GÖTEBORGS UNIVERSITET
CHALMERS TEKNISKA HÖGSKOLA
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LABORATION T 10

KYLMASKINEN

(REFRIGERATION CYCLE DEMONSTRATION UNIT.)

INNEHÅLL:

Handledare: ....

Namn: .... Nr: .... Labplats: ....

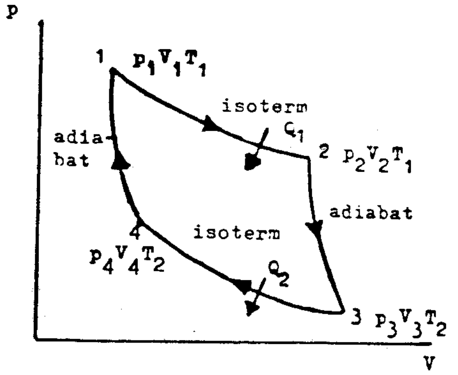
Laborationen utförd den ....
Laborationen inlämnad den ......

Godkänd den .... av ....

1. Principen för kylmaskinen och värmepumpen.

Carnotprocessen består av fyra delprocesser, två isotermer och två adiabater. Består systemet av en gas ser pV-diagrammet ut enligt figur 1. Systemet absorberar värmeenergin Q_1 vid temperaturen T_1 och avger värmeenergin Q_2 vid temperaturen T_2. Nettovärmemängden absorberad är $Q_1 + Q_2$ (Q_2 neg. kvantitet enligt våra teckenregler). Då systemet föres tillbaka till begynnelsetillståndet är $U_1 = U_2$ (U inre energin) och om W är nettoarbetet:

$$Q_2 + Q_1 - W = 0$$



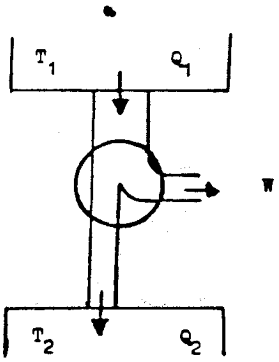
Figur 1.

Kretsprocesser kan allmänt illustreras enligt figur 1. Detta gäller ej enbart för Carnotmaskinen utan för varje värmemaskin. Värme absorberas vid hög temperatur, arbete utföres. Återstoden av värmen avges vid en lägre temperatur. Kretsprocessens termiska verkningsgrad definieras allmänt som:

$$\eta = \frac{\text{Uträttat arbete}}{\text{Tillförd värmeenergi}} =$$

$$= \frac{W}{Q_1} = \frac{Q_1 + Q_2}{Q_1}$$

Q_2 negativ kvantitet. Av ekvationen framgår att η alltid är mindre än 1.



Figur 2.

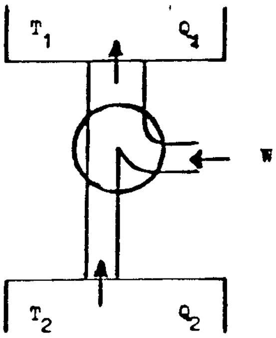
Det utförda arbetet är vad man erhåller och tillförd värme vad man får betala för. Q_2 är på sätt och vis något man får men det har vanligen inget ekonomiskt värde. Genomlöpes processen i omvänd riktning kastas riktningen på pilarna i figuren om (se figur 3). Q_2 borttransporteras från en värmekälla med den lägre temperaturen T_2, arbetet W utföres på systemet, W således negativ kvantitet, och värmemängden $Q_1 = W - Q_2$

avges till värmekällan med den högre temperaturen T_1. Detta kan betraktas som kylmaskinens kretsprocess. Vi definierar kylmaskinens verkningstal (köldfaktorn):

$$\varepsilon = - \frac{\text{Från reservoaren med lägre temperatur borttransporterad värmeenergi (vad vi får).}}{\text{Tillfört arbete (vad vi får betala för).}}$$

$$\varepsilon = - \frac{Q_2}{W} = - \frac{Q_2}{Q_1 + Q_2}\text{, } Q_1 \text{ negativ kvantitet}$$

I praktiken ligger ε mellan 1 och 10.



Figur 3.

En kylmaskin kan under vissa förhållanden betraktas som en värmepump. Värmemängden Q_1 överföres till en hög temperatur. Värmepumpens verkningstal eller värmefaktorn anges som förhållandet mellan till den högre temperaturen överförd värmemängd Q_1 och det för processen erforderliga arbetet W:

$$\phi = \frac{Q_1}{W} = \frac{Q_1}{Q_1 + Q_2}\text{, } \phi > 1\text{, } Q_1 \text{ negativ kvantitet.}$$

Används värmepumpen för uppvärmning av byggnaden kan reservoaren vid den lägre temperaturen vara den omgivande marken eller någon förbiflytande flod. Denna möjlighet till uppvärmning påpekades först av lord Kelvin 1850.

Verkningsgraden för en Carnotmaskin med en idealgas som arbetsmedium kan enkelt beräknas och blir:

$$\boxed{\eta = \frac{T_1 - T_2}{T_1}}$$

För kylmaskinen blir $\varepsilon = \frac{T_2}{T_1 - T_2}$ och för värmepumpen $\phi = \frac{T_1}{T_1 - T_2}$.

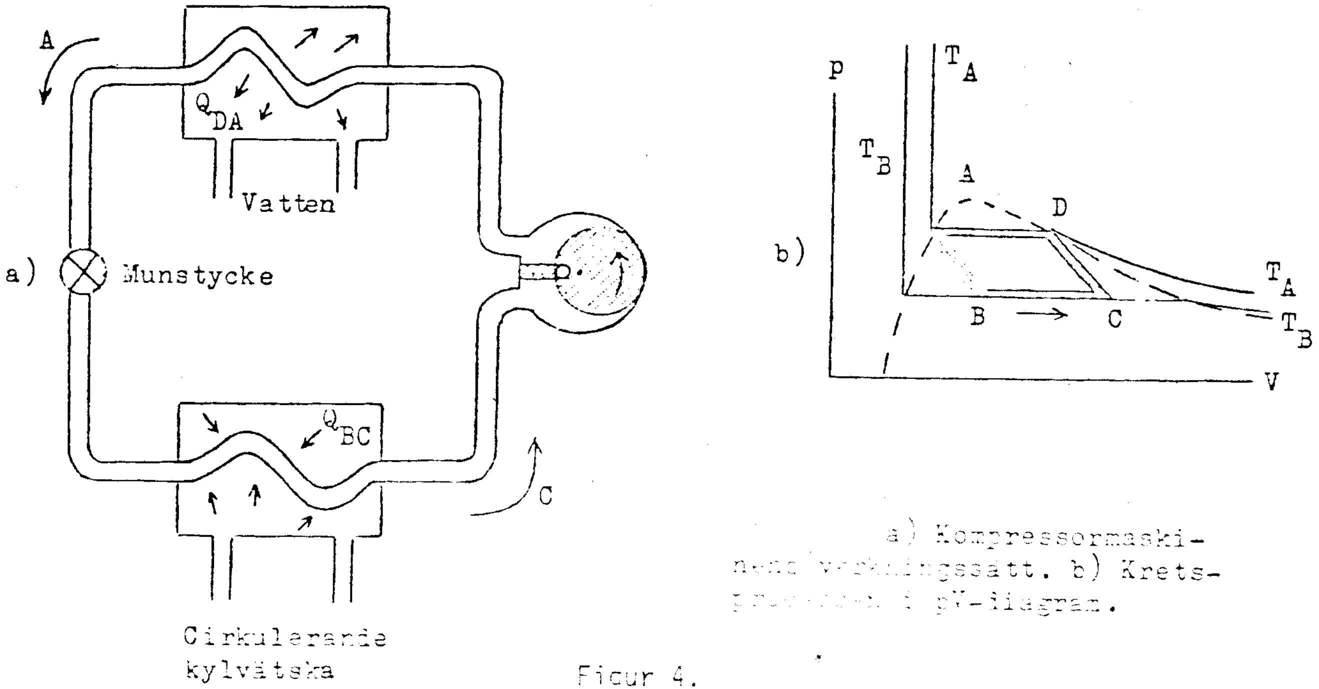
För bägge omloppsriktningarna gäller: $\frac{Q_1}{T_1} + \frac{Q_2}{T_2} = 0$ eller $\Sigma \frac{Q}{T} = 0$.

Det kan visas att Carnotmaskinens verkningsgrad ej beror på arbetsmediets art. Den har ovan angivits för en ideal gas som arbetsmedium och detta uttryck gäller således allmänt.

Den begränsade verkningsgraden, η, är karaktäristisk för kretsprocesser. Teoretiskt kan man omvandla värme fullständigt i arbete genom exempelvis isoterm expansion av idealgas. Men då har man olika tillstånd i början och slutet av processen och den kan ej utnyttjas för att kontinuerligt överföra obegränsade värmemängder till arbete. Det är omöjligt att konstruera en motor som arbetar enligt Carnots princip. I en sådan maskin måste kolven röra sig mycket långsamt under den isoterma tillståndsförändringen och snabbt under den adiabatiska. Denna varierande kolvhastighet skulle betyda en starkt varierande vinkelhastighet på maskinen, vilket är opraktiskt. Vidare är pV-diagrammet långt och smalt. Betyder långt kolvslag och stor friktion i motorns rörliga delar. Carnotprocessens stora betydelse ligger i dess användning i vissa teoretiska resonemang.

2. Kompressormaskinen.

I kompressorkylmaskinen följes i princip en omvänd Carnotprocess. Arbetssubstansen kan vara freon e d. Kretsprocessen framgår av figur 4.



a) Kompressormaskinens verkningssätt. b) Kretsprocessen i pV-diagram.

Figur 4.

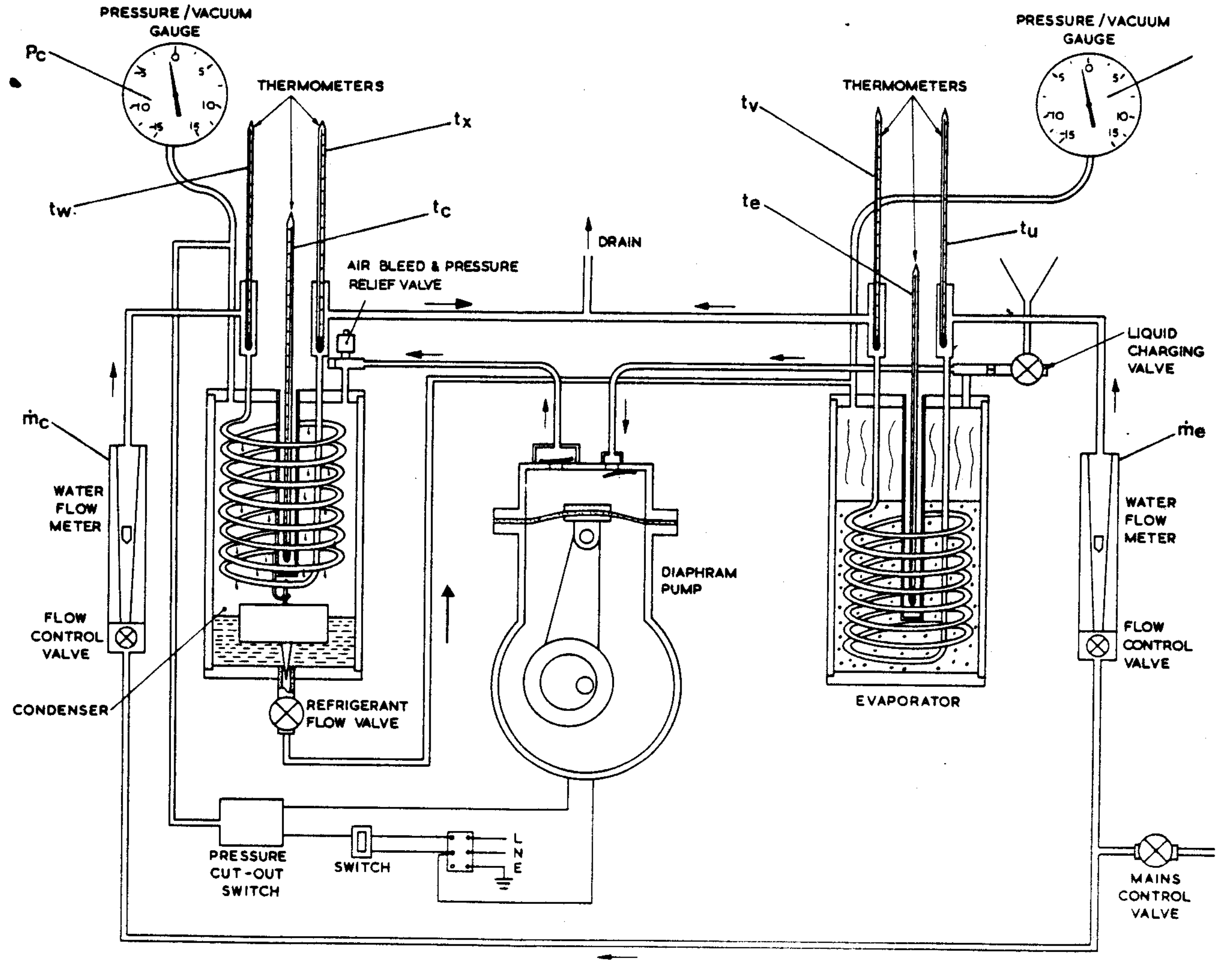
A : Vätska vid den högre temperaturen, T_A, får expandera genom ett munstycke.
AB: Adiabatisk expansion. I praktiken är utströmningen irreversibel.
BC: Isoterm ångbildning vid den lägre temperaturen på grund av kompressorns pumpverkan. Värme tas från kylrummet.

CDA: Adiabatisk kompression i kompressorn. Ångan kondenseras vid förhöjt tryck. Värme avges till "kyl"-vattnet vid den högre temperaturen.

Värme överförs här från lägre till högre temperatur. Detta sker med hjälp av kompressorarbetet och innebär inget brott mot andra huvudsatsen.

3. Beskrivning av kylmaskinen.

REFRIGERATION CYCLE
DEMONSTRATION UNIT
Schematic Diagram



Description

Compressor

This is a diaphragm compressor of the oil free type which has proved suitable for this application for a number of reasons which include —

(a) It is unlikely to be damaged by mal-operation of the equipment (e.g. liquid carry over from the evaporator).

(b) The volumetric efficiency falls fairly rapidly as the pressure ratio is increased which clearly demonstrates the reduction of refrigerator performance under these conditions.

(c) The system does not have to be contaminated with oil for lubrication purposes.

Evaporator

This is constructed from a thick walled glass cylinder with machined brass end plates. The refrigerant can be seen to boil as heat is transferred from a coil of copper tube through which water flows. A combination pressure/vacuum gauge indicates the pressure of the refrigerant, and thermometers in pockets are provided to indicate the refrigerant temperature and the water temperatures at inlet and outlet of the copper coil.

Condenser

This is a similar glass cylinder in which the higher pressure vapour can be seen to condense as heat is transferred to a similar coil of copper tubing through which water flows. Thermometers and a combination pressure/vacuum gauge are provided as in the evaporator.

Expansion Valve

A float operated needle valve is situated in the bottom of the condenser. The valve controls the refrigerant liquid flow rate to match the rate of condensation at all operating conditions.
Due to the relatively small mass of the unit and the way in which the refrigerant flow rate is controlled, the plant reaches stable conditions very quickly and a number of different operating conditions can be evaluated in a normal class period.

GENERAL OPERATING INSTRUCTIONS

1. Ensure that the refrigerator contains sufficient refrigerant liquid to cover evaporator cooling coil under working conditions.
2. Open up water control valves on both evaporator and condenser to high rate of flow at between 10–20°C.
3. Switch on compressor and leave for about ten minutes.
4. Adjust the water flow rates and if necessary the cold water temperatures to obtain the desired evaporator and condenser pressures.

   Note 1. An increase of condenser water or a reduction in its temperature will cause the condenser temperature to fall and hence the pressure, and vice versa.

   Note 2. An increase of evaporator water flow or an increase in its temperature will cause an increase in the evaporation temperature, hence the pressure will rise and vice versa.

   Note 3. Excessively warm evaporator water may cause violent boiling of the refrigerant liquid, some of which may be carried over into the compressor. Whilst moderate quantities of liquid will not harm the compressor, large quantities are not recommended.

Uppgifter.

1. Demonstration of the Vapour Compression Refrigeration Cycle

Cold water is circulated through the evaporator and condenser coils and the unit is switched on. As soon as the pressures in the evaporator and condenser have stabilised, condensation and boiling can be observed. A few minutes later temperature differences in the water outlets will be noticed. The unit may now be examined in detail.

(a) The boiling process which by common experience requires a positive heat transfer is seen to take place at a low pressure and temperature, taking energy from the water passing through the coil and causing a fall in water temperature.

(b) The condensation process is seen to require a negative heat transfer causing the water temperature to rise and occurs at a higher pressure and temperature.

(c) The throttling process in which high pressure liquid from the condenser is reduced to the pressure in the evaporator takes place across the needle valve controlled by the liquid level in the condenser. Liquid is seen to leave the condenser, but a mixture of liquid and vapour is seen to enter the evaporator. In addition, the temperature after the needle valve is found to be much lower than that of the liquid in the condenser and the process of cooling by flash evaporation is demonstrated.

2. Visual Demonstration of the Variation of Refrigeration Effect with Evaporator and Condenser pressure.

Water is passed through the evaporator and condenser coils at a high rate. The unit switched on and allowed to stabilise. The evaporator and condenser pressures and temperatures are noted and the rates of boiling and condensation are observed visually. The evaporator water flow rate is considerably reduced and the unit again allowed to stabilise. The temperatures and pressures are again noted as well as the rate of boiling and condensation. The condenser pressure can be controlled by adjustment of the water flow rate.

The following important observations and deductions can be made —

When the water flow rate is reduced

(1) The evaporator temperature falls (because the average temperature of the coils falls).

(2) The evaporator pressure falls.

(3) The compressor pressure ratio rises.

(4) The rate of boiling (i.e., the cooling effect is reduced).

The test may be repeated with reduced water flow through the condenser but with constant evaporator pressure, when the following observations are made —

(5) The condenser temperature rises (because the average temperature of the coils increases).

(6) The condenser pressure rises.

(7) The compressor pressure ratio rises.

(8) The rate of boiling (i.e., the cooling effect) is reduced.

The foregoing demonstrates that for the largest refrigerating effect from a given unit, the pressure ratios (controlled by evaporator and condenser temperatures) should be as small as possible.
These observed effects are explained by —

(a) The volumetric efficiency of a compressor decreases as the pressure ratio increases.

(b) The density of refrigerant vapour decreases as the evaporator pressure and temperature falls, thus further reducing the mass flow rate.

3. Measurement of Heat Transfer and Estimation of Coefficient of Performance

Symbols

t_c	Refrigerant saturation temperature in condenser
p_c	Refrigerant saturation pressure in condenser
t_e	Refrigerant saturation temperature in evaporator
p_e	Refrigerant saturation pressure in evaporator
$\dot{m}_c$	Water flow rate through condenser
t_w	Water temperature at inlet to condenser
t_x	Water temperature at outlet from condenser
$\dot{m}_e$	Water flow rate through evaporator
t_u	Water temperature at inlet to evaporator
t_v	Water temperature at outlet from evaporator
t_a	Ambient air temperature
A_c	Heat transfer area condenser
A_e	Heat transfer area evaporator

PLANT DATA

Refrigerant

R11	CCl_3F	Quantity — 400-500 m.litre.
R113	$CCl_2F—CClF_2$	Quantity — 400—500 m.litre.

Compressor

Diaphragm type gas compressor. Compton D/180.

Condenser

Glass cylinder 66·6mm bore. 200 mm. long. 4·2 mm. wall thickness.
Copper tubing 6·35 mm. diameter.
Heat transfer area (mean) A_c = 0·032 m^2.
Heat transfer to or from surroundings = 0·8 (t_c - t_a) Watts.*

Evaporator

Similar dimensions to Condenser.
Heat transfer area (mean) A_e = 0·032 m^2.

*By experiment it has been found that the heat exchange rate between the evaporator or condenser and the atmosphere has been established as 0·8 watts per degree K difference between the saturation temperature and ambient temperature.

The unit is set to operate on any evaporator and condenser temperature within its range, and allowed to stabilise. All temperatures, pressures and flow rates are recorded.

Atmospheric temperature	(t_a)

Evaporator–

Refrigerant saturation temperature	(t_e)
Refrigerant saturation pressure	(p_e)
Water flow rate	
or	$(\dot{m}_e)$
Water inlet temperature	(t_u)
Water outlet temperature	(t_v)

Condenser–

Refrigerant saturation temperature	(t_c)
Refrigerant saturation pressure	(p_c)
Water flow rate	
or	$(\dot{m}_c)$
Water inlet temperature	(t_w)
Water outlet temperature	(t_x)

Beräkningsexempel.

Typical results with R11 are shown below–

Atmospheric temperature	(t_a) 21·2°C

Evaporator–

Refrigerant saturation temperature	(t_e) 5·6°C
Refrigerant saturation pressure	(p_e) –52 kN/m^2
Water flow rate	36 kg/h
or	($\dot{m}_e$) 0·01 kg/s
Water inlet temperature	(t_u) 16·8°C
Water outlet temperature	(t_v) 12°C

Condenser–

Refrigerant saturation temperature	(t_c) 27·4°C
Refrigerant saturation pressure	(p_c) 17 kN/m^2
Water flow rate	25·2 kg/h
or	($\dot{m}_c$) 0·007 kg/s
Water inlet temperature	(t_w) 16·9°C
Water outlet temperature	(t_x) 25·6°C

There are three significant energy transfer rates in the refrigeration cycle —

(1) The (positive) heat transfer rate to the refrigerant as it is evaporated $\dot{Q}_e$.

(2) The (negative) heat transfer rate as the refrigerant is cooled and liquified in the condenser $\dot{Q}_c$

(3) The (negative) work transfer rate $\dot{W}$ to the refrigerant as it passes through the compressor.

According to the First Law of Thermodynamics, in a cycle, the nett heat transfer is equal to the nett work transfer, thus

$$\dot{W} = \dot{Q}_e + \dot{Q}_c$$

Calculations

$\dot{Q}_e$ = Heat transfer rate from (Water + Atmosphere)

$$\dot{Q}_e = \dot{m}_e C_p (t_u - t_v) + 0{\cdot}8\,(t_a - t_e)$$

$$\dot{Q}_e = 0{\cdot}01 \times 4180\,(16{\cdot}8 - 12) + 0{\cdot}8\,(21{\cdot}2 - 5{\cdot}6)\ \text{Watts}$$

$$\dot{Q}_e = 200{\cdot}6 + 15{\cdot}6$$

$$\dot{Q}_e = 216{\cdot}2\ \text{W}\ (+\text{ve})$$

$\dot{Q}_c$ = Heat transfer rate to (Water + Atmosphere)

$$\dot{Q}_c = \dot{m}_c C_p (t_x - t_w) + 0{\cdot}8\,(t_c - t_a)$$

$$\dot{Q}_c = 0{\cdot}007 \times 4180\,(25{\cdot}6 - 16{\cdot}9) + 0{\cdot}8\,(27{\cdot}4 - 21{\cdot}2)\ \text{Watts}$$

$$\dot{Q}_c = 254{\cdot}6 + 5$$

$$\dot{Q}_c = 259{\cdot}6\ \text{W}\ (-\text{ve})$$

From the first law–

$$\dot{W} = \sum Q = 216{\cdot}2 - 259{\cdot}6$$

$$\dot{W} = \underline{\underline{43{\cdot}4\ W}}\ (-\text{ve})$$

It should be appreciated that this is the difference between the work input by the compressor motor and the heat losses at the compressor.

Neglecting this heat loss, estimated coefficient of performances.

$$= 216{\cdot}2/43{\cdot}4$$

$$\text{C of P} = 4{\cdot}98$$

4. Measurement of heat transfer coefficient during evaporation and condensation

The overall heat transfer coefficient (U) is the heat transfer rate per unit area of heating surface, when a temperature difference of one degree exists between the hot and cold fluids.

In the case of a heat exchanger, the temperature difference is the log mean temperature difference which in the case of the condenser is —

$$\frac{(t_c - t_w) - (t_c - t_x)}{\ln \dfrac{t_c - t_w}{t_c - t_x}} = \Delta t_c$$

and in the case of the evaporator is —

$$\frac{(t_u - t_e) - (t_v - t_e)}{\ln \dfrac{t_u - t_e}{t_v - t_e}} = \Delta t_e$$

The heat transfer rate to the water at the condenser

$$\dot{Q}_c = \dot{m}_c C_p (t_w - t_x)$$

$$\text{and} \quad U_c = \frac{\dot{m}_c C_p (t_w - t_x)}{A_c \Delta t_c}$$

$$\text{Similarly} \quad \dot{Q}_e = \dot{m}_e C_p (t_u - t_v)$$

$$\text{and} \quad U = \frac{\dot{m}_e C_p (t_u - t_v)}{A_e \Delta t_e}$$

Calculation of heat transfer coefficient at condenser

Beräkningsexempel.

Using previous results –

$$\Delta t_c = \frac{(27{\cdot}4 - 16{\cdot}9) - (27{\cdot}4 - 25{\cdot}6)}{\ln \dfrac{27{\cdot}4 - 16{\cdot}9}{27{\cdot}4 - 16{\cdot}9}}$$

$$= \frac{10{\cdot}5 - 1{\cdot}8}{\ln \dfrac{10{\cdot}5}{1{\cdot}8}} = 4{\cdot}93 \text{ K}$$

$$\dot{m}_c C_p (t_w - t_x) = 259{\cdot}6 \text{ W}$$

Area of heat transfer surface $A = 0{\cdot}032 \text{ m}^2$

$$U_c = \frac{259{\cdot}6}{0{\cdot}032 \times 4{\cdot}93} = 1644 \frac{\text{W}}{\text{m}^2 \text{ K}}$$

$$= 1{\cdot}64 \frac{\text{kW}}{\text{m}^2 \text{ K}}$$

Calculation of heat transfer coefficient at evaporator

$$\Delta t_e = \frac{(25{\cdot}6 - 5{\cdot}6) - (16{\cdot}9 - 5{\cdot}6)}{\ln \dfrac{25{\cdot}6 - 5{\cdot}6}{16{\cdot}9 - 5{\cdot}6}}$$

$$= \frac{20 - 11{\cdot}3}{\ln \dfrac{20}{11{\cdot}3}}$$

$$= 15{\cdot}2 \text{ K}$$

$$m_e C_p (t_u - t_v) = 216{\cdot}2 \text{ W}$$

$$A = 0{\cdot}032 \text{ m}^2$$

$$U_e = \frac{216{\cdot}2}{15{\cdot}2 \times 0{\cdot}032} = 444 \frac{\text{W}}{\text{m}^2 \text{ K}}$$

$$= 0{\cdot}444 \text{ kW/m}^2 \text{K}$$

5. **Production of Performance Curves**

It is readily appreciated that the Refrigeration Effect of a given compressor and its associated equipment varies as the temperatures of the evaporator and condenser are changed. This has already been demonstrated qualitatively and it is possible with this unit to produce performance curves which demonstrate this characteristic in a quantitative manner.

It is convenient to keep the evaporator saturation temperature constant by adjustment of the evaporator water flow rate and inlet temperature and to run a series of tests at a variety of condenser pressures (controlled by variation of condenser water flow rate).

The refrigeration effect Q_e is calculated as before and typical results are given below

Evaporation saturation temperature °C	10	10	10	10·3
Condenser saturation temperature °C	36	32	29·3	26
Refrigeration Effect W	178	214	255	274

Other evaporator saturation temperatures are selected and the tests repeated. The results may be presented in a number of ways but the method shown is recommended. See Fig. 2.
The characteristics of any vapour compression refrigerator are clearly shown in an exaggerated form for the reason mentioned.

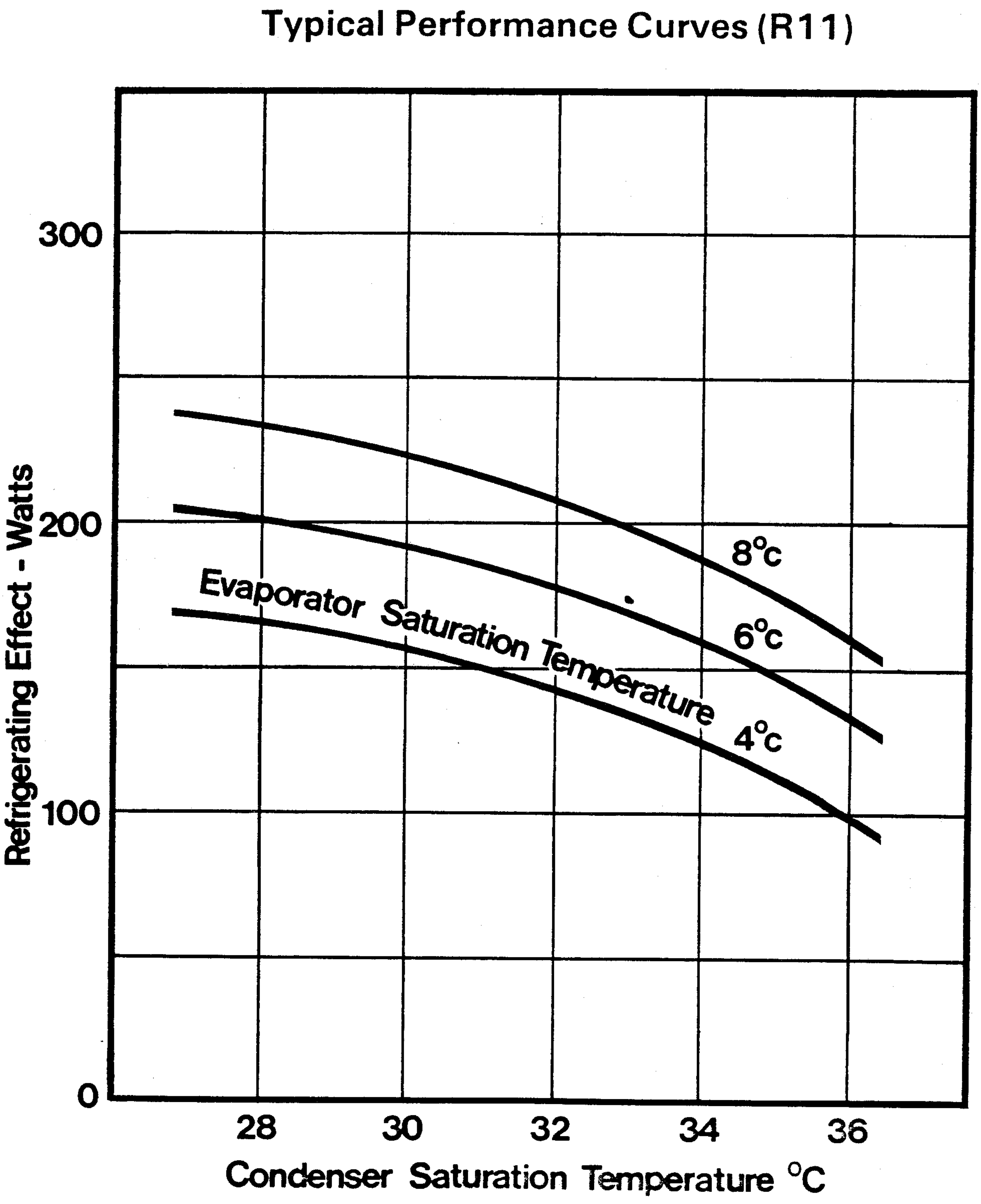
(a) At any given evaporator temperature the refrigerating effect decreases as the condenser temperature increases and would eventually fall to zero.

This is due to the increasing pressure ratio across the compressor which reduces its volumetric efficiency and hence the refrigerant flow rate.

(b) At any given condenser pressure the refrigerating effect decreases as the evaporator temperature is reduced.

This is accounted for by the reason given in (a) and, in addition, as the evaporator temperature falls, the density of the vapour at compressor suction falls, and this further reduces the refrigerant mass flow rate, e.g., the density of vapour at 0°C is 63% of that at 12°C.

FIG. 2


Typical Performance Curves (R11)
Refrigerating Effect - Watts
300
200
100
0
28
30
32
34
36
Condenser Saturation Temperature °C
Evaporator Saturation Temperature
8°c
6°c
4°c