March 26, 1963 C. F. KAYAN 3,082,951

METHOD FOR CALCULATING PERFORMANCE OF REFRIGERATION APPARATUS

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FIG.1

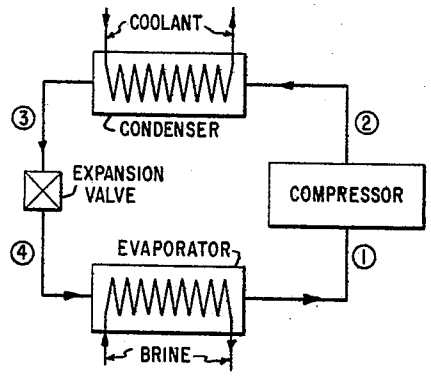
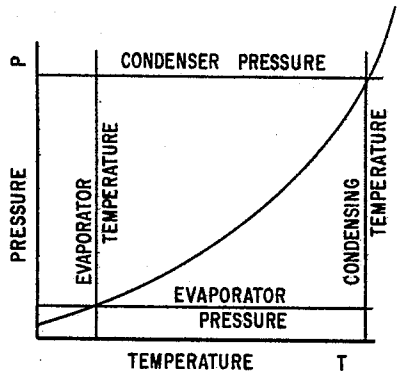


FIG.2



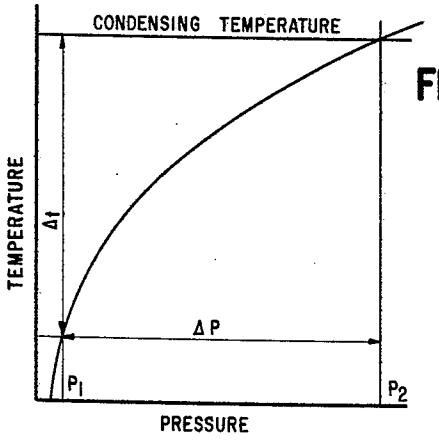
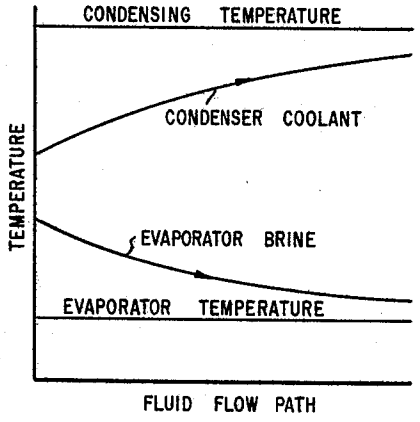


FIG.3

FIG.4



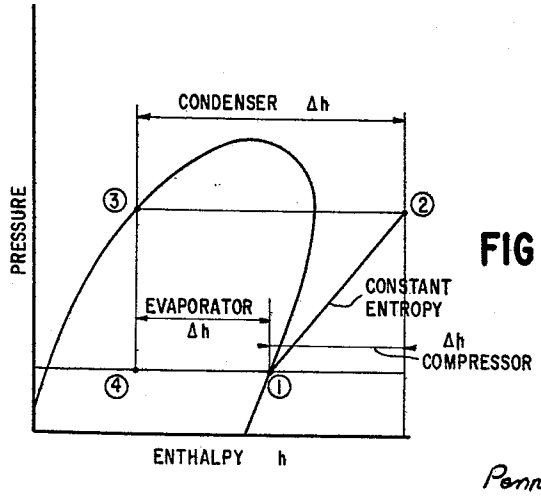


FIG.5

INVENTOR.
CARL F. KAYAN

BY
Pennie Edmonds Morton Barrows & Taylor
ATTORNEYS

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FIG.6

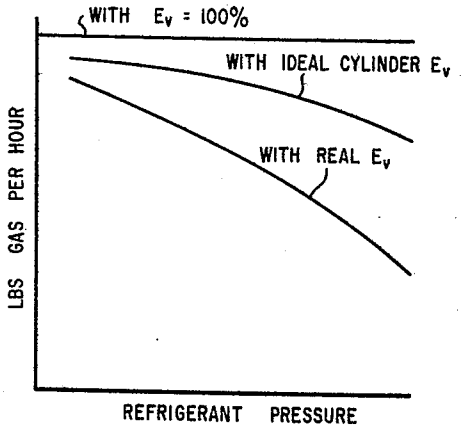


FIG.7

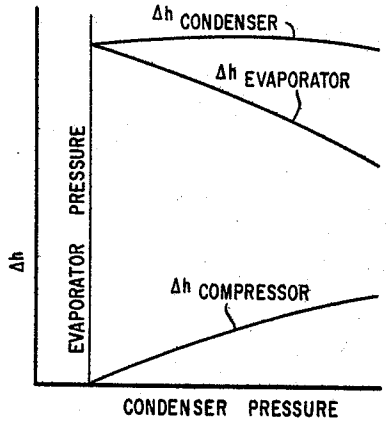


FIG.8

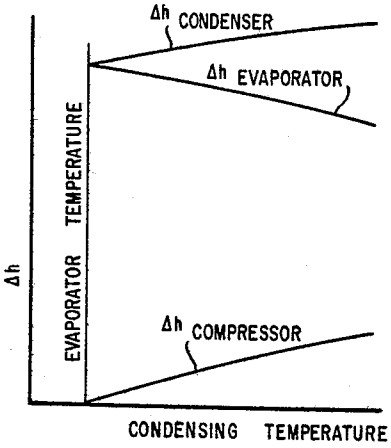
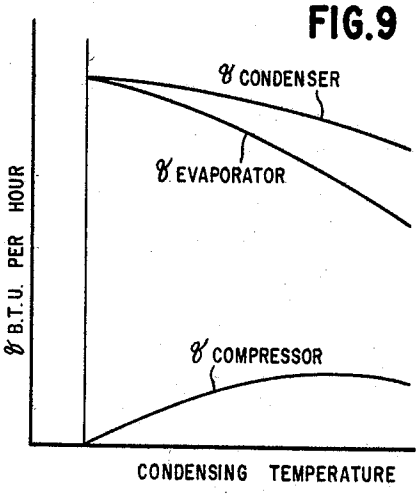


FIG.9



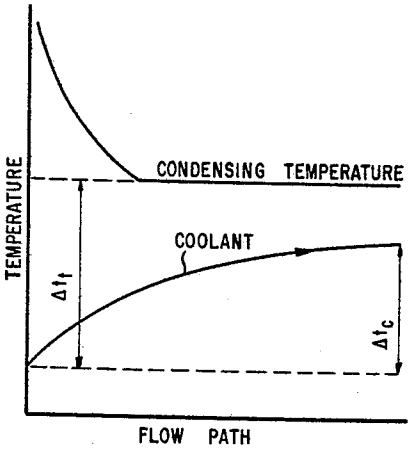


FIG.10

INVENTOR.
CARL F. KAYAN

BY

Pennie Edmonds Morton Barrows & Taylor

ATTORNEYS

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FIG. 11

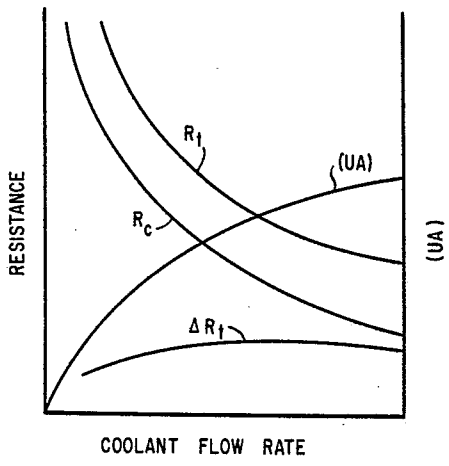


FIG. 12

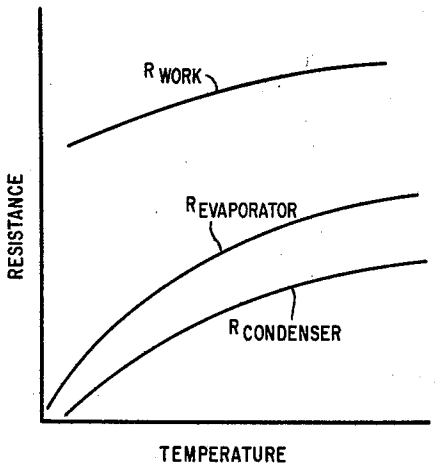
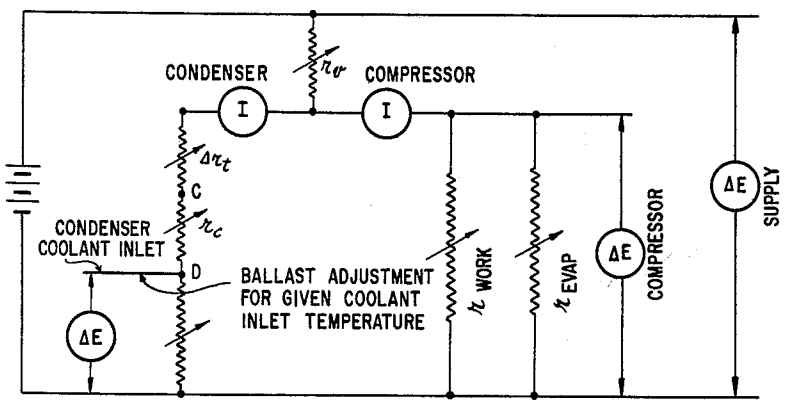


FIG. 13



INVENTOR
CARL F. KAYAN

BY Pennie Edmonds Morton Barrows & Taylor
ATTORNEYS

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FIG.14

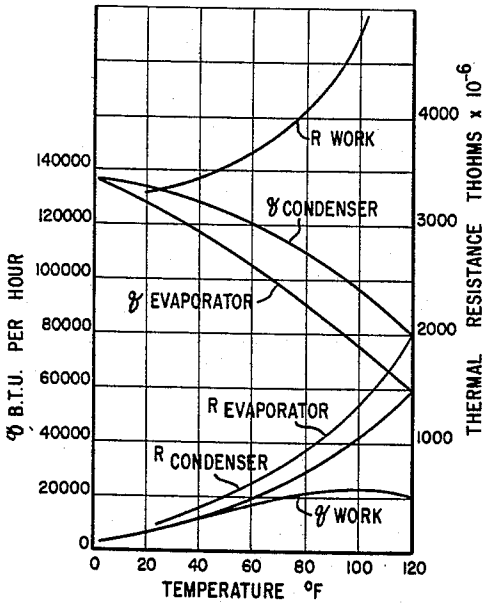
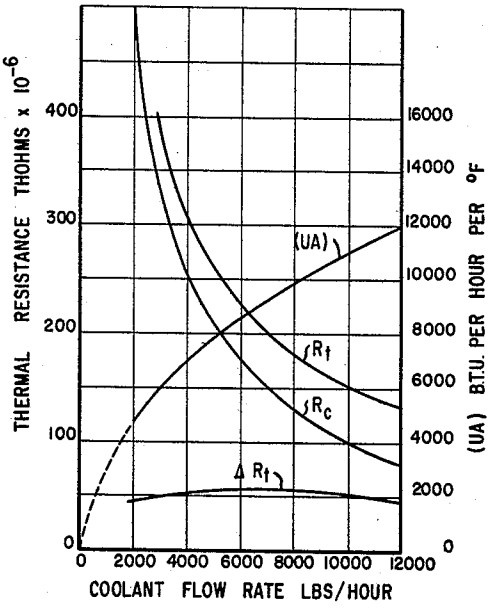


FIG.15



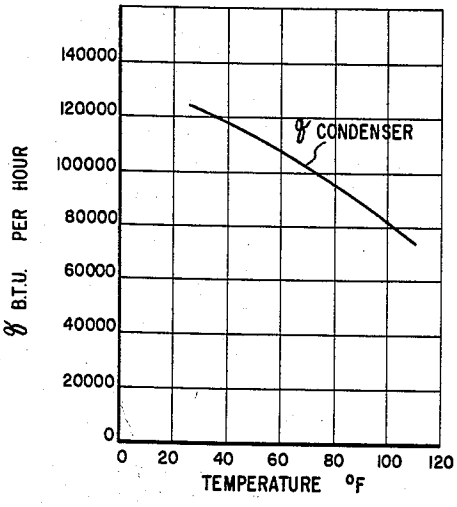


FIG.16

INVENTOR
CARL F. KAYAN

BY Pennie Edmonds Morton Barrows & Taylor
ATTORNEYS

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3,082,951

METHOD FOR CALCULATING PERFORMANCE OF REFRIGERATION APPARATUS

Carl F. Kayan, % Columbia University, Dept. of Mechanical Engineering, Engineering Center, New York 27, N.Y.

Filed July 1, 1953, Ser. No. 365,353
2 Claims. (Cl. 235—184)

This invention relates to method and apparatus for calculating the performance of refrigerating apparatus and particularly to the application of the electrical Analogger to such purpose.

Refrigerant-vapor compression systems, whether for heat-pump or for refrigerating purposes, will operate in steady state at some equilibrium condition determined by relative component performance characteristics. This means that the condensing temperature and the compressor delivery pressure, under fixed evaporator conditions, are the result of comparative performance characteristics of both the condenser and the compressor, as subject to their imposed conditions. Thus, variations in the condenser coolant conditions will be reflected in a variation of the condensing temperature; this, in turn, will affect the resultant compressor performance.

These interdependent relationships for the system may be investigated by electrical Analogger methods, wherein system performance values may be represented by electrical quantities.

This tool, in general, may serve for the ready determination of overall performance characteristics in the analysis of composite integrated systems comprising fluid machines and associated heat transfer equipment under various imposed conditions and heat load.

In the accompanying drawings I have shown a typical refrigerating system and an electric Analogger set up to calculate the performance of such system and it will be understood that my invention is not limited to the set-up there shown but may be modified in accordance with the principles hereinafter disclosed for calculating the performance of many different systems involving the heat-pump process.

In the said drawings:

FIG. 1 shows diagrammatically the essential apparatus of a typical refrigerating system,

FIGS. 2 to 12 inclusive are charts illustrating the temperature-pressure of the system under various operating conditions, as will be later described,

FIG. 13 shows my improved electric Analogger for calculatinug the performance of the refrigerating system illustrated in FIG. 1, and

FIGS. 14, 15 and 16 are charts illustrating the application of my improved method to a typical problem.

The conventional refrigerant-vapor compression system, in terms of a simple component circuit, is illustrated in FIG. 1. Power is supplied to run the compressor; heat is abstracted in the condenser through the medium of a coolant such as water flowing at a given rate; heat is supplied to the evaporator by brine of a refrigerating system, by chilled-water of an air-conditioning system, by atmospheric air flow of an air-source heat pump system, or perhaps by waste warm liquid in an industrial heat pump process.

The pressure-temperature relations for a refrigerant are indicated in FIG. 2, with pressure-temperature levels of

system operation shown in FIG. 3, and general corresponding fluid heat-transfer temperatures indicated in FIG. 4.

In the following description and the accompanying charts, the letters and symbols used have the values indicated below.

A=heat transfer surface area, square feet.
C=clearance fraction, based on displacement.
c=coolant means specific heat, B.t.u. per (pound) (Fahrenheit degree).
E_v=volumetric efficiency.
n=exponent in Pv^n=Constant (ideally $n=k=Cp/Cv$.)
P=pressure, pounds per square inch, absolute.
ΔP=pressure difference between evaporator and condenser, pounds per square inch.
$P1$=intake pressure of compressor, pounds per square inch.
$P2$=delivery pressure of compressor, pounds per square inch.
q=heat energy as applied to different elements of the system and identified by subscripts, B.t.u. per hour.
R_c=heat exchange resistance, thohms=Fahrenheit degrees per (B.t.u. per hour).
R_t=equivalent thermal resistance for condenser process, Fahrenheit degrees per (B.t.u. per hour).
R_{UA}=net heat transfer resistance, thohms=$1/(UA)$ Fahrenheit degrees per (B.t.u. per hour).
r=electrical resistance corresponding to thermal ohms.
Δ_t=temperature difference between evaporator and condenser saturation values, Fahrenheit degrees.
Δt_t=temperature difference between condensing temperature and inlet coolant temperature on condenser, Fahrenheit degrees.
U=overall heat transfer conductance, B.t.u. per (hour) (square foot) (Fahrenheit degree).
W=pounds coolant per hour.

Pressure difference ΔP and the corresponding temperature difference Δt effected by the thermodynamic system are likewise indicated in FIG. 3. The pressure-enthalpy ($P-h$) diagram for an ideal refrigerant cycle is shown in FIG. 5 with particular reference to component anthalpy differences Δh, i.e., for evaporator, compressor, and condenser.

For the purposes of this specification a positive displacement compressor system with the usual machine suction and discharge valves is assumed. Equivalent analysis, however, could be carried out with other types of compressor systems. The performance of the compressor depends on its construction features, i.e., the ideal displacement gas volume is modified by the effect of the overall volumetic efficiency. Clearance in conjunction with pressure ratio, pressure drops in valves, suction gas heating, as well as the specific gas properties, directly affect the effective gas handling capacity, as measured by real volumetric efficiency. On an idealized basis, without considering pressure drops through suction and delivery valves, and suction heating, the volumetric efficiency may be shown as:

$$E_v=1+C-C(P2/P1)^{1/n} \quad (1)$$

The ideal volumetric efficiency, though itself less than 100 percent, is still further reduced by the practical effects of valve drops and heating of suction gas. FIG. 6 shows the general variation of the hourly mass of gas handled vs. gas delivery pressure, for E_v=100 percent (unimpaired

displacement), ideal volumetric efficiency as given by Equation 1, and the real volumetric efficiency of a practical machine. Fixed suction pressure is assumed.

Based on fixed suction conditions such as suction pressure and suction gas quality, and variable discharge pressures, the ideal compressor work per pound of refrigerant (at constant entropy) may be calculated. Similarly, based on the assumption of liquid refrigerant leaving the condenser at the temperature corresponding to the condensing pressure and entering the pressure drop device (expansion valve) at the same condition, values of the heat per pound given up in the condenser may likewise be calculated; and also for the heat absorbed in the evaporator. Such values for a fixed evaporator pressure are shown plotted against cycle condensing pressure in FIG. 7, and against corresponding condensing temperature in FIG. 8.

Using values of pounds gas pumped per hour from FIG. 6 as based on real volumetric efficiency, and combined with the data of FIG. 8, equivalent hourly heat values representing practical performance are obtained and shown in FIG. 9. It is immediately evident that the heat to be rejected in the condenser is made up of that acquired in the evaporator plus that contributed by the work effect accomplished in the compressor.

Disregarding the effect of superheat on condenser performance, the flow of heat in a refrigerant condenser may be considered to vary with the temperature difference between the condensing temperature and the inlet coolant temperature, with fluid temperature relationships as shown in FIG. 10. The hourly heat transfer may be shown in terms of resistance:

$$q=\Delta t_t/R_t \tag{2}$$

$$R_t=\frac{R_c}{(1-1/e^{R_c/R_{UA}})} \tag{3}$$

$$R_c=I/(Wc) \tag{4}$$

$$\Delta R_t=R_t-R_c \tag{5}$$

These relationships are shown on a general basis in FIG. 11, with values plotted against coolant flow rate, W.

Considering the energy relationships for a compressor system as shown in FIG. 9, just as the resistance concept has been extended to cover heat exchanger transfer due to overall temperature difference, so may it similarly be extended to cover energy flow in a machine, as for example, in a compressor. Equivalent resistance values may be used to represent the compressor performance of the present system. Assuming a fixed evaporator pressure and corresponding temperature, and a variation of compressor performance in terms of the overall temperature difference on the system, resistance values may be obtained by dividing the temperature difference by hourly heat quantities:

$$R=\Delta t/q \tag{6}$$

FIG. 12 shows resistance values plotted against temperature difference, as follows:

R_{evap}=resistance value corresponding to hourly evaporator heat.

R_{cond}=resistance value corresponding to hourly condenser heat.

R_{work}=resistance value corresponding to hourly compressor work.

Using electrical resistance values proportional to the thermal resistances cited in FIGS. 11 and 12, an electrical Analogger circuit may be arranged to facilitate ready evaluation of heat pump or refrigeration system performance under different operating conditions. Although the use of other driving forces such as enthalpy differences could be considered, temperature difference seems to offer the greatest advantage, with electrical voltage as its counterpart. Furthermore, as far as the compressor system is concerned, although a variable voltage electrical generator might be employed, the simplest approach appears to be to use an impressed overall voltage as in the case of the condenser system.

It is of course obvious that whatever heat is discharged from the compressor system must in turn flow through the condenser system. The temperature difference that the condenser heat flow requires under the condenser fluid conditions must in turn set the condensing vapor temperature; this in turn determines the corresponding compressor delivery pressure. The actual compressor system energy, depending on real volumetric efficiency and hourly displacement, is directly dependent on the delivery pressure.

In setting up the Analogger circuit, separate adjustable resistances, in parallel, and representing the combined compressor work and the evaporator phases, are employed. The sum-total current contributed by these two branches must in turn equal the total flowing in the condenser circuit, with the same top potential in effect for both phases. Potential level corresponding to the condenser inlet coolant temperature is maintained on the condenser circuit, using an adjustable ballast resistance. (If, beyond the condenser, the evaporator circuit is to be detailed separately, there must be an additional parallel circuit whose flow must obviously correspond to that for the work-component on the compressor side. Thus the circuit below the condenser would have two branches.)

FIG. 13 shows the simulation circuit, with resistances for R_c and ΔR_t adjustable in accordance with coolant flow rate derived from a working curve such as FIG. 10 for the specific compressor-matching condenser of the system. Fluid temperature corrections may be introduced into FIG. 10 if desired. Also shown in FIG. 13 are resistances for the compressor-work and the evaporator-energy components, adjustable to keep step with the overall value of the compressor-plant potential, in conformance with the appropriate compressor-plant performance characteristic as shown in FIG. 12.

Adjustment of the circuit is made by changing the top-side potential value through the variable resistance r_v, such that the condenser current Icond equals the total compressor-energy current Icomp, and with parallel minor corrections made in r_{work} and r_{evap} in accordance with the operating overall potential differences (FIG. 12). The effect of the coolant entering temperature of a given value is achieved by maintaining the potential at point D at proper value. Coolant outlet temperature is determined by potential measurement at point C on the condenser circuit. The various current meters show the equivalent energy flowing at any given point. Adjustment of the different elements of the Analogger for this type of heat-power system is a matter of manual dexterity, somewhat simplified by the use of automatic equipment, such as, for example, a voltage regulator.

FIGS. 14 and 15 show data for a given vapor-compression system, with ammonia as the refrigerant and the evaporator operated at a fixed temperature of 0 F. With allowance for real volumetric efficiency as illustrated in FIG. 6, hourly heat values as typically illustrated in FIG. 9 are shown in FIG. 14. Corresponding resistance values as calculated according to Equation 6 are likewise shown on the diagram, as thohms. FIG. 15 shows values for the condenser typified by FIG. 11, and calculated according to Equations 4 and 5. Values for the product UA are included, all values being plotted against hourly rate of condenser coolant (water) circulation. (Compressor displacement, c.f.m.=37.3; condenser heat transfer surface=56.2 sq. ft.)

Circuit values, as translated from FIGS. 14 and 15 for the arrangement of FIG. 13, are on the basis of 1 ohm electrical resistance=10^{-6} thohms, 0.10 volt=1 F; 0.100 amp.=1,000,000 B.t.u. per hr.

FIG. 16 shows the variation of condenser heat, in B.t.u. per hr., vs. inlet water temperature, with a fixed flow rate of 6,000 lbs. water per hour for the condenser with evaporator at 0 F.

I claim:

1. Method of calculating the performance of refrigerating systems embodying heat transfer apparatus to and from a circulating fluid, which consists in providing an electric circuit with two branches, each branch containing adjustable resistanes, adjusting the resistances of one branch to represent the calculated performance characteristics of the apparatus from which the fluid absorbs heat, adjusting the resistances of the other branch to represent the calculated performance characteristics of the apparatus to which heat is delivered and noting the current values of the two branches and of the individual circuits containing the adjustable resistances.

2. The method of claim 1 wherein selected circuits containing resistances representing performance characteristics of apparatus which may be altered by adjustment of such apparatus are then adjusted to obtain equal current through the two branches.

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