

            e.   the cost to operate said refrigera-
                  tion system, and

            f.   difference between said evaporator
20   air temperature and said evaporator
                  refrigerant temperature;

     E.   control relays (108) connected to said
computer means for controlling, in said refrigeration
system,

25   a.   condenser fans in response to the
                  calculated value of said difference
                  between said condensing temperature
                  and said wet bulb temperature,

            b.   condenser pumps responsive to the
30   calculated value of said difference
                  between said condensing temperature
                  and said wet bulb temperature,

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     c.   compressor loading responsive to calculated value of said difference between said head pressure and said booster suction pressure; and

5   F. first alarm means (160, 161, 162) responsive to a deviation of said head pressure relative to said condensing temperature to indicate presence of non-condensible gases in said refrigeration system, and second alarm means (160, 10 161, 162) responsive to said difference between said condensing temperature and said wet bulb temperature to indicate fouled condensers in said refrigeration system.

     3. The system as defined in claim 1 or 2 which 15 further comprises display means (163, 165) connected to said computer means for displaying said operating parameters including said other operating parameters.

     4. The system as defined in claim 1 or 2 in which said temperature sensors include a resistance 20 element having a change in resistance with temperature.

     5. The system as defined in claim 1 or 2 in which said pressure sensors are sources of current in which said current varies with pressure applied to 25 said sensors.

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6. The system as defined in claim 1 or 2 in which said signal processing circuits include:

analog multiplexer (102) for selectively scanning said plurality of electrical sensors (142);

5   programmable gain amplifier (104) connected to said analog multiplexer for sequentially receiving respective electrical signals from said electrical sensors, said amplifier controllable by said computer means to provide scaling of said electrical signals;

10 and

analog-to-digital converter (106) connected to receive amplified electrical signals from said amplifier and to convert said amplified signals to said sequence of binary digital signals.

15   7. The system as defined in claim 1 or 2 in which:

said computer means includes a microprocessor (125) programmed to perform said periodic calculations and to activate said control relays when required

20 responsive to said calculations; and

said memory means includes random access memory (126), programmable read only memory (127), and parameter memory (114), said parameter memory having a parameter patch panel (121) to permit insertion of

25 system specific data for said refrigeration system, said parameter memory thereby adapted to be modified on occasion.

8. The system as defined in claim 7 in which said computer means further includes printer control

30 logic (133) and external printer (112) for periodically logging refrigeration system operating parameters and controls effected.

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9. The system as defined in claim 1 or 2 in which said control relays include override means (172) for manually disabling said control of said relays by said computer.

5   10. The system as defined in claims 1 or 2 in which said control relays include means (172) for automatically overriding said relays when a failure in said energy management system occurs.

11. The system as defined in claim 1 or 2 in 10 which said first and second alarm means include visual (161) and audible (162) alarms.

12. The system as defined in claim 2 which further comprises evaporator defrosting means (176, 177, 178, 179, 180) responsive to said difference 15 between said evaporator air temperature and said evaporator refrigerant temperature.

13. The system as defined in claim 12 in which the design value of said difference between said evaporator air temperature and said evaporator 20 refrigerant temperature is stored in said memory means, and said evaporator defrost means is activated when said difference between said evaporator air temperature and said evaporator refrigerant temperature is greater than said design difference.

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14. The system as defined in claim 13 in which:

a low threshold difference between said evaporator air temperature and said evaporator refrigerant temperature is stored in said memory
5 means; and

said control relays include control of the evaporator fans (175) in said refrigeration system, said control relays deenergizing said evaporator fans when said calculation of the difference between said
10 evaporator air temperature and said evaporator refrigerant temperature is less than said stored lower threshold difference.

15. The system as defined in claim 2 in which:

said control relays control loading and
15 unloading of the main compressors (88) in said refrigeration system; and

said computer means is programmed to periodically calculate an optimum value of said intermediate pressure from said sequences of binary digital signals
20 representative of said head pressure and said booster suction pressure, said computer thereby controlling the loading of said main compressors to produce said optimum intermediate pressure.

16. The system as defined in claim 15 in which
25 said calculation of optimum value is in accordance with the relationship that the absolute value of said optimum intermediate pressure is the absolute booster suction pressure multiplied by the square root of the quotient of the absolute head pressure and the
30 absolute suction pressure.

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17. The system as defined in claim 1 or 2 in which said control relays deenergize said condenser pumps when said wet bulb temperature signals indicate freezing conditions in said condensers.

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18. An energy management system for a refrigera-
tion system or the like having evaporative condensers
comprising:

5   first temperature sensing means (46, 66) for
producing a first signal representative of the wet
bulb temperature in and around said condensers;

second temperature sensing means (74) for
producing a second signal representative of the
condensing temperature of said refrigeration system;

10   pressure sensing means (72) for producing a
third signal representative of the head pressure of
said refrigeration system;

memory means (114) for storing pressure-
temperature data representative of the head
15 pressure/condensing temperature relationship of the
refrigerant of said refrigeration system, and a design
difference-temperature between the design wet bulb
temperature and the design condenser temperature; and

calculation means (100) connected to said
20 first and second temperature sensing means for
receiving said first and second signals, to said
pressure sensing means for receiving said third signal
and to said memory means, said calculation means for
comparing the sensed head pressure and the sensed
25 condensing temperature to said pressure-temperature
data in said memory, and for calculating the
difference between said sensed wet bulb temperature
and said sensed condensing temperature, and for
comparing such difference temperature to said design
30 difference-temperature stored in said memory, said
calculation memory means thereby determining the
efficiency of said refrigeration system.

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19. The system as defined in claim 18 which includes display means (110) for displaying the value of such calculated efficiency.

20. The system as defined in claim 18 in which said calculation means initiates an alarm indicating a fouled condenser when such temperature difference is greater than the design difference temperature.

21. The system as defined in claim 18 in which said calculation means initiates a non-condensible gases alarm when the relationship of the sensed values of head pressure and condensing temperature deviate from the relationship indicated in said stored pressure-temperature data.

22. The system as defined in claim 18 in which said first temperature sensing means is a resistance bulb, whose resistance varies with temperature, disposed adjacent to an evaporative condenser in said refrigeration system, and said second temperature sensing means is a resistance bulb, whose resistance varies with temperature, disposed in contact with condensed refrigerant in said refrigeration system.

23. The system as defined in claim 18 in which said pressure sensing means is a pressure transducer producing a current proportional to the pressure thereon, said transducer connected to the refrigerant line between said condensers and the receiver of said refrigeration system.

24. The system as defined in claim 18 in which
said calculation means includes means (143, 153) for
displaying the calculated difference temperature.

25. The system as defined in claim 18 in which
5   said calculation means controls relay means (170) for
deenergizing condenser fans and condenser pumps when
the calculated temperature difference is less than a
preselected lower value.

26. The system as defined in claim 18 in which
10 said calculation means controls relay means (170) for
deenergizing said condenser pumps when the sensed wet
bulb temperature indicates a freezing value.

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27. An energy management system for a single
stage refrigeration system or the like having
evaporative condensers comprising:

first pressure transducer means (72)
5 connected to the liquid refrigerant line between said
condensers and the receiver of said refrigeration
system for producing an electrical signal proportional
to head pressure of said refrigeration system;

pressure transducer means (86) connected to
10 the suction line of the compressors of said
refrigeration system for producing an electrical
signal proportional to the suction pressure at said
compressor;

memory means (114) for storing a
15 predetermined design cost for operation of said
refrigeration system under maximum load; and

calculation means (100) connected to said
first and second pressure transducers and to said
memory for calculating the pressure difference between
20 said head pressure and said suction pressure, for
comparing such difference to said stored design cost,
and thereafter calculating the operating costs of said
refrigeration system when the load on said
refrigeration system varies from the maximum load.

25   28. The system as defined in claim 27 in which
said calculated operating cost is displayed.

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29. An energy management system for a two-stage refrigeration system or the like having evaporative condensers, booster compressors, and main compressors, comprising:

5   first pressure transducer means (72) connected into said refrigeration system to produce a first electrical signal representative of the head pressure;

       second pressure transducer means (86)
10 connected into said refrigeration system to produce a second electrical signal representative of the intermediate suction pressure;

       third pressure transducer means (82) connected in said refrigeration system to produce a
15 third electrical signal proportional to the booster suction;

       memory means (114) for storing the preselected design value of booster suction pressure; and

20   calculation and control means (100, 108) connected to said first, second, and third pressure transducers and to said memory, said calculation and control means for comparing said booster suction pressure to said stored design booster suction
25 pressure, said calculation and control means controlling loading and unloading of said booster compressor to achieve said design booster suction pressure.

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30.   The system as defined in claim 29 in which said calculation and control means calculates an optimum intermediate suction pressure from said first and third electrical signals from said first and third
5 pressure transducers, said calculation and control means controlling said main compressors to achieve said optimum intermediate suction pressure.

31. The system as defined in claim 30 which further comprises:
10   a hot gas defrosting system (176, 177, 178, 180, 179) associated with the evaporator of said refrigeration system;
        a first temperature sensing bulb (78) disposed in said evaporator to produce a fourth
15 electrical signal representative of the air temperature in said evaporator;
        second temperature sensor bulb (80) disposed in contact with the refrigerant in said evaporator for producing a fifth electrical signal proportional to
20 the refrigerant temperature;
        a preselected design difference-temperature value between said air temperature and said refrigerant temperature stored in said memory means;
        said calculation and control means connected
25 to said first and second temperature sensor bulbs, said calculation and control means for calculating the temperature difference between said evaporator air temperature and said evaporator refrigerant temperature and thereafter comparing such calculated
30 difference · to said stored design difference, said calculation and control means energizing said hot gas defrosting system when such calculated difference is greater than said stored design difference.

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32.   The system as defined in claim 31 in which:

a preselected lower value threshold of the temperature difference between said evaporator air temperature and said evaporator refrigerant temperature is stored; and

said calculation and control means deenergizing the evaporator fans when such calculated difference is less than said stored lower threshold value.

33. In a single stage refrigeration system having evaporative condensers, the method of managing energy in the system to minimize operating costs comprising the steps of:

5   storing in a memory the pressure and condensing temperature characteristics of the refrigerant, the design value of the wet bulb temperature in the condensers, the design value of condensing temperature, the design value of the head

10 pressure, the design value of the suction pressure, and the cost per ton per hour for maximum load;

            monitoring electrically the operational para-meters of dry bulb temperature, wet bulb temperature, head pressure, condensing temperature, and suction

15 pressure;

            processing the monitored electrical signals for input to a digital computer;

            periodically calculating in the computer the relationship of the head pressure and the condensing

20 temperature during operation of the refrigeration system;

            comparing such relationship to the stored characteristics of the refrigerant;

            energizing a non-condensible gases alarm when

25 such relationship deviates from the stored relation-ship;

            calculating in the computer the temperature difference between the condensing temperature and the wet bulb temperature;

30   comparing such temperature difference to the difference between the stored design condensing temperature and design wet bulb temperature;

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energizing a fouled condenser alarm when such calculated temperature difference is greater than such stored design difference;

deenergizing condenser fans and condenser
5   pumps when such temperature difference is a selected value less than such stored difference;

calculating in the computer the pressure difference between the head pressure and the suction pressure;

10   comparing such pressure difference to the difference between the stored design head pressure and stored design suction pressure;

correcting the stored cost per ton per hour for a maximum load when such calculated pressure
15   difference is less than such stored difference; and

displaying the corrected cost per ton per hour.

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34. In a two stage refrigeration system having booster compressors, main compressors, and evaporative condensers, the method of managing energy in the system to minimize operating costs comprising the

5   steps of:

storing in a memory the pressure and condensing temperature characteristics of the refrigerant, the design value of the wet bulb temperature in the condensers, the design value of

10 condensing temperature, the design value of the head pressure, the design value of the booster suction pressure, the design value of the intermediate suction pressure, and the cost per ton per hour for maximum load;

15 monitoring electrically the operational parameters of dry bulb temperature, wet bulb temperature, head pressure, condensing temperature, booster suction pressure, intermediate suction pressure, evaporator air temperature, and evaporator refrigerant

20 temperature;

processing the monitored electrical signals for input to a digital computer;

periodically calculating in the computer the relationship of the head pressure and the condensing

25 temperature during operation of the refrigeration system;

comparing such relationship to the stored characteristics of the refrigerant;

energizing a non-condensible gases alarm when

30 such relationship deviates from the stored relationship;

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calculating in the computer the temperature difference between the condensing temperature and the wet bulb temperature;

comparing such temperature difference to the difference between the stored design condensing
5   temperature and design wet bulb temperature;

energizing a fouled condenser alarm when such calculated temperature difference is greater than such stored design difference;

deenergizing condenser fans and condenser
10   pumps when such temperature difference is a selected value less than such stored difference;

calculating in the computer the pressure difference between the head pressure and the booster suction pressure;

15   comparing such pressure difference to the difference between the stored design head pressure and stored design booster suction pressure;

correcting the stored cost per ton per hour for a maximum load when such calculated difference is
20   less than such stored difference; and

displaying the corrected cost per ton per hour.

35. The method as defined in claim 33 or 34 which further comprises the steps of:
25   calculating in the computer the efficiency of the refrigeration system when the relationship of the monitored head pressure and condensing temperature deviate from the stored relationship; and

displaying the calculated efficiency.

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36. The method as defined in claim 33 or 34 which further comprises the steps of:

storing in the memory a preselected lower threshold of the temperature difference between the
5 condensing temperature and the wet bulb temperature;

comparing the calculated temperature difference between the condensing temperature and the wet bulb temperature with the stored lower threshold temperature difference;

10   · deenergizing condenser fans and condenser pumps when such calculated temperature difference is less than the stored lower threshold temperature difference; and

displaying the calculated temperature
15 difference.

37. The method as defined in claim 33 or 34 in which the step of monitoring the wet bulb temperature includes the step of deenergizing condenser pumps when the monitored wet bulb temperature indicates freezing
20 conditions in the condensers.

38. The method as defined in claim 34 which
further comprises the steps of:

       calculating in the computer from the
monitored head pressure and the monitored booster

5   suction pressure the optimum value of the intermediate
suction pressure;

       comparing such calculated optimum suction
pressure value to the monitored intermediate suction
pressure value; and

10   controlling the loading on the main compres-
sors when the monitored intermediate suction pressure
is not equal to the calculated optimum intermediate
suction pressure to thereby optimize the imtermediate
suction pressure.

15   39. The method as defined in claim 34 which
further comprises the steps of:

       comparing the monitored booster suction
pressure to the stored design value of booster suction
pressure; and

20   controlling the loading of the booster
compressors when the monitored value of booster
suction pressure is not equal to the stored design
value thereof to thereby optimize the booster suction
pressure.

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40. The method as defined in claim 34 which further comprises the steps of:

storing in the memory a design value of the difference between the evaporator air temperature and 5 the evaporator refrigerant temperature;

periodically calculating in the computer the temperature difference between the monitored evaporator air temperature and the monitored evaporator refrigerant temperature;

10   comparing such evaporator temperature difference to the stored design evaporator temperature difference; and

deenergizing the evaporator fans and initiating a defrost cycle in the evaporator when the 15 evaporator temperature difference is greater than the stored design evaporator temperature difference.

41. The method as defined in claim 40 which further comprises the steps of:

storing in the memory a predetermined lower 20 limit of the difference between the evaporator air temperature and the evaporator refrigerant temperature;

deenergizing the evaporator fans when the calculated evaporator temperature difference is less 30 than the stored lower limit of the evaporator temperature difference; and

displaying the calculated evaporator temperature difference.

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FIG. 1

FIG. 2

FIG.3

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FIG. 4

FIG. 5

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FIG. 6

SENSORS — 142

ANALOG MULTIPLEXER — 104

SIGNAL CONDITIONING AMPLIFIER — 102

A/D CONVERTER — 106

COMPUTER INTERFACE — 136

105 — TO COMPUTER

FIG. 7

110

FROM COMPUTER — 111

COMPUTER INTERFACE — 139

DISPLAY MEMORY & DRIVER — 143

DISPLAY MEMORY & DRIVER — 144

ALARM DRIVER — 160

DISPLAY DRIVER — 163

RELAY MONITER DISPLAY — 164

ALARM DISPLAY — 161

AUDIO ALARM — 162

DISPLAY MEMORY & DRIVER — 165

T_WB DISPLAY — 150

T_DB DISPLAY — 151

T_C DISPLAY — 152

ΔT_C DISPLAY — 153

T_A DISPLAY — 154

T_R DISPLAY — 155

ΔT_E DISPLAY — 156

P_H DISPLAY — 157

P_MS DISPLAY — 158

P_BS DISPLAY — 159

COST/TON/HR DISPLAY — 166

HUMIDITY DISPLAY — 167

EFF. DISPLAY — 168

FIG. 8

170

SOLID STATE RELAYS

TO CONTROLLED ELEMENTS

COMPUTER INTERFACE — 137

FROM COMPUTER — 107

MANUAL OVERIDE RELAYS — 172

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FIG. 9.

I. CLASSIFICATION OF SUBJECT MATTER (If several classification symbols apply, indicate all) [3]

According to International Patent Classification (IPC) or to both National Classification and IPC

INT. CL.[3] F25 B 49/00
U.S. CL.   62 /126

II. FIELDS SEARCHED

Minimum Documentation Searched [4]

Classification System	Classification Symbols
U.S.	62/176.C,171,126,127,475,DIG.17,129,156

Documentation Searched other than Minimum Documentation
to the Extent that such Documents are Included in the Fields Searched [5]

III. DOCUMENTS CONSIDERED TO BE RELEVANT [14]

Category [°]	Citation of Document, [16] with Indication, where appropriate, of the relevant passages [17]	Relevant to Claim No. [18]
X	US,A, 4,133,338 Published 9 January 1979 Honikman.	17
X	US,A, 3,082,951 Published 26 March 1963 Kayan	19
X	US,A, 4,215,554 Published 5 August 1980 Pohl	12
X	US,A, 4,161,106 Published 17 July 1979 Savage	1
A	US,A, 4,193,781 Published 18 March 1980 Vogel et al	1-41
A	US, A, 4,146,085 Published 27 March 1979 Wills	1-14
A	US,A, 3,707,851 Published 2 January 1973 Mc Ashan, Jr.	1-41

° Special categories of cited documents: [16]

"A" document defining the general state of the art

"E" earlier document but published on or after the international filing date

"L" document cited for special reason other than those referred to in the other categories

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but on or after the priority date claimed

"T" later document published on or after the international filing date or priority date and not in conflict with the application, but cited to understand the principle or theory underlying the invention

"X" document of particular relevance

IV. CERTIFICATION

Date of the Actual Completion of the International Search [2]	Date of Mailing of this International Search Report [2]
19 May 1982	2 8 MAY 1982
International Searching Authority [1] ISA/US	Signature of Authorized Officer [10] William E. Wayner