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Simulation of Condition Sequence During Start-Up of an Evaporation Refrigerating System

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RESUME

A traditional evaporation refrigerating system with hermetic compressor and capillary tube displays a complicated condition sequence during start-up from a pressure-equalized condition.

Since maximum load on the compressor, as regards both torque demand and mechanical load, occurs during start-up it is important that the operation of the refrigerating system is such that the compressor is not loaded too severely. Additionally, it is desirable to hold motor breakdown torque as low as possible in order to obtain a maximum motor efficiency approximately corresponding to the normal operating condition of the system.

Danfoss have for some time been working on the matching of compressor and system characteristics. To help in this work they have developed a condition diagram in which both maximum compressor capacity (stall characteristic) and maximum system load on the compressor (system characteristic) are delineated.

The stall characteristic of a compressor can be determined theoretically or by measurement. Conversely, up till now, it has only been possible to determine the system characteristic by measurement. To be able to determine this characteristic by simulation and to be able to more closely study the condition sequence during start-up a dynamic simulation model of an evaporation refrigerating system has been developed.

The descriptive equations for the system are obtained by expressing the mass and energy balance of the control volume for each of the main components involved: condenser, evaporator and compressor.

Throttling is expressed by an ideal loss-free orifice, the compressor by a modified polytropic process and a set of motor data.

From the comparisons between simulations of the system and results from corresponding laboratory experiments, the usefulness of the model and its limitations are discussed.

INTRODUCTION - SYSTEM CHARACTERISTIC AND STALL CHARACTERISTIC

It is well-known that in spite of the simple build-up of small household refrigerating systems with hermetic compressor and capillary tube (see fig. 1), the overall dimensioning of the compressor and the system are especially critical in obtaining satisfactory function. The problems involved fall naturally into two groups concerning.

a) a marked dynamic phase during the start-up and pulldown of the system. Herein, for normal systems, lie the heaviest demands on the mechanical strength of the compressor (maximum differential pressure) and on the compressor breakdown torque.

b) a more stable phase in the operating area proper in which system demands are to a greater extent concentrated around a particular mass flow and a reasonable efficiency.

In the following the problems associated with group a) are dealt with.

To be able to compare the condition sequence after start and max. compressor capacity, Danfoss have developed a condition diagram to supplement the traditional diagram in which the condition sequence is delineated as a function of time, (see fig. 2). In the new evaporating pressure (compressor suction pressure) is drawn as the abscissa while the condensing pressure (compressor discharge pressure) is drawn as the ordinate.To be able to immediately compare the operating condition of the refrigerating system using different refrigerants a transformation has been made of the condition diagram, from absolute pressure to the corresponding evaporating temperatures. The curve for maximum compressor capacity (its stall characteristic) can be found from the diagram by drawing in conditions requiring a given torque exactly equal to the motor breakdown torque. It can be seen at once that the compressor will operate in all conditions lying under the stall characteristic while it will stall at all overlying points. Thus, the stall characteristic is exclusively representative of the compressor not the system as a whole.

The same diagram delineates the conditions the system goes through after start-up, i.e. evaporation and condensing pressures from fig. 2. This delineation produces the system characteristic which unlike the stall characteristic is for the whole system inclusive of the compressor.

In fig. 3, by comparing the minimum distance between the two characteristics at point a, it can be seen that at this point (typically approx. 1-2 secs. after start) the highest torque requirement occurs. At this point the compressor might stall if the system demands are too high compared to the breakdown torque of the compressor motor. At point b the maximum differential pressure the compressor is exposed to can be seen. The diagram thus clearly indicates whether the system design including the selection of compressor capacity is reasonable for the properties of the given compressor. It has of course been assumed that the ambient temperature and supply voltage are within the application range of the system.

Danfoss have for some time been using these characteristics in motor dimensioning and system design. Regarding the stall characteristic, both experimentally and theoretically found values have been used whereas the system characteristic has previously only been arrived at by experiment. Although this type of measurement is simple in principle, to change the parameters and take the associated series of measurements of the system is relatively time-consuming and expensive. It can also be difficult to foresee precisely the effect of such parameter variations. With these problems as a background, efforts have been made to develop a simulation programme to describe the start-up and pull down of an evaporation refrigerating system with hermetic compressor and capillary tube.

MODEL CONSIDERATIONS

The traditional refrigerating system with capillary tube consists, principally, of a series of components which all, with the exception of the compressor, can be approximately described as having two-phase flow in horizontal circular tubes with internal and external heat exchange. A number of models can be found in literature associated with this description; the newest of them, e.g. [1] work with fully acceptable precision. Unfortunately, most of these models are designed to describe stable conditions and even though it is in principle simple to add the necessary number of partial derivatives with regard to time, the total number of partial derivatives to be determined during a simulation will grow to the extent that the model will become unacceptable as a design tool because of its size and the amount of calculation time involved.

Neither is a quasi-stable description directly possible with these models because no calculation appears involving the refrigerant distribution in the system. This distribution calculation is decisive during the whole pulldown phase: assuming the whole system is pressure and temperature equalized before start-up, a large part of the refrigerant charge will be absorbed in the compressor oil. When a stable operating condition is reached most of the charge will be transferred to the evaporator. That is to say, most capillary tube systems run with very little subcooling in the condenser, i.e. liquid content in the condenser is relatively low. From this the conclusion is that at any rate during the pulldown phase the mass balance of the individual components must have a significant effect on the condition sequence and that the heat exchange has correspondingly less significance.

Danfoss have sought to utilize these factors in setting up a very simple model (see fig. 4) of the system shown in fig. 1. Main emphasis has been placed on the calculation of mass distribution in the refrigerating system. As can be seen in fig. 4, the central element in this system is a "pot" with the following features:

a) a volume in which refrigerant/oil is present at average stagnation condition. Oil in compressor pot only.

b) an infinitely large internal heat transfer coefficient, i.e. the pot shell has the same temperature as the charge.

c) thermal capacity in the shell, but no transmission resistance.

d) average external heat transfer coefficient.

e) loss-free inlet and outlet tubes.

In the model "pot" is used to describe the compressor housing, condenser and evaporator. The capillary tube is described as an ideal nozzle, the compressor as a polytropic pump driven by a real asynchronous motor. The refrigerated compartment is described as a thermal capacity with a fixed heat transfer coefficient to its surroundings. Refrigerant and oil are described with real conditions.

More detailed descriptions of component features are contained in the following.

SIMULATION MODEL

In setting up the model emphasis was placed on making the calculation simple and there by rapid and clear rather than refined. This means e.g. ignoring the pressure drop in all components except in the restrictor. By this all condition changes can be determined from a known initial condition by pure integration without the use of iteration. Correspondingly, the individual components are coupled via a set of rough assumptions:

a) Quality at condenser outlet (see symbol list for definition of quality):

$$x_2 \geq 0.85 \Rightarrow x_3 = x_2$$

$$x_2 < 0.85 \Rightarrow x_3 = 0$$

b) Quality at evaporator and compressor pot outlet (see fig. 4):

$$x > 1 \quad \Rightarrow x_e = x$$

$$x \leq 1 \quad \Rightarrow x_e = 1$$

i.e. ignoring the fact that with less superheat liquid entrainment can occur.

c) Complete pressure equalizing between evaporator and compressor pot: (equations from App. A)

$$P_7 \equiv P_5 \Rightarrow \dot{P}_7 \equiv \dot{P}_5 \quad \Rightarrow$$

$$\dot{P}_5 = (\dot{Q}_5 + A_5 \cdot \dot{m}_4 - B_5/A_7 \cdot \dot{Q}_7 - B_5/A_7 \cdot B_7 \cdot \dot{m}_1) / (B_5/A_7 \cdot C_7 - C_5)$$

$$\dot{m}_6 = -(\dot{Q}_5 + A_5 \cdot \dot{m}_4 + C_5 \cdot \dot{P}_5)/B_5$$

The calculation is terminated if any of the pots are filled with liquid, i.e. $x \leq 0$.

Integration is carried out as a simple Euler integration of the time derivatives of mass and pressure for each of the pots and of the time derivative of the temperature for the refrigerated compartment.

Pump model

To avoid describing the internal heat exchange in the compressor the suction gas temperature (T_s) at inflow to the cylinder and the discharge gas temperature (T_1) at the compressor discharge connector are related to the pot temperature (T_7) by using two empirical expressions:

$$T_s = T_7 + C_1$$

$$T_1 = T_7 + C_2 \cdot \dot{m}_1$$

Internal pressure drops in the suction and discharge systems of the pump are ignored.

The compression torque is described polytropically:

$$M_{gas} = V_c \cdot n/(2\pi \cdot (n-1)) \cdot P_5 \cdot \left[(P_2/P_5)^{(n-1)/n} - 1\right] \cdot \left[1 - \varepsilon \cdot \left((P_2/P_5)^{1/n} - 1\right)\right]$$

The mechanical torque loss is described by the empirical expression:

$$M_{fr} = M_{fr1} + M_{fr2} \cdot M_{gas}$$

giving the total torque demand of the pump

$$M_p = M_{fr} + M_{gas}$$

The mass flow is calculated by a modified polytropic formula

$$\dot{m}_1 = \nu \cdot V_c/v_s \left[(1-y_s) - (\varepsilon + y_t) \cdot \left((P_2/P_5)^{1/n} - 1\right)\right]$$

The heat flow from the pump to the compressor pot is obtained from

$$\dot{Q}_8 = \dot{m}_1 \cdot (h_8 - h_1) + PW$$

The mass and heat capacity of the pump are included in M_{h7} and C_{h7} for the compressor pot.

Motor model

The motor is described as three vectors with associated values for torque (M_p), power consumption (PW) and speed (ν) in the range from stalling speed to synchronous speed based on static measurements. Motor start is left out of the calculations in that the start phase will normally be over after 0.3 sec. Compared with this the system is normally using 1-2 s to reach point "a" in fig. 3.

Restrictor model

For vapor ($x_3 \geq 1$) the formula for an adiabatic nozzle are used:

$$\dot{m}_4 = \alpha \cdot F \cdot \sqrt{2\kappa/(\kappa-1)\cdot P_2/v_3} \cdot \sqrt{(P_5/P_2)^{2/\kappa}-(P_5/P_2)^{(\kappa+1)/\kappa}}$$

$$\text{for } P_5/P_2 \geq (2/(\kappa+1))^{\kappa/(\kappa-1)} = f_{crit}$$

and $$\dot{m}_4 = \alpha \cdot F \sqrt{\kappa \cdot P_2/v_3 \left(2/(\kappa+1)\right)^{(\kappa+1)/(\kappa-1)}}$$

$$\text{for } P_5/P_2 \leq f_{crit}$$

For liquid ($x_3 = 0$):

$$\dot{m}_4 = \alpha \cdot F \cdot \sqrt{2/v_3 \cdot (P_2-P_5)}$$

For the wet area ($0.85 \leq x_3 < 1$):

$$\dot{m}_4 = \dot{m}_{4V} \cdot x_3 + \dot{m}_{4L} \cdot (1-x_3)$$

Pot model

According to Appendix A the condition change of the pot charge is calculated in the form of the time derivatives for pressure and mass as:

$$\dot{P} = (\dot{Q}+A\cdot\dot{m}_i+B\cdot\dot{m}_e)/C$$

$$\dot{M} = \dot{m}_i-\dot{m}_e$$

After the integration the temperature and other parameters of state are dertermined from the equation of state.

In the above expression for $\dot{P}$ the external heat flow $\dot{Q}$ is calculated from:

a) compressor:

$$\dot{Q}_7 = F_7 \cdot C_{K7} \cdot (T_7-T_a)^{1.25} + \varepsilon \cdot \sigma \cdot F_7 \cdot (T_7{}^4-T_a{}^4)-\dot{Q}_8$$

b) evaporator:

$$\dot{Q}_5 = F_5 \cdot C_{K5} \cdot (T_5-T_R)^{1.25}$$

c) condenser:

$$\dot{Q}_2 = F_2 \cdot C_{K2} \cdot (T_2-T_a)^{1.25}$$

The internal heat flow is not found because of the assumption of infinite internal heat transfer coefficient (see App. A).

Refrigerated compartment model

The refrigerated compartment is described as a simple heat capacity in heat exchange with the evaporator and the ambient:

$$\dot{T}_R = (\dot{Q}_R-\dot{Q}_5)/C_R \text{ , where}$$

$$\dot{Q}_R = KA\cdot(T_5-T_a)$$

COMPARISON BETWEEN SIMULATION AND MEASUREMENT.

The simulation model has been used on a set of data from an experimental system which consisted of a modified standard household chest freezer of 300 l. The changes involved condenser, compressor, restrictor and the charge was changed to R 502. This freezer was also tested in the laboratory and the two sets of results are compared in figs. 5 and 6.

It can be seen that despite the clear differences at individual points the type of condition sequence is the same in both cases. This leads to the conclusion that the most important processes which occur are contained in the simulation while the variations can be attributed to:

a: inadequite parameter matching.
Fig. 5 shows, for example, that evaporator and freezing compartment temperatures drop too rapidly in the simulation. This may be due to an overestimation of the effective values for evaporator heat capacity and heat transfer coefficients.

b: model limitations
It should be noted that the equalizing pressure before start was calculated too low (fig. 6). This means that the oil absorption model is inadequite. The above - mentioned differences in the time sequence could also have something to do with fact that the internal pressure drop in the compressor is not included: thus the compressor yield at high suction pressure becomes too high.

Fig. 2 shows that under the whole start sequence severe subcooling in the condenser occurs. This is not included in the model.

No account is taken of the heat exchange between restrictor and suction line. This can be of significance especially during the last part of the start sequence.

In spite of its inadequacies the model fulfils the most inportant expectations: it can be used to study the processes that occur during start and pull down, it can be used to predict the relative effects of a parameter variation, and it is simple, clear and therefore fast. On the computer used (IBM 370-158) approximatly 3 s. CPU-time was used for pull down phase (3-4000 s.). In order to increase model precision it is the intention to set up better models for oil absorption and compressor and possibly a two-phase description of the capillary tube (which will also make it possible to include the heat exchanger).

CONCLUSION.

With a simple mass and energy balance it is possible to describe the essential processes in the start and pull down phase of a household refrigerating system without a detailed description of the heat exchange and two phase flow. This principle is utilized in the simulation model, which can be used to vary parameters and to study the process sequences involved. The model precision must be improved by building in better component models.

LIST OF SYMBOLS

a	constant in expression for spec. vol. of oil
A	coefficient in expression for time derivative of pressure
b	constant in expression for spec. vol. of oil
B	coefficient in expression for time derivative of pressure
C	coefficient in expression for time derivative of pressure
C_1	constant in motor model
C_2	constant in motor model
C_h	specific heat capacity of house
C_k	constant in expression for heat transfer coefficient
C_{P1}	constant in heat capacity for oil
C_{P2}	constant in heat capacity for oil
C_R	heat capacity of refrigerated compartment
E	energy
F	area
h	specific enthalpy
KA	total heat transfer coefficient for refrigerated compartment
$\dot{m}$	mass flow
M	mass
$\dot{M}$	time derivative of mass
M_h	mass of house
M_{gas}	compression torque
M_{fr}	friction torque
M_{fr1}	constant in expression for friction torque
M_{fr2}	constant in expression for friction torque
M_p	total pump torque
M_X	molecular weight of oil
M_y	molecular weight of refrigerant
n	polytropic coefficient
P	pressure
$\dot{P}$	time derivative of pressure
PW	motor power consumption
$\dot{Q}$	heat flow
T	temperature
$\dot{T}$	time derivative of temperature
u	specific internal energy
V	volume
V_C	cylinder volume
v	specific volume
x	quality of refrigerant: $x = (h-h_L)/h_F$
y_s	loss coefficient in expression for pump mass flow
y_t	loss coefficient in expression for pump mass flow
α	nozzle coefficient
ϵ	dead volume or emission coefficient
κ	isentropie coefficient
ν	pump frequency or speed
σ	Boltzmann constant
τ	time

Indices

1	compressor (pump) outlet fitting
2	condenser
3	condenser outlet
4	nozzle outlet
5	evaporator
7	compressor pot
8	compressor pot outlet = pump inlet
a	ambient
e	exit
F	latent
i	inlet
L	liquid
m	mixture
o	oil
R	refrigerated compartment
s	suction muffer in pump or saturation condition
V	vapor

APPENDIX A: DETERMINATION OF CONDITION CHANGE IN POT

Equation of state for oil/refrigerant mixture (compressor pot)

In the description of the oil/refrigerant mixture a series of simplifying assumptions are made, the authenticity of which are assessed from the results of the total situation:

1. Specific volume.

   $$v_m = (v_o \cdot M_o + v_L \cdot M_L)/(M_o + M_L) \cdot x_k$$

   where x_k is set = 1. According to [2]

   $$x_k = 0.97\text{–}1.0$$

2. Vapour pressure on mixture (to [4]).

   $$P_m = P_s/(M_o/M_L \cdot M_y/M_x + 1)$$

   [3] gives a significantly more complicated correlation containing a series of material constants which can be difficult to find for actual oil and refrigerant types.

3. Enthalpy.

   $$M_m \cdot h_m = h_o \cdot M_o + h_L \cdot M_L + h_V \cdot M_V + h_x \cdot (M_v + M_o),$$

   where the "mixture enthalpy" h_x is set = o, thus, according to [3] the error is up to 20%.

4. Complete miscibility.

The time derivatives is found by using the following equations (ref. to control volume in "POT" on fig. 4):
Equation of continuity:

$$dM = dm_i - dm_e \quad \Rightarrow \quad \dot{M} = \dot{m}_i - \dot{m}_e$$

$$M = M_L + M_V \quad \Rightarrow \quad \dot{M}_V = \dot{m}_i - \dot{m}_e - \dot{M}_L$$

$$M_o = \text{const.}$$

$$V = M_o \cdot v_o + M_V \cdot v_V + M_L \cdot v_L \quad \Rightarrow$$

$$M_L = (M_o \cdot v_o + M \cdot v_V - V)/(v_V - v_L) \equiv g(T,M)$$

$$v = V/M \quad \Rightarrow \quad \dot{v} = -V/M^2 \cdot (\dot{m}_i - \dot{m}_e)$$

Equation of state:

$$P_s = P_s(T)$$

$$v_L = v_s(T) \quad \Rightarrow \quad \dot{v}_L = \frac{dv_s}{dT} \cdot \dot{T} = v_s' \cdot \dot{T}$$

$$h_L = h_s(T) \quad \Rightarrow \quad \dot{h}_L = h_s' \cdot \dot{T}$$

$$v_V = v(P_m,T) \quad \Rightarrow \quad \dot{v}_V = \left(\frac{\partial v}{\partial P}\right)_T \cdot \dot{P} + \left(\frac{\partial v}{\partial T}\right)_P \cdot \dot{T}$$

$$h_V = h(P_m,T) \quad \Rightarrow \quad \dot{h}_V = \left(\frac{\partial h}{\partial P}\right)_T \cdot \dot{P} + \left(\frac{\partial h}{\partial T}\right)_P \cdot \dot{T}$$

$$v_o = a \cdot T + b \quad \Rightarrow \quad \dot{v}_o = a \cdot \dot{T}$$

$$h_o = (C_{P1} + C_{P2} \cdot T) \cdot T \quad \Rightarrow \quad \dot{h}_o = (C_{P1} + C_{P2} \cdot T) \cdot \dot{T}$$

$$P_m = P_s/(M_o/M_L \cdot M_y/M_x + 1)$$

With the inclusion of the above expression:

$$P_m = P_s/\left[M_o M_y/M_x \cdot \left(v(P_m,T) - v_s\right) / \left(M_o \cdot (aT+b) + Mv(P_m,T) - V\right) + 1\right]$$

$P_m \equiv f(T,M)$ is found by iteration from which $M_L \equiv g(T,M)$ can be determined.

$$\dot{P}_m = \left(\frac{\partial f}{\partial M}\right)_T \cdot (\dot{m}_i - \dot{m}_e) + \left(\frac{\partial f}{\partial T}\right)_M \cdot \dot{T}$$

$$\dot{M}_L = \left(\frac{\partial g}{\partial M}\right)_T \cdot (\dot{m}_i - \dot{m}_e) + \left(\frac{\partial g}{\partial T}\right)_M \cdot \dot{T}$$

Assumption 3 $\Rightarrow$

$$\dot{M} \cdot h_m + M_m \cdot \dot{h}_m = M_o \cdot \dot{h}_o + \dot{M}_L \cdot h_L + M_L \cdot \dot{h}_L + \dot{M}_V \cdot h_V + M_V \cdot \dot{h}_V$$

Energy equation:

$$h_i \cdot dm_i - h_e \cdot dm_e - dQ = dE,$$

where $\quad dE = d(u_m \cdot M_m) + M_h \cdot C_h \cdot dT$

Because of the assumption of infinite internal heat transfer coefficient the energy content of the pot house is included in E.

$$u_m = h_m - P_m \cdot v_m \quad \Rightarrow$$

$$\dot{Q} + h_e \cdot \dot{m}_e - h_i \cdot \dot{m}_i + M_h \cdot C_h \cdot \dot{T} - V \cdot \dot{P}_m + h_m \cdot (\dot{m}_i - \dot{m}_e) + M_m \cdot \dot{h}_m = 0$$

By inserting the above expression the energy equation can be converted to

$$\dot{Q} + A\dot{m}_i + B\dot{m}_e + C\dot{P}_m = 0$$

where
$$A = XX - h_i$$
$$B = h_e - XX$$
$$C = -V + M_V \left(\frac{\partial h}{\partial P}\right)_T + X \cdot \left(\frac{\partial T}{\partial f}\right)_M$$

and

$$X = M_h \cdot C_h + M_o (C_{P1} + C_{P2} T) + h_L \left(\frac{\partial g}{\partial T}\right)_M + M_L h_s' - h_V \left(\frac{\partial g}{\partial T}\right)_M + M_V \left(\frac{\partial h}{\partial T}\right)_P$$

$$XX = h_L \left(\frac{\partial g}{\partial M}\right)_T + h_V \left[1 - \left(\frac{\partial g}{\partial M}\right)_T\right] - X \cdot \left(\frac{\partial f}{\partial M}\right)_T \left(\frac{\partial T}{\partial f}\right)_M$$

The time derivative of the pressure, P_m, is thus determined explicitly and can subsequently be used direct in the integration. The temperature T is determined iteratively from $P_m = f(T,M)$. The differentials entering the equation of state are determined as difference quotients.

Equation of state for pure refrigerant in evaporator and condenser pot.

Using a method corresponding to that used for the compressor pot, the time derivative of the pressure in the evaporator and condenser pot is determined:

$$\dot{Q}+A\dot{m}_i+B\dot{m}_e+C\cdot\dot{P} = 0, \text{ where}$$

a. $x \geq 1$ (Vapor):

$$B = V/M\left[\left(\frac{\partial h}{\partial T}\right)_V \cdot \left(\frac{\partial T}{\partial v}\right)_P + \left(\frac{\partial h}{\partial v}\right)_T + M_h/M\cdot C_h\cdot\left(\frac{\partial T}{\partial v}\right)_P\right]$$

$$A = h-h_i-B$$

$$C = M\cdot\left(\frac{\partial h}{\partial T}\right)_V\cdot\left(\frac{\partial T}{\partial P}\right)_V - V + M_h\cdot C_h\cdot\left(\frac{\partial T}{\partial P}\right)_V$$

b. $0 < x < 1$ (Wet range):

$$A = h_L+v_L(h_L-h_V)/(v_V-v_L)-h_i$$

$$B = h_e-(A+h_i)$$

$$C = M_L\cdot h'_L+M_V\cdot h'_V-V + (h_L-h_V)\cdot(M_L\cdot v'_L+M_V v'_V)/(v_V-v_L) + M_h\cdot C_h\cdot T'_s$$

c. $x \leq 0$ (liquid): Calculation concluded.

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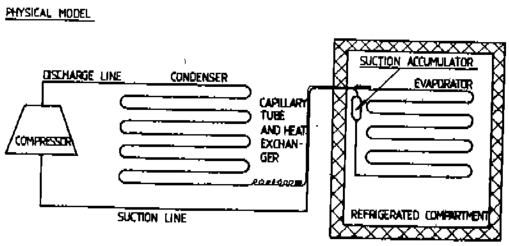

Fig. 1 Physical model of a traditional household refrigerating system

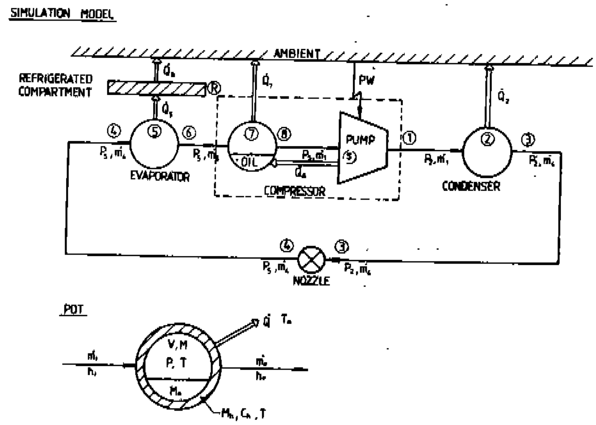

Fig. 4 The simulated system

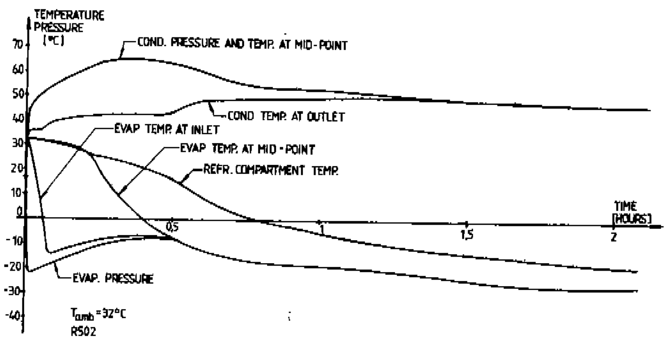

Fig. 2 Traditional time diagram of pull down for an evaporating refrigeration system

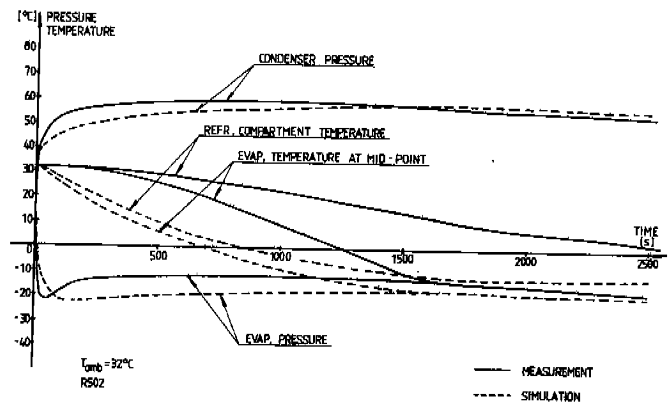

Fig. 5 Time diagram of pull down for test system

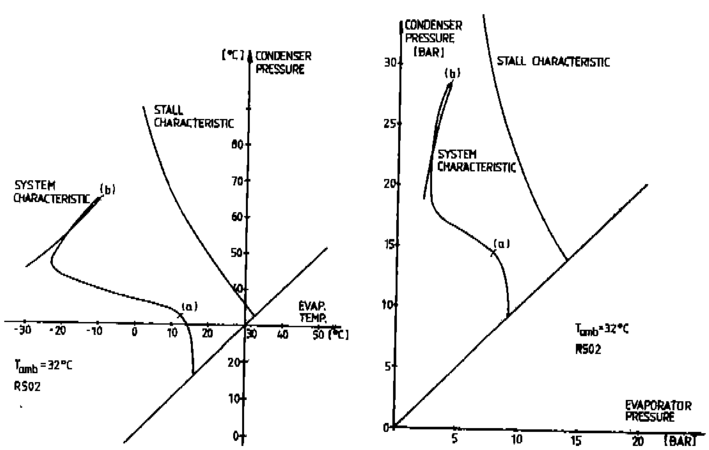

Fig. 3 System and stall characteristics

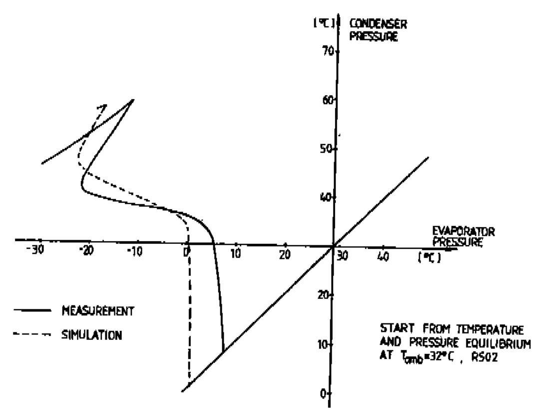

Fig. 6 System characteristics for test system