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SECOP

CONDENSERS FOR REFRIGERATION APPLIANCES

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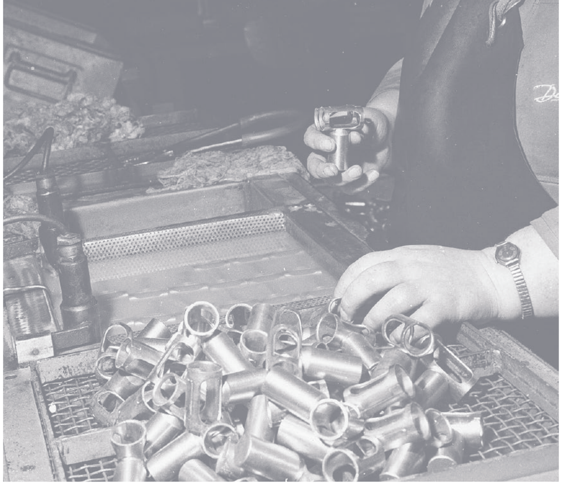
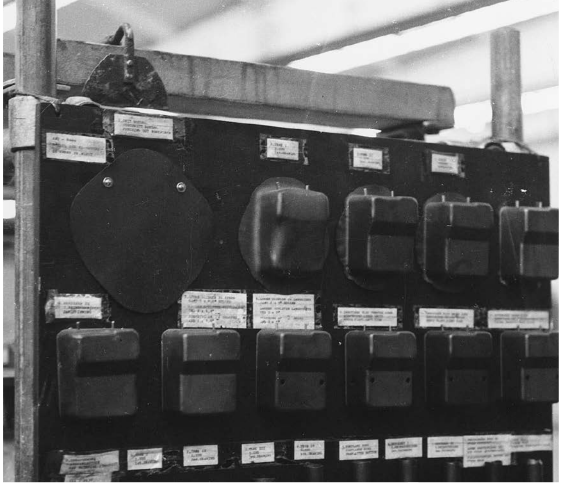
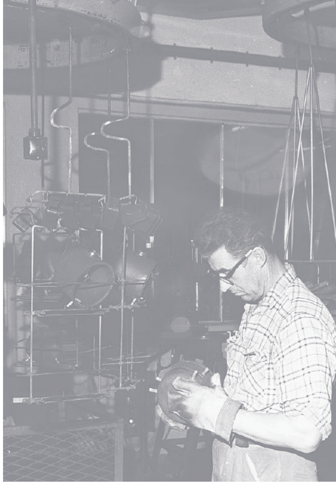
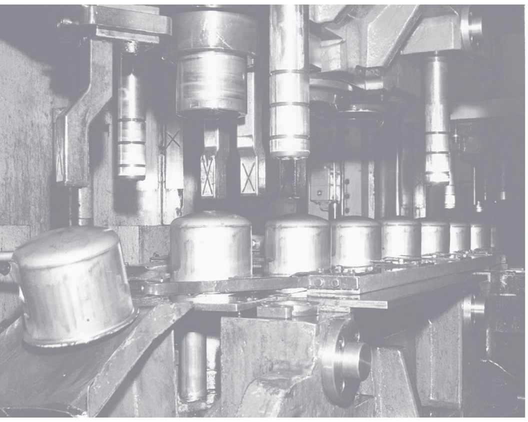
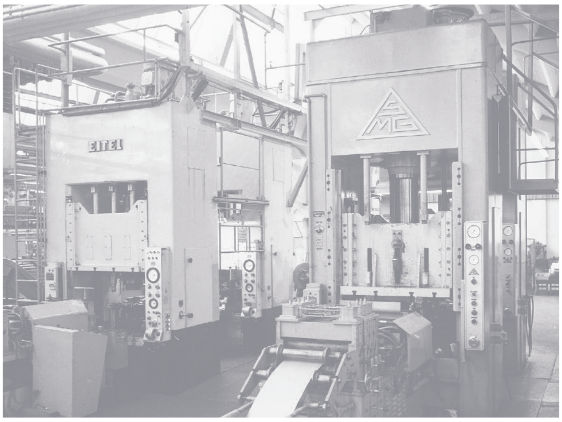
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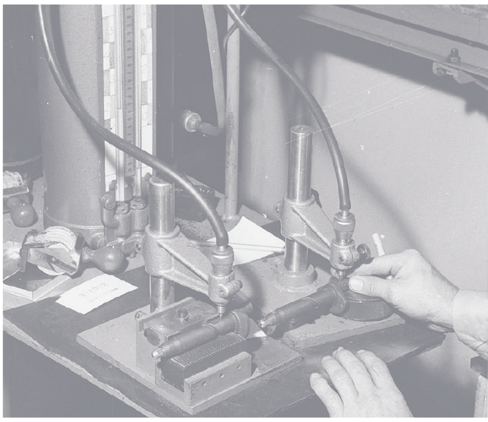

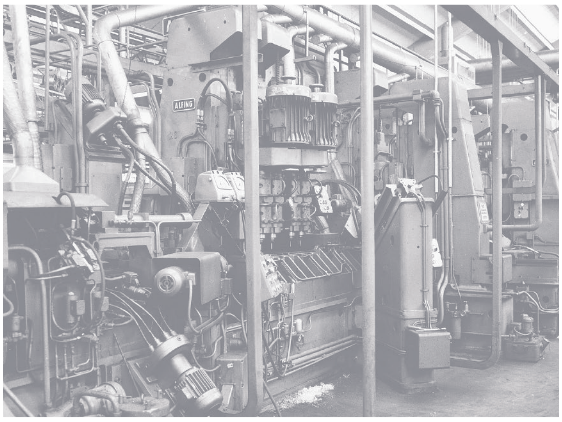
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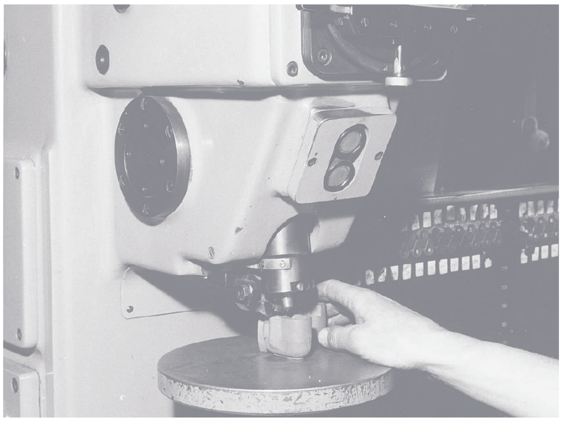

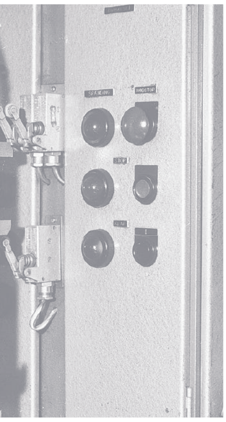
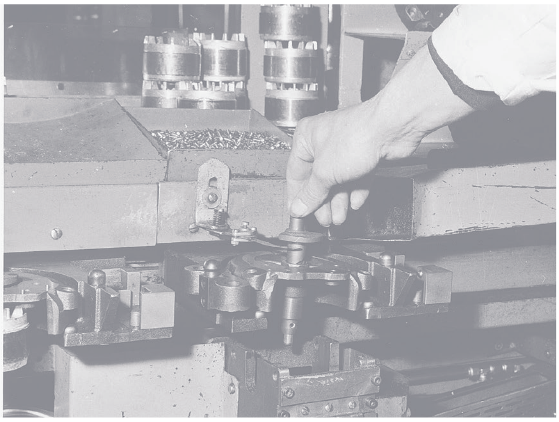
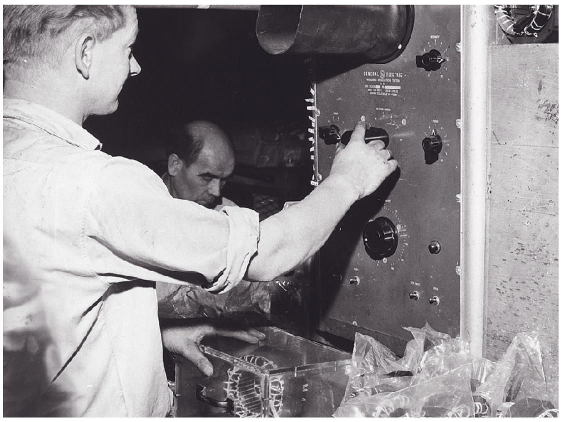


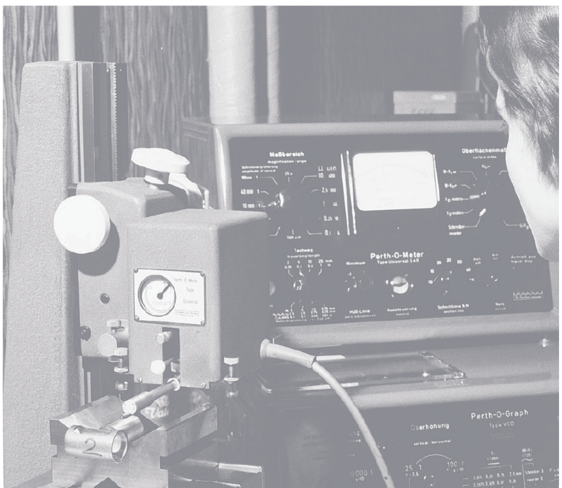
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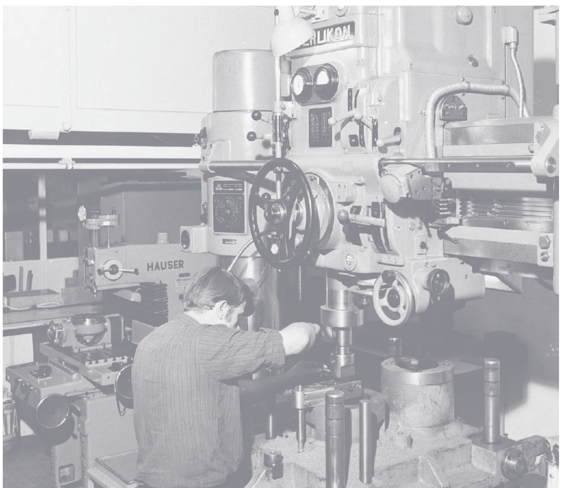
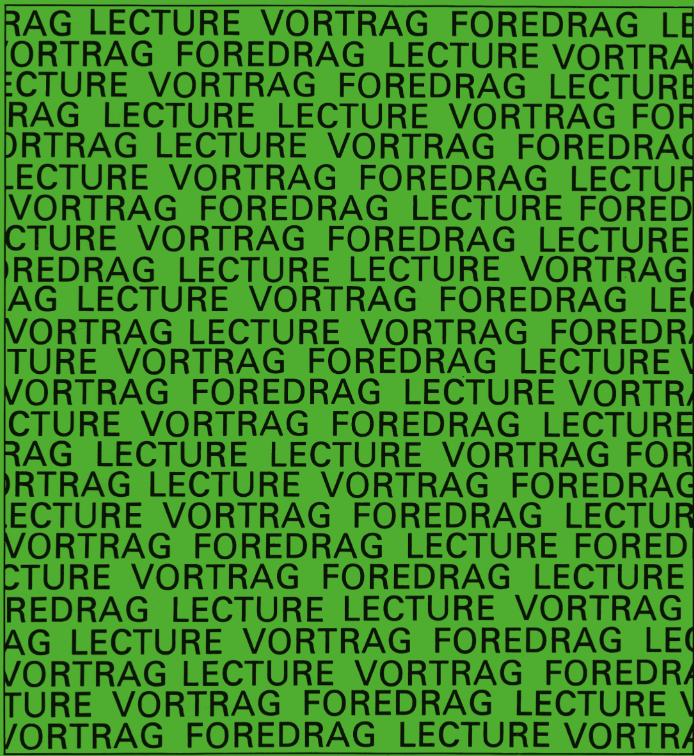

HAUSER


DANFOSS

Condensers for refrigeration appliances

Product Line: Compressors for refrigerators and freezers



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Condensers for Refrigeration Appliances

Contents

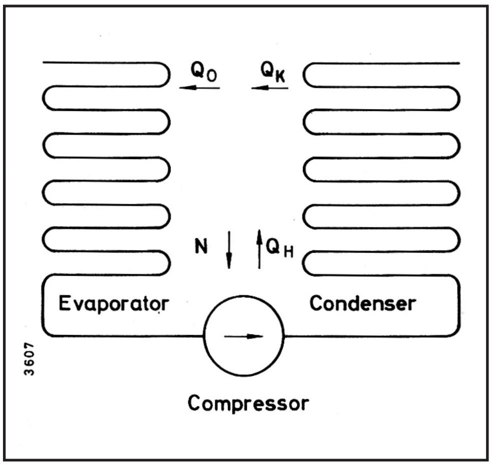
01. General

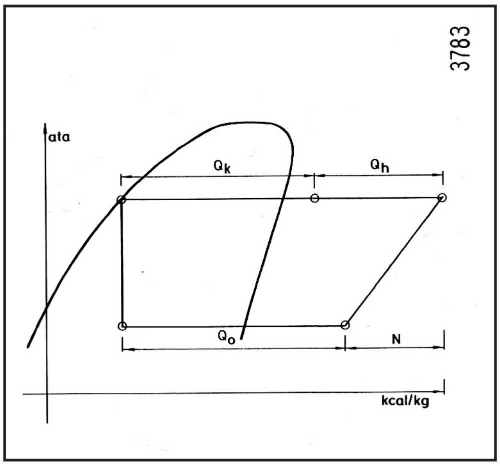
Heat transport in refrigeration system

Fig. 1

Process in Mollier diagram

Fig. 2





The object of the condenser in the refrigeration system is to remove the heat supplied to the refrigerant in the evaporator and during the compression in the compressor, as well as to condense the refrigerant vapours into liquid.

The condenser capacity (Q_k) can be formulated as follows:

$$Q_k = Q_o + N - Q_h \text{ (kcal/h)}$$

Q_0 = Evaporator capacity + increase of enthalpy up to the cylinder
N = Compression work in kcal/h
Q_h = Heat emission from compressor pot

This section deals with the selection of air-cooled condensers.

---

02. Selection

The condenser capacity Qk can be calculated according to the formula:

$$Q_k = K \times A \times \Delta t \text{ (kcal/h)}$$

K = Transmission coefficient (kcal/h x m^2 x °C)
A = Condenser area (m^2)
t = Temperature differential between condensing temperature and air (°C)

For calculation, see section 23.
The condensing temperature should be as low as possible, - in regard to both compressor life and energy consumption. This can be achieved by using well dimensioned condenser surface and good ventilation under all operating conditions. The following dimensioning rules refer to condensing temperatures, measured at the built-in refrigeration system at continuous operation.

At 32°C ambient temperature the condensing temperature should not exceed 50°C.

When applied in statically cooled systems under tropical conditions the condensing temperature normally cannot be kept below 50°C, but at any rate it must not exceed 60°C.

If the temperature limits cannot be maintained, oil or fan cooling of the compressor must be used. If these limits are exceeded, a considerable reduction of the compressor life will have to be reckoned with since a high condensing pressure increases the compressed gas temperature, valve temperature, winding temperature, etc., of the compressor. A high compressed gas temperature and hence a high valve temperature promote the chemical decomposition of refrigerant and oil and thus constitute an important factor in regard to the occurrence of such faults as valve coking and other chemical decomposition.

Finally, the compressor motor has not been designed for operation at condensing pressures which are considerably higher than the temperatures mentioned above. Exceeding these limits can, therefore, also result in the compressor stalling and cut-out of the motor protector.
Building-in and contamination may result in impaired operating conditions, and a safety

margin should, therefore, always be maintained to ensure against such unintentional load conditions.

03.
Condenser types

Air-cooled condensers can be devided into statically cooled and fan-cooled types.

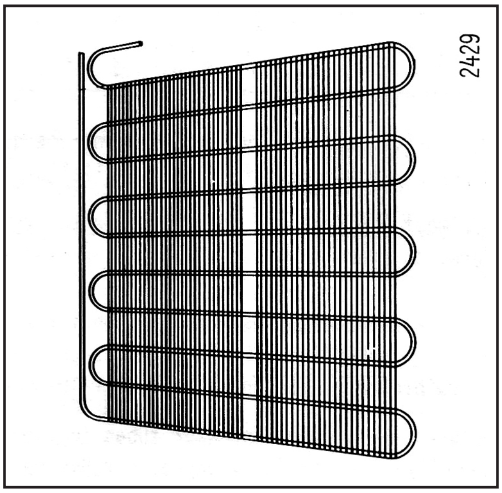
The statically cooled condenser is designed for use in small refrigeration appliances with sufficient space for the necessary condenser area. In large units it is necessary to use fan-cooled condensers in order to keep the condenser area to a reasonable size.

Statically cooled condensers can be divided into skin type condensers, used primarily for chest freezers, and rear condensers which can be obtained in versions with wire on tube or tube on plate.
Fan-cooled condensers can be divided into the finned condenser, which is most common, and the tube-on-plate condenser.
A description of the different types is given in the following sections.

04.
Rear condenser

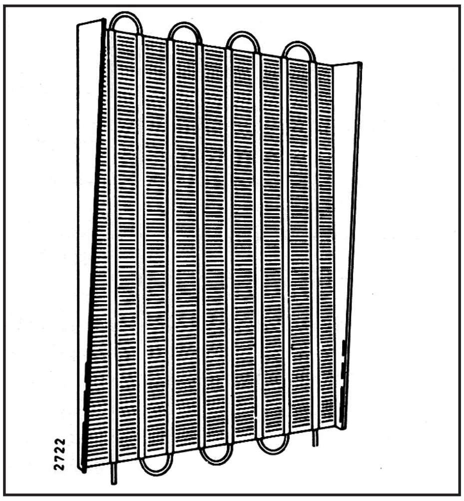
Wire-on-tube condenser

Fig. 3



Tube-on-finned-plate condenser

Fig. 4



The rear condenser, which is primarily fitted on the back of the refrigeration appliance, is especially used for refrigerators and upright freezers, but in some cases also for small chest freezers.
The rear condenser is available in two versions, viz. with wire on tube or with tube on plate.
The wire-on-tube condenser shown in fig. 3 has thin iron wires spot-welded to both sides of a tube coil.
Fig. 4 shows a condenser with tube on finned plate. It consists of an approx. 0.5 mm iron sheet with fins spot-welded to a tube coil.

05.
Choice of rear condenser

Tube-on-finned-plate condensers have a degree of pipe utilization which is up to 40-50% better than in the case of the wire-on-tube design, which means that more tube is used for each square metre of condenser surface with the wire-on-tube design than with the tube-on-plate design.
In actual practice, refrigeration systems with tube-on-plate condensers require a smaller charge, resulting in a lower equalizing pressure.
There is no clear-cut difference between the pressure equalizing time of the two condenser types.
The tube-on-plate condenser with a vertical tube layout is arranged for the longest pressure equalizing time, but the wire-on-tube condenser contains the largest volume and hence the largest charge of Rl2 per square metre of surface and tends to give a longer equalizing time.
For domestic refrigeration appliances, 5mm tube is normally used and this does not give pressure equlizing problems.
Refrigeration systems with tube-on-plate condensers often have a higher noise level than systems with wire-on-tube condensers.

This is due to the fact that the low-frequency noise which is transmitted from the compressor is amplified in a tube-on-plate condenser.
In addition, the noise from the refrigerant will often be strongest when the tube-on-plate condenser is used since the vertical tube layout of this condenser type may cause or amplify pulsations in the refrigeration system.

06.
Transmission coefficient

Fig. 5

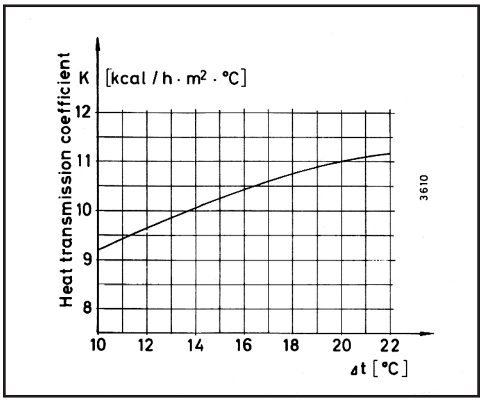
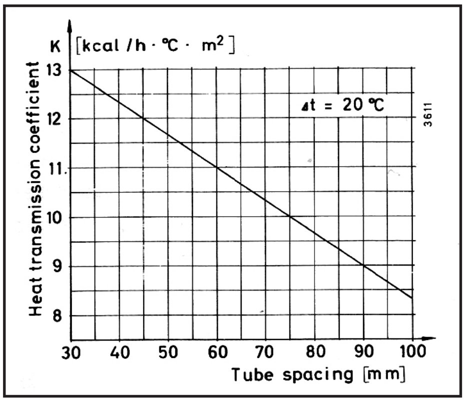


Fig. 6



The transmission coefficient or U-value expresses the quantity of heat in kcal/h which can be removed per m at a temperature differential of 1°C.

The U-value is almost the same for the two condenser types. The U-values specified below must be considered as average values which can be used for estimating both the condenser types.
Fig. 5 shows how the U-value varies depending on the temperature differential at a tube spacing of 60mm. The temperature differential is generally 15-16°C for rear condensers so that it is possible to reckon on a U-value of 10-11 kcal/h x m^2 x°C.

The distance between the single condenser tubes influences the U-value of the condenser, which can be seen from fig. 6. For example, a tube spacing of 45mm gives a U-value which is almost 50% higher than at 100mm. However, the maximum U-value does not express the most economical condenser design. Several examinations have shown that the lowest relative material costs per output unit are obtained with a tube spacing of 60mm.

On wire condensers the most economical distance is 6mm between wires. Normally, a wire diameter of 1.5mm is used.
Literature: H.L. von Cube, G. Tofahrn: "Formes economiques de condenseurs pour armoires frigorifiques menagers".

07.
Tube size

Either 5 or 6 mm steel tubes are used for rear condensers. The tube size has a great influence on the operating conditions of the refrigeration system.

The refrigerant charge required will be reduced by reducing the tube size from 6 mm to 5 mm. However, the refrigerant charge is not reduced to the same degree as the internal condenser volume. Especially on somewhat larger refrigeration systems therefore, a change to 5 mm tubes can result in extremely high maximum condensing pressures due to lack of space for the refrigerant charge in the condenser during evacuation of the system.

The pressure equalizing time of the refrigeration system is increased if 6 mm tubes are used, among other reasons because a larger R12 charge is required.

In the case of large 6 mm condensers, with some thermostat positions it can be difficult to obtain sufficient time for pressure equalization. The result can be starting trouble which can be overcome by changing to 5 mm tubes.
The reduced charge in 5 mm condensers will result in a lower equalizing pressure and hence improved starting conditions for the smaller compressors during the initial start-up. Finally, it must be mentioned that the larger pressure drop in 5 mm condensers and discharge tubes results in the build-up of a higher pressure in connection with the start

and pull-up of the compressor, and hence a reduction of the starting capacity of the compressor, especially when starting from high equalizing pressures.

The discharge connector sizes of Danfoss compressors can be used as a guide in selecting tube size for the rear condenser.

08.
Condenser area

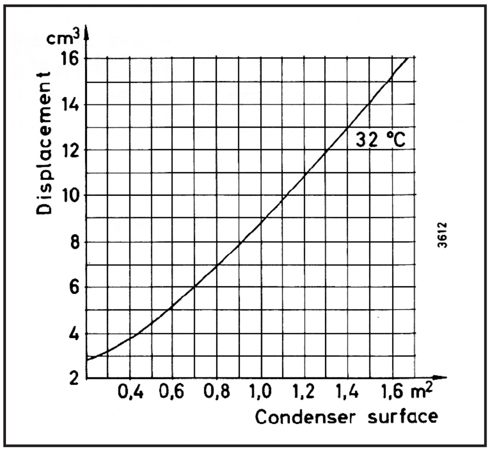


Fig. 7

Fig. 7 can be used as a starting point in selecting a statically cooled rear condenser. The graph which indicates the total condenser area as a function of the compressor displacement, only applies to condensers for domestic refrigeration appliances in the M/LBP range. The graph should be considered with certain reservations, partly because the U-value of the condenser changes with the air circulation conditions, and partly because the U-value varies somewhat with the type and manufacture, etc. The diagram applies to 2-pole compressors(50Hz).

For finned condensers see section 18.

09.
Inatallation

Built-in refrigeration condenser

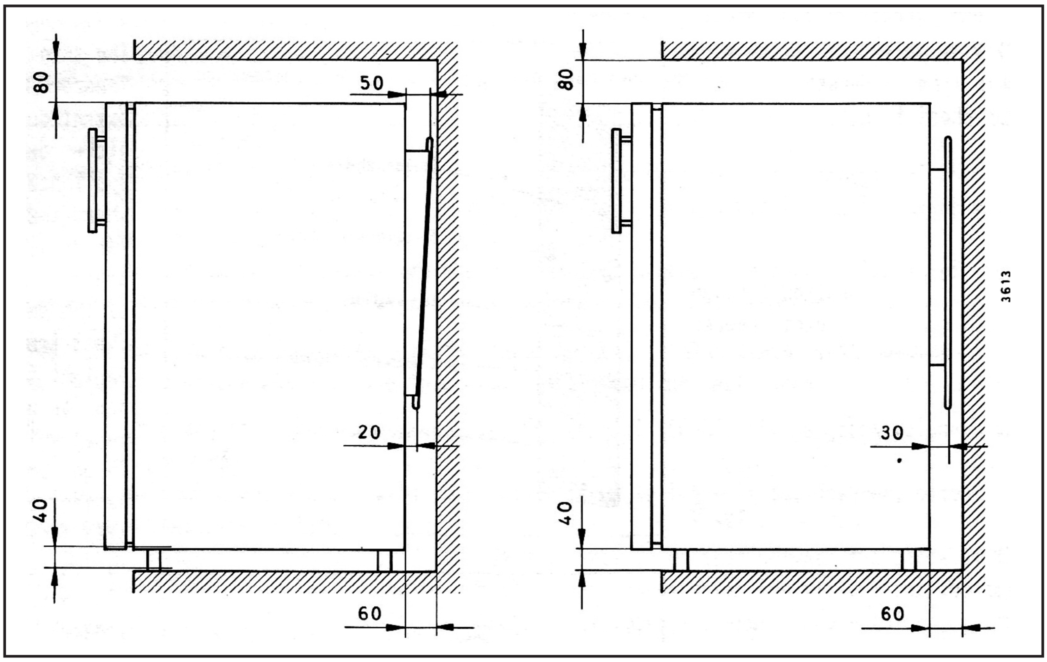


Fig. 8

Fig. 8 shows air gaps around a built-in refrigerator which ensure a sufficient air supply to compressor and condenser. The distance to the rear wall of the refrigerator should be approx. 60 mm, and the condenser should be mounted in the centre of the air gap.

It should be noted that the condenser with tube on finned plate is placed obliquely in the air gap so that the U-value is increased.
The gap depths stated are for guidance only since other depths can be used where necessitated by the dimensions of the refrigeration appliance as well as by the size of the compressor and its position in the motor compartment.

10.
Skin type condenser

When a skin type condenser is used, the surface (the shell) of the refrigerator or freezer is used partially or fully as condenser surface since the condenser coil is distributed and fixed outside or inside the shell which thus acts as heat emission surface.

The skin type condenser offers the following advantages:

The surface temperature of the shell is increased and the risk of condensation thus reduced.

Reduced space requirements for the refrigeration appliance.

The condenser cannot collect dirt.

The most serious drawbacks are:

Increased ingress of heat through the insulation of the appliance, especially under the most rigorous Toad conditions (e.g. freezing). In actual practice this means that a heavier insulation must be used for appliances with a skin type condenser.

Inaccessibility for servicing.

For example, leaks from a skin type condenser in a freezer with foamed insulation can be very unpleasant.

The skin type condenser is available in three fundamentally different designs:

Skin type condenser for chest freezers, with the condenser coil distributed on the four inner sides of the shell.
Skin type condenser for refrigerators and upright freezers, with the condenser coil distributed on the outside of the shell back.
Anti-dew coil which serves to increase the surface temperature on the frame of the upright freezer so that dew formation can be avoided. The anti-dew coil is run along the frame inside the shell and is often used with a rear condenser

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11.
Skin type condenser for chest freezers

Section throught chest freezer

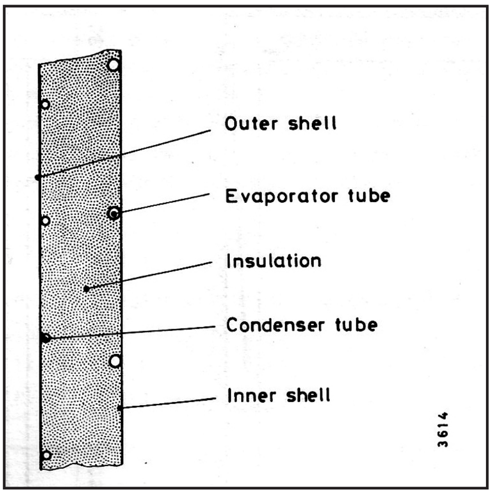


Fig. 9

Skin type condenser on chest freezer

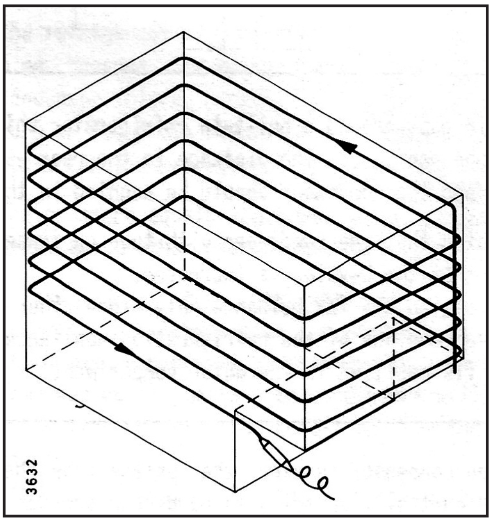


Fig. 10

The skin type condenser is used on chest freezers in particular. This is to some extent due to the fact that, to a far higher degree than with other types of appliance, chest freezers are placed where the risk of dewfall on the shell is far greater than normal.
Another important fact is that all sides of the shell can be used for condensing on chest freezers, while refrigerators and upright freezers, which are normally designed for building in, can only use the rear panel which is often insufficient for the large compressor capacities.
As mentioned above, the tube coil for the skin type condenser of the chest freezer is fitted inside the shell.
Fig. 9 shows a section through a chest freezer. It can be seen that the evaporator tubes and the condenser tubes are fitted on the inner and outher shells respectively, embedded in insulation material.

In order to achieve a good heat emission the tubes must be in good contact with the shell. This is done by retaining the tubes by means of clamps or bars. The principles of these types of fixing are dealt with under evaporators (CN.82.00.02) which also shows a number of mounting examples. The contact can be improved further by means of thermal mastic.
If the insulation is foamed directly into

the chest freezer, care must be taken that the insulation material does not come between the tube coils and the shell which would cause the heat transmission to be impaired. This can be prevented by covering the condenser tubes with plastic foil or tape before the foaming.
The tube layout of the shell condenser should be chosen so that "pockets" are avoided since efforts must always be made to reduce the pressure-equalizing time of the system as much as possible. This can be done in the form of a horizontal tube layout with the inlet at the top.
The most common procedure is to shape the condenser coil as a continuously declining spiral. See fig. 10.

Skin type condenser with oil-cooling section

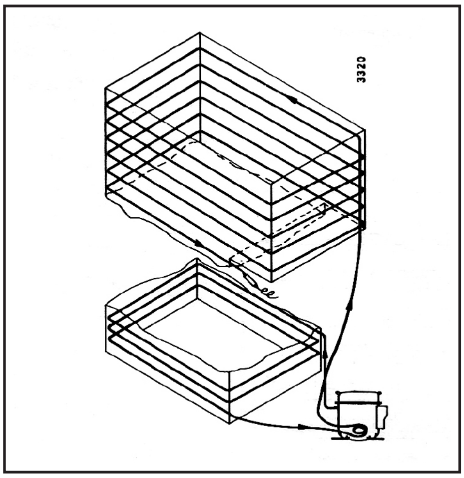


Fig 11

If part of the skin type condenser is used as an oil-cooling condenser, a tube layout like the one shown in fig. 11 may be used. If optimum oil cooling of the compressor is aimed at, approximately one half of the total tube length should be used as oil-cooling condenser.
If part of a given skin type condenser is used as an oil-cooling condenser, it will be necessary to reckon with some increase of the condensing pressure and hence of heat incidence on the evaporator.

Since this is an undesired side effect of oil cooling, the total tube length should be increased by 20-30% to keep down the condensing pressure. Alternatively, the condenser surface can be increased by using an oil cooling condenser. For oil cooling, see also section 15.

---

12. Dimensions of skin condenser

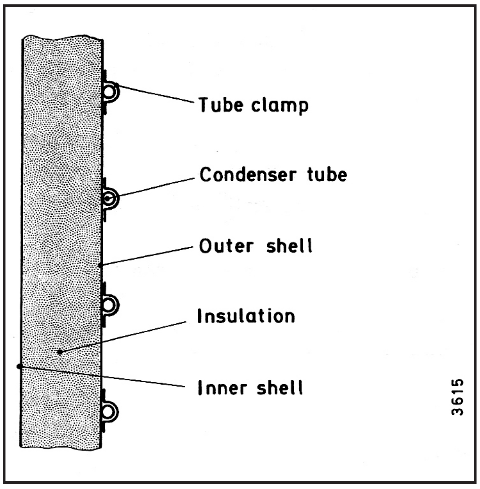
The contact between tube and plate is decisive for the heat emission capacity of the shell (the U-value of the shell), and it is, therefore, impossible to specify exact values for the length of the condenser coil.
On page 20 some empirical values are shown. The diagram shows the average length of the condenser coil as the function of the gross volume of the chest freezer.

The tube spacing resulting from a given condenser coil is also shown. This spacing is calculated on the basis of the total tube length and the shell area.

Normally, for domestic chest freezers, i.e. freezers of 150-600 litres gross volume 6 mm steel tube is used.

For the smallest chest sizes - with compressors of up to TL4 - it is, however, often possible to use 5 mm tubing.
If the so-called "D"-tubes are used, and good contact is ensured between the plane of the tube and the shell, this will lead to a better heat transmission coefficient and thereby reduced tube length. Thus, a smaller volume and refrigerant charging are obtained, which means improved start and pull-up conditions for the compressor.

13.
Skin type condenser for refrigerators and upright freezers

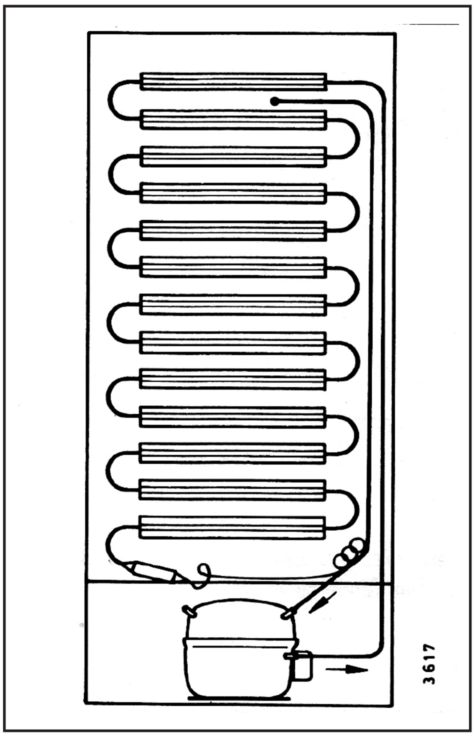


Section throught upright freezer

Fig. 12

When a skin type condenser is used for refrigerators or upright freezers, the condenser coil is usually mounted on the outside of the rear of the cabinet, as shown in fig. 12. As in the case of skin type condensers for chest freezers, a good contact should be ensured between tubing and shell by the use of suitable tube clamps. When the condenser coil is fitted outside the shell, thermal mastic is not used.
The tube layout also follows the same lines as in chest freezers, i.e. horizontal and with the inlet at the top.

Fig. 13 shows an example of a refrigerator or upright freezer with a skin type condenser. This type of skin type condenser will provide a sufficient condenser area for most refrigerators when the entire rear of the cabinet is used.
In upright freezers it will, normally, also be necessary to use an anti-dew coil in order to keep the condensing temperature at a reasonably low level.
It is necessary for this condenser type to have an adequate air circulation around the condenser, and the shell should, therefore, be designed so that there is air duct of 40-50 mm behind the condenser.
When selecting the necessary condenser coil, the entire rear of the shell should be utilized as condenser area. The tube spacing should be selected between 60-100 mm to obtain a reasonable U-value.



Upright freezer with skin type condenser

Fig. 13

14.
Anti-dew coil

The object of the anti-dew coil is to raise the surface temperature of the frame of the upright freezer, thus avoiding dewfall.
Fig. 14 shows the anti-dew coil in dotted lines around the frame of an upright freezer.

The anti-dew coil is only used in upright freezers because there is a rather large temperature differential between the inside and outside (as compared with refrigerators, for example), and because the insulation thickness in the frame is smaller than in chest freezers.
The anti-dew coil can be connected to the system in two different ways. The most effective heating of the frame is obtained when the anti-dew coil is inserted between the discharge connector of the compressor and the inlet of the condenser.

If pronounced cold bridges exist in the frame between the outer and inner shells it may be advantageous to reduce the effect of the anti-dew coil so that heat ingress into the cabinet becomes limited. If this is done, the anti-dew coil should be inserted between the condenser outlet and the capillary inlet.
If a suspicion exists that the anti-dew coil causes too strong an ingress of heat into the cabinet, a test can easily be made by interrupting (short-circuiting) the anti-dew coil and then recording any temperature drop in the air inside the cabinet.

Anti-dew coil on upright freezer

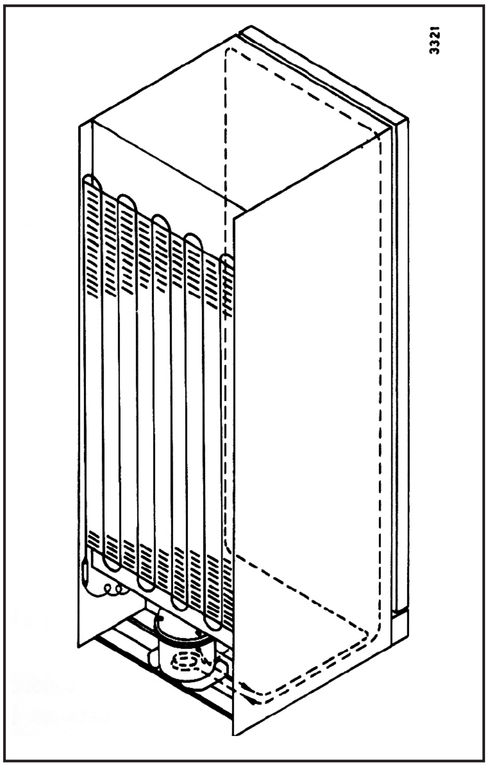


Fig. 14

In systems with oil-cooled compressors the anti-dew coil is often used as an oil-cooling condenser. Since the heat-emission effect of the anti-dew coil is relatively large, it is sufficient for some systems to use the anti-dew coil as oil-cooling condenser. However, if the compressor works under stressed operating conditions, and if optimum oil cooling is required, the antidew coil should be supplemented by part of the rear condenser.
Fig. 15 shows two examples of a frame design for upright freezers.

(A) shows a suitable link between the inner and outer shells which does not contain a cold bridge, and where the anti-dew coil is, therefore, fitted in close contact with the frame.
(B) shows a cold bridge in the frame. In order to avoid too strong an ingress of heat into the cabinet, the anti-dew coil is fitted farther away from the frame. The optimum solution will, however, be a change in frame design to eliminate the cold bridge

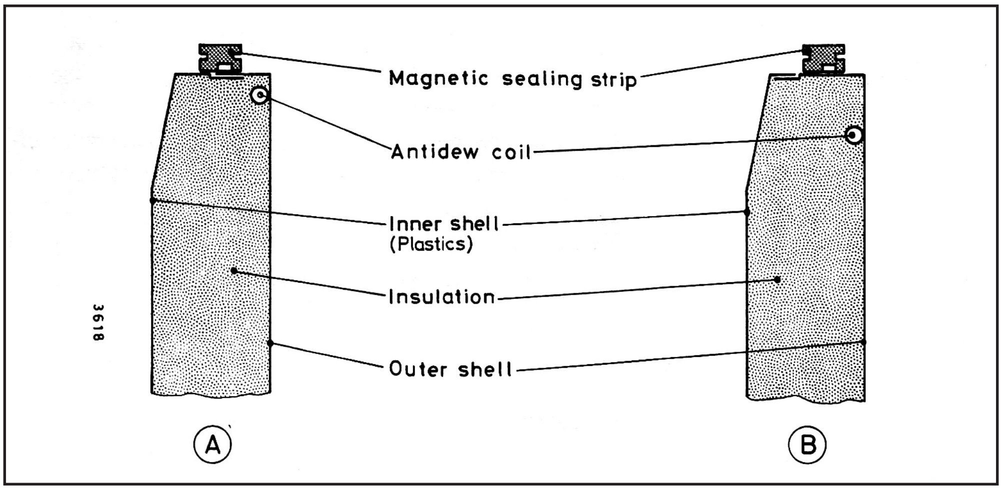


Fig. 15

15.
Oil-cooling
condenser

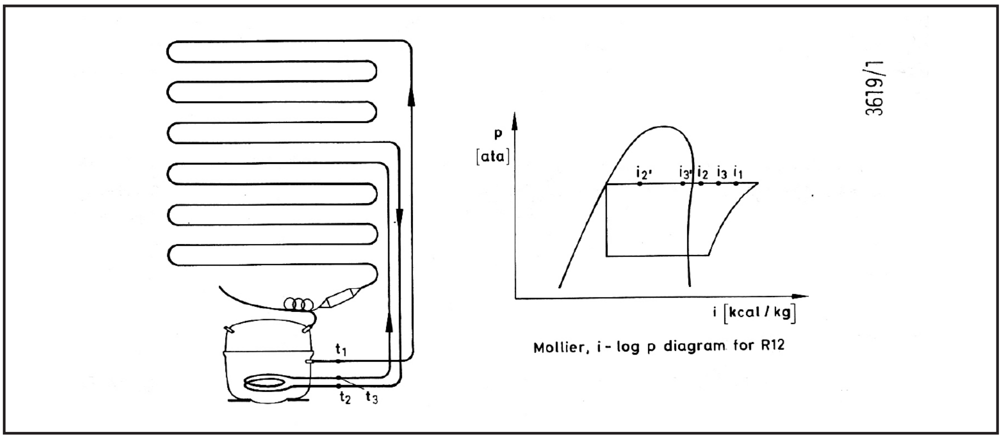


Fig. 16

Fig. 16 shows a refrigeration system with oil cooling of the compressor. The compressed gas (t_1) from the compressor is supplied to the oil cooling condenser where the superheat is removed; besides which a partial condensation of the refrigerant often takes place in the oil-cooling condenser.
From the oil-cooling condenser the refrigerant is conducted through the oil-cooling coil of the compressor where the enthalpy increases because the refrigerant absorbs heat from the compressor oil. From here the medium is drawn to the main condenser where the final condensation takes place.

16.
Dimensioning

Fig. 16 shows a Mollier, i-log p diagram with a working line drawn in for an oil-cooled hermetic refrigeration system. The figures 1, 2 and 3 refer to the corresponding designations of the refrigeration system.
In can be seen from the diagram how a large portion of the superheat of the compressed gas is removed in the oil-cooling condenser between points 1 and 2. It can also be seen that the enthalpy of the refrigerant receives an increase of $i_3 - i_2$ while the refrigerant passes through the oil-cooling coil from point 2 to point 3.

The effect of the oil cooling depends on how strongly the refrigerant is cooled in the oil-cooling condenser. Fig. 16 also shows the working line for a hermetic refrigeration system in which the oil-cooling coil of the compressor is utilized to the best possible degree. Optimum oil cooling is obtained by dimensioning the oil-cooling condenser so that the superheat of the compressed gas is eliminated and, furthermore, so that a partial condensation of the refrigerant is produced to prevent the thermal increase in the oil-cooling coil from creating superheat. It is in this way the increased enthalpy of $i_3 - i_2$ is achieved.
Two results will be obtained by using the optimum oil-cooling condenser:

As low a temperature as possible on the entire oil-cooling coil.

Optimum value of the internal heat transmission coefficient of the oil-cooling coil, α

If optimum oil-cooling is required, however, it is recommended that 50% of the total condenser surface should be used as oil-cooling condenser.

The temperature can be measured at the inlet and outlet of the oil-cooling coil on heavy loading of the compressor (for example, the combination of maximum ambient temperature and maximum overvoltage), and for this condition the oil-cooling condenser can be made larger until there is no temperature difference between inlet and outlet.

If oil-cooling is introduced in a system not previously oil-cooled, and if part of the original condenser is used as oil-cooling condenser some increase in condensing temperature will occur. Therefore, usually, it will be necessary to increase the area of the main condenser in order to keep the condensing temperature to an acceptable low level. If, on the other hand, an optimization is made of the oil-cooling condenser in an existing system with oil cooling, it will often be unnecessary to increase the total condenser area since the increase of the condensing temperature will be of a limited order.

When considering whether the area of the main condenser should be increased by the introduction or optimization of oil cooling, it should be remembered that chest freezers with a skin type condenser are very sensitive to increases in the condensing temperature since this will mean increased ingress of heat into the evaporator.

At optimum oil cooling, i.e. with approx. 50% of the total condenser area used as the oil-cooling condenser, it is reckoned that no temperature increase will occur from the inlet to the outlet of the oil-cooling coil, and it is possible to anticipate a reduction of the winding temperature of 15-30°C.

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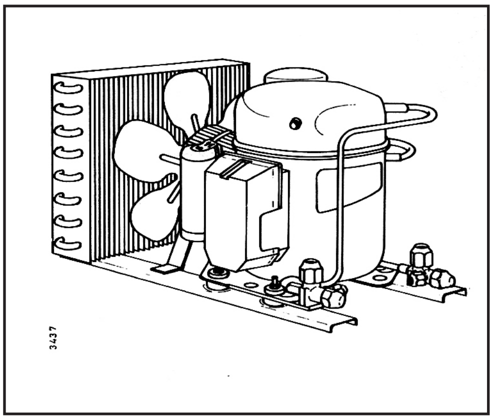
17. Fan-cooled condenser



Fan cooled condensing unit

Fig. 17

For domestic refrigeration appliances and small commercial refrigeration systems, the fan-cooled condenser is, normally, only used in connection with "fan-cooled condensing units" which consist of a condenser, a fan, and a compressor (see fig. 17). An example of this field of application is the chest freezer designed for tropical conditions where, with a statically cooled condenser, it can be difficult to provide space for the necessary condenser area.

The fan-cooled condensing units are also seen in special designs of domestic refrigeration appliances, e.g. chest freezers where the outer tank is made of plastic so that it is impossible to use a skin type condenser. Additionally the unit is used for types of refrigeration appliance where special building-in conditions leave no space for a statically cooled condenser, or alternatively do not permit sufficient air supply to the condenser.

Liquid coolers, air coolers, vending machines, and refrigerated shelves can be mentioned as more typical applications of the fan-cooled condensing unit.

The most important advantages of using fan-cooled condensing units are:

High condenser capacity for where building-in conditions are poor or for where climatic conditions are unfavourable.

Improved facilities for servicing the finished refrigeration system.

Installation advantages, especially for the small-scale producer of refrigeration appliances.

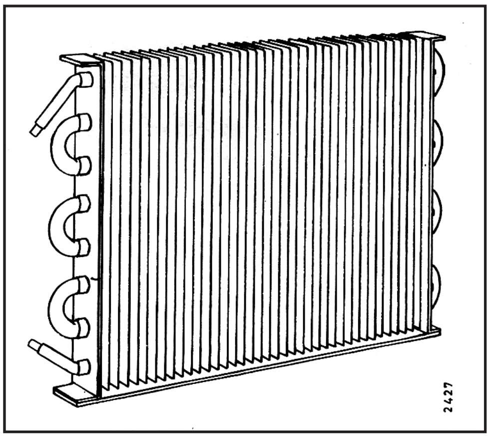
Among the disadvantages are:

Fan cooling increases the noise level.

The use of a fan makes the need for servicing more frequent (fan failure, condenser collects dirt).

Usually, the fan-cooled unit is a more expensive solution.

A finned condenser, as shown in fig. 18, is normally used as condenser for the fan-cooled unit.



Finned condenser

Fig. 18

Tube-on-plate condenser

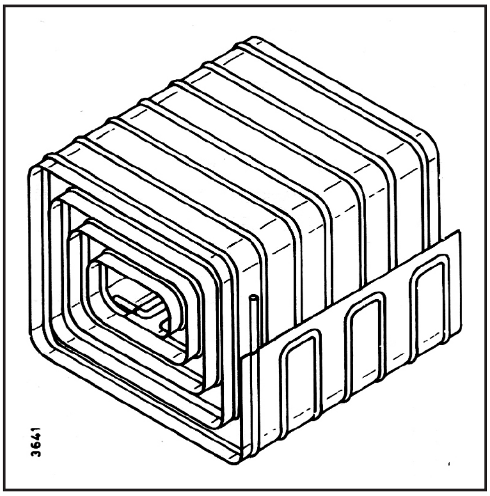


Fig. 19

There are other condenser designs too and fig. 19 shows a tube-on-plate condenser for fan cooling.
This condenser is in the form of a coiled plate, which may be made of a roll-bond panel.
However, the finned condenser is most common and the following description deals with this type only.

18. Finned condenser

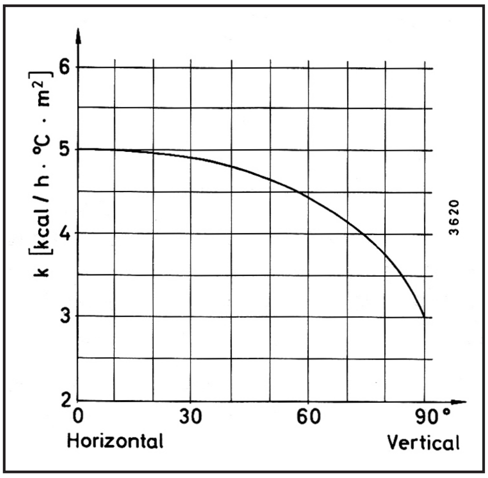


Fig. 20

Fig. 20 shows the U-value of a statically cooled finned condenser, depending on the inclination of the condenser.

It should be noted that the U-value is highest when the condenser is mounted horizontally, and lowest for the vertical position.

The U-value of the condenser, and hence its capacity, increase strongly with increasing air velocity. See section 21.

Literature reference: H.L. von Cube.
G. Tofahrn: "Formes economiques de condenseurs pour armoires frigorifiques menagers".

19. Fins and tubes

Condenser fins are characterized by their spacing, thickness, material, and the method of fixing to the tubes. Also of importance is whether the fins are plane or corrugated. However, this complex of problems is not dealt with here.

In the case of fan-cooled condensers for small refrigeration systems the fin spacing varies between approx. 2 and 10 mm.
When the fin spacing is determined, the risk of dirt collection will point to a large space between fins. Demands for small overall dimensions result in small fin spacing. However, the U-value falls when fin spacing is reduced, therefore the larger surface area obtained in this way cannot be fully utilized to reduce the overall dimensions - if condenser capacity must be maintained.
Examinations would indicate that the noise level of the condensing unit increases when the fin spacing is reduced. The fins are made of steel or aluminium of a thickness 0.2-0.5 mm.
When the material and thickness of the fins are to be chosen, a sufficiently rigid and robust construction with an adequate heat transmission capacity must be ensured.

In case of fin spacing less than 6 mm and of more than 2 rows of tubes a cooling baffle between condenser and fan must be used.

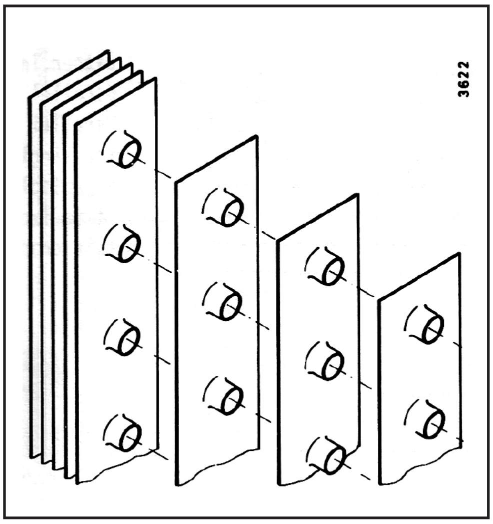


Fig. 21

The contact between condenser tubes and fins is of great importance to the condenser capacity. The fins are equipped with holes which may have a small flange so that the surface in contact with the tube is increased. This flange can also serve as a spacer between the fins. Some makes of condenser utilize the flange in a sandwich design where the tube is omitted and the single fins are copper soldered to form a tube, as shown in fig. 21.
Determining the diameter of the condenser tube is a compromise between, on the one hand, the material cost and limitation of the pressure-equalizing time which both point to a reduction of the tube diameter, and on the other hand, a limitation of pressure loss and provision of a sufficient internal volume through a sufficiently large tube cross-section.
Furthermore, it must be remembered that fan-cooled condensers are used for highly different applications and load conditions.
The typical tube diameter of fan-cooled finned condensers for small refrigeration systems is 10 mm.
The spacing of the condenser tubes affects the U-value of the condenser since a smaller tube spacing results in a higher U-value.
However, there are many indications that the economical optimum tube spacing ranges between 25 and 35 mm (from tube centre to tube centre).

Normally condenser tubes are made of steel.

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20. Tube layout

Incorrect an correct tube layouts

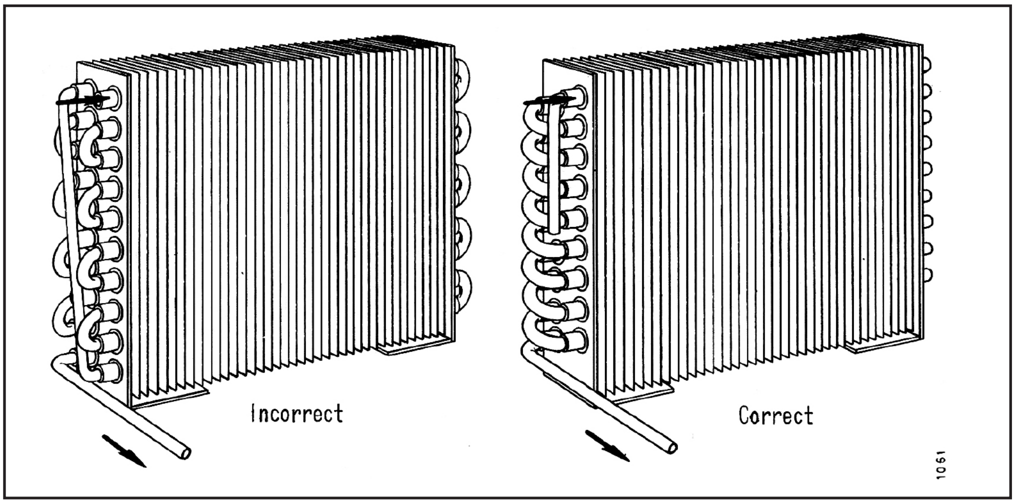


Fig. 22

As is the case with all other condenser types, the tube layout of the finned condenser should ensure a minimum of pressure-equalizing time.

This can be achieved by locating inlet at the top, outlet at the bottom, and continuously declining the tube coil so that liquid pockets are avoided. Fig. 22 shows examples of incorrect and correct tube layouts. In the former case it can be seen that the continuity has been broken, resulting in a liquid pocket with a prolonging effect on the pressure-equalizing time.

Series and parallel flow systems

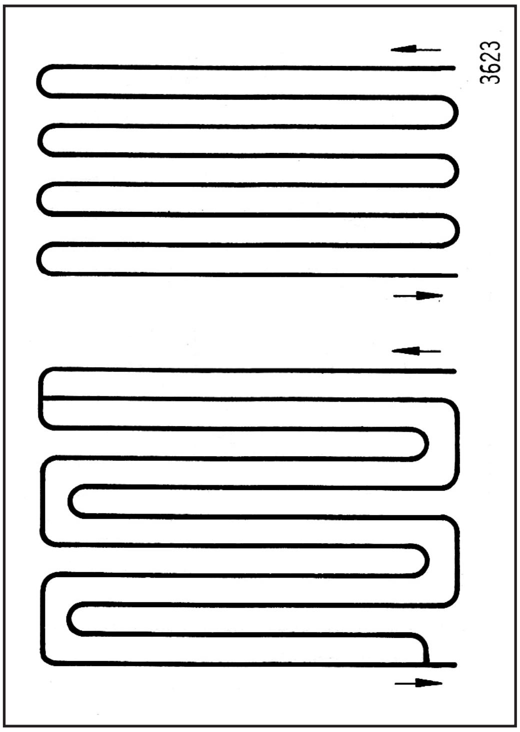


Fig. 23

In principle, the tube layout can be either in the torn of a single flow or a parallel flow system.
Parallel flow should only be used where the single flow system results in an unacceptable large pressure drop and where it is impossible to change to a larger tube cross-section.

21. Condenser area

Condenser area
(32°C ambient temp.)

Condenser	Code No.	119-0081	119-0082*	119-0085*	119-0088*
	Surface area m^2	1.16	1.73	3.22	4.28
Compressor capacity cm^3 displacement	M/LBP R12	0-10	8.5-15	12-21	18-
	HBP R12	0-5.5	4.5-8.5	7.5-15	12-21
	LBP R502			10-15	12-18
			*With cooling baffle		
K-values (kcal/h x °C x m^2)		18	15.6	17.3	18.7

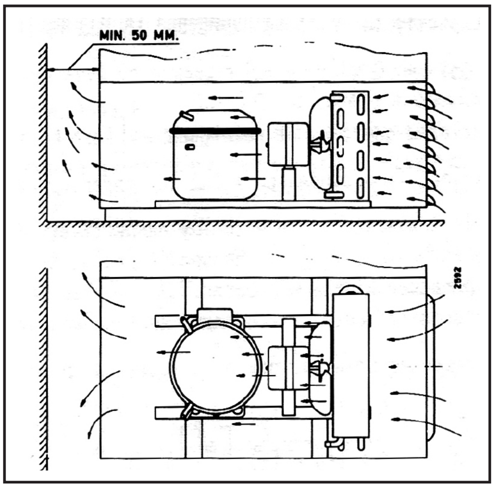
Fig. 24

The U-value of the condenser and hence the condenser capacity depend to a high degree on air yield, condenser position, fin spacing, etc., and it is, therefore, impossible to lay down unambiguous rules for the necessary condenser area in fan-cooled condensers.

Fig. 24 shows relations between compressor size and condensers for Danfoss condensing units.
The compressor capacity is expressed by compressor displacement.

The table applies to two-pole 50 Hz compressors in the ranges of M/LBP (-40°C to -5°C) and HBP (-5°C to +15°C]. The recommended condenser sizes are sufficient for ambient temperatures of up to 32°C.
The air velocity at the condenser inlet is assumed to be approx. 1 m/s.

22.
Conditions- for built-in units



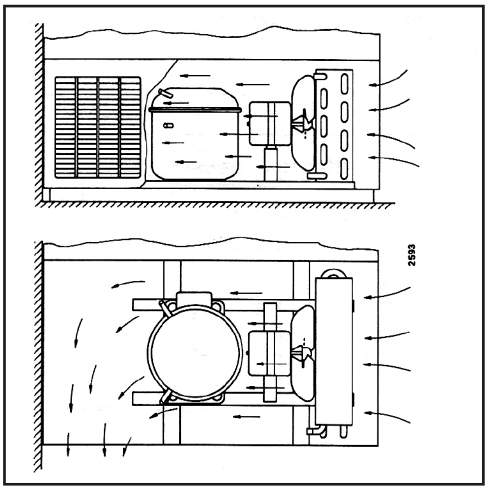
Unit in motor compartment

Fig. 25

Common sense considerations govern the installation of a built-in unit. If the fan is prevented from drawing the required amount of air through the condenser and across the compressor, performance will not be as anticipated.

Fig. 25 shows a condensing unit built in under ideal conditions. The inlet opening, which can be in the form of ventilation slits, has a flow area which corresponds to the front of the condenser. Air can pass right through the motor compartment so that it circulates evenly around the compressor. Moreover, the installation of the refrigeration appliance in this way ensures that the air is conducted away with no major resistance.

If the refrigeration appliance is to be placed directly against a wall, it will be necessary to accept a compromise in some form. For instance by placing the vent opening in the free side of the motor compartment. Acceptable conditions can be provided in this way - more especially if the unit is shifted nearer towards the air intake side at the same time (fig. 26).



Unit in motor compartment

Fig. 26

23.
Condenser calculations

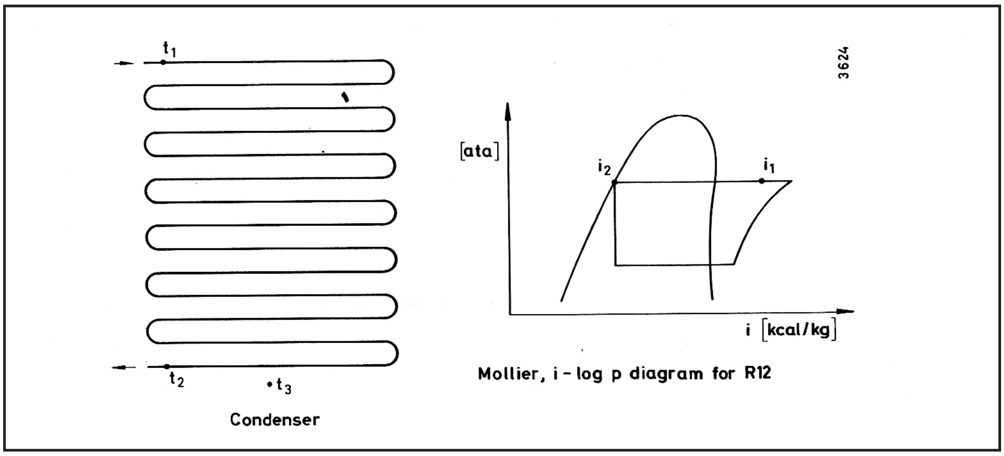


Fig. 27

The condenser capacity can be calculated by means of the formula:

$$Q_k = K \times A \times \Delta t = K \times A \times (t_2 - t_3)$$

Q_k : Condenser capacity [kcal/h]

K : Heat transmission coefficient of condenser [kcal/m^2 x h x $^\circ$C]

A : Effective condenser area [m^2]

Δt : Temperature differential between condensing temperature and the ambient air [$^\circ$C]

The condenser capacity can also be expressed in the following way:

$Q_k = G \times \Delta i = G \times (i_1 - i_2)$

G = The circulated quantity of refrigerant [kg/h]

Δi = Enthalpy loss from condenser inlet to condenser outlet [kcal/kg]

t_1 (i_1) designate temperature and enthalpy respectively at the compressor discharge connector/condenser inlet, t_1 (i_2) the conditions at the condenser outlet while t_3 is the ambient temperature at the condenser. $t_2 - t_3$ can be used with good approximation as temperature differential between condenser surface and air.

The necessary condenser area can now be calculated in the following way:

$$A = \frac{G \times \Delta i}{K \times \Delta t} = \frac{G \times (i_1 - i_2)}{K \times (t_2 - t_3)} \ [m^2]$$

To determine the necessary condenser area, the dimensioning rules mentioned in section 02 should be taken into consideration.

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24. Measuring the condenser capacity

Measuring set-up

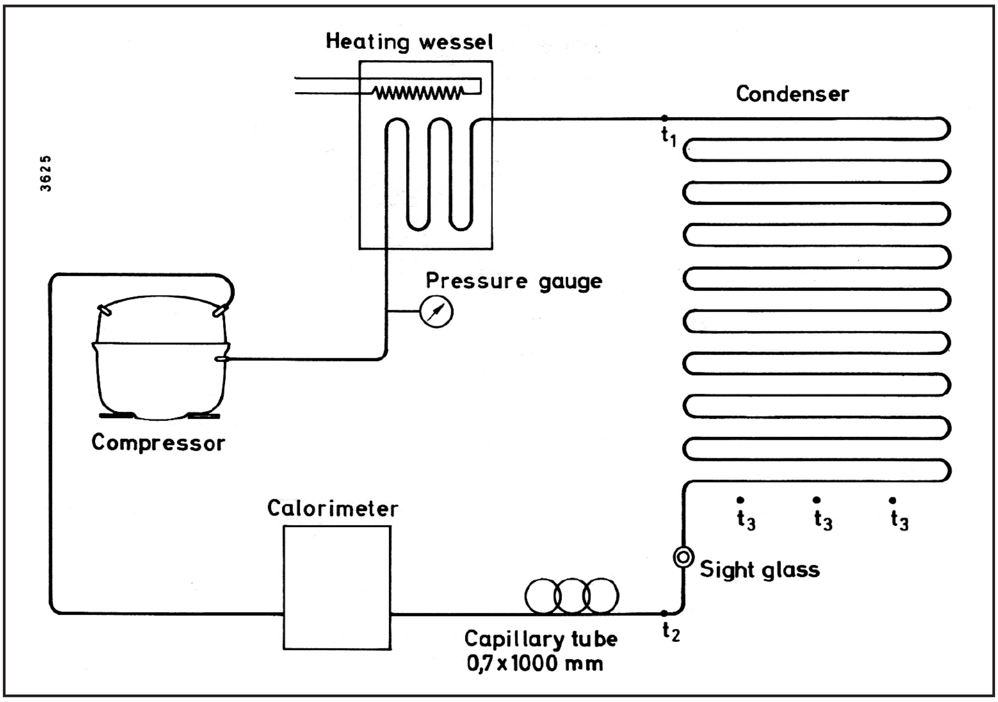


Fig. 28

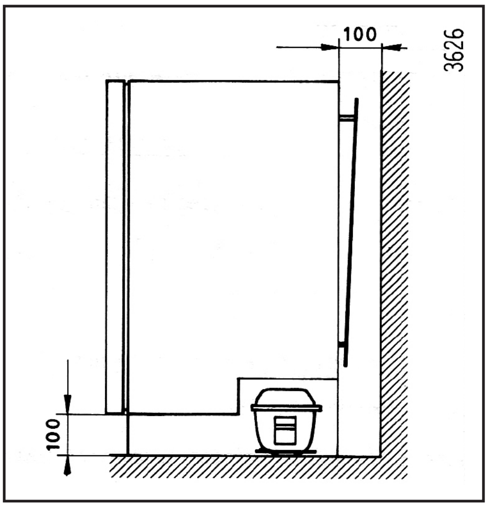


Fig. 29

The condenser capacity is determined either by measurement on the condenser alone or by testing the condenser mounted in the refrigeration appliance.
Measuring the condenser capacity can be done by the use of a set-up as shown in fig. 28.
The condenser must be fitted under conditions as close to reality as possible, as shown in fig. 29.
If the compressor in fig. 29 is replaced by a heating element, it will be easier to keep the inlet temperature constant.

The trial set-up is designed so that the R12 gas is supplied to the condenser at a superheat of 100°C and is removed as condensate. At the condenser outlet a capillary tube is fitted which makes setting to the required conditions easier.

The following procedure should be used to operate the measuring set-up:

The suction pressure (evaporating pressure) and hence the circulated quantity of R12 are regulated until the liquid level is constant in the sight glass. Since there is only one point of loading where this position of equilibrium occurs in a given system, only a single measurement per condenser size is possible. Therefore, this measuring method offers no possibility of determining the U-value of the condenser as a function of the temperature differential.
The condenser capacity can be calculated by multiplying the circulated quantity in kg/h by enthalpy difference between the compressed gas superheated to 100°C and the liquid at the lower limit curve.

$$Q_k = G \times (i_1 - i_2) = K \times A \times \Delta t = K \times A \times (t_2 - t_3) \text{ [kcal/h]}$$

The U-value is calculated on the basis of the condensing temperature minus the air temperature at the condenser inlet.
The measuring set-up shown has a standard calorimeter inserted in the circuit so that the circulated quantity can be calculated.

Measuring set-up

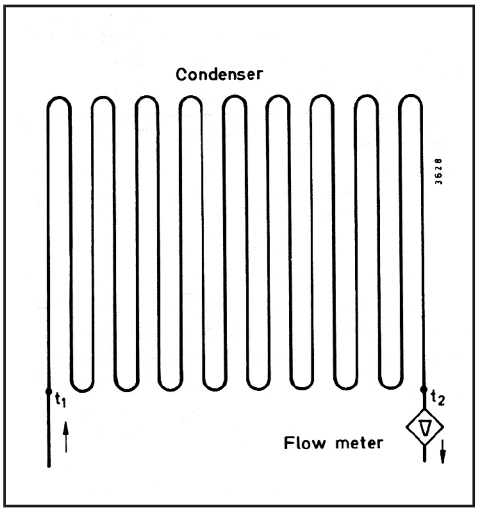


Fig. 30

For a relative comparison between different compressor types or makes of condenser a set-up like the one shown in fig. 30 can be used. Hater at a constant velocity and inlet temperature is fed through the condenser.

The temperature differential between inlet and outlet is recorded, and the water flow is measured with a flowmeter.

In comparing two condensers, it is possible either to keep the circulated quantity of water constant and to record the difference in temperature differential, or the temperature differential can be kept constant and the difference in the circulated quantity of water can be recorded.

The capacity can be determined according to the formula:

$$Q_k = G \times c \, (t_1 - t_2)$$

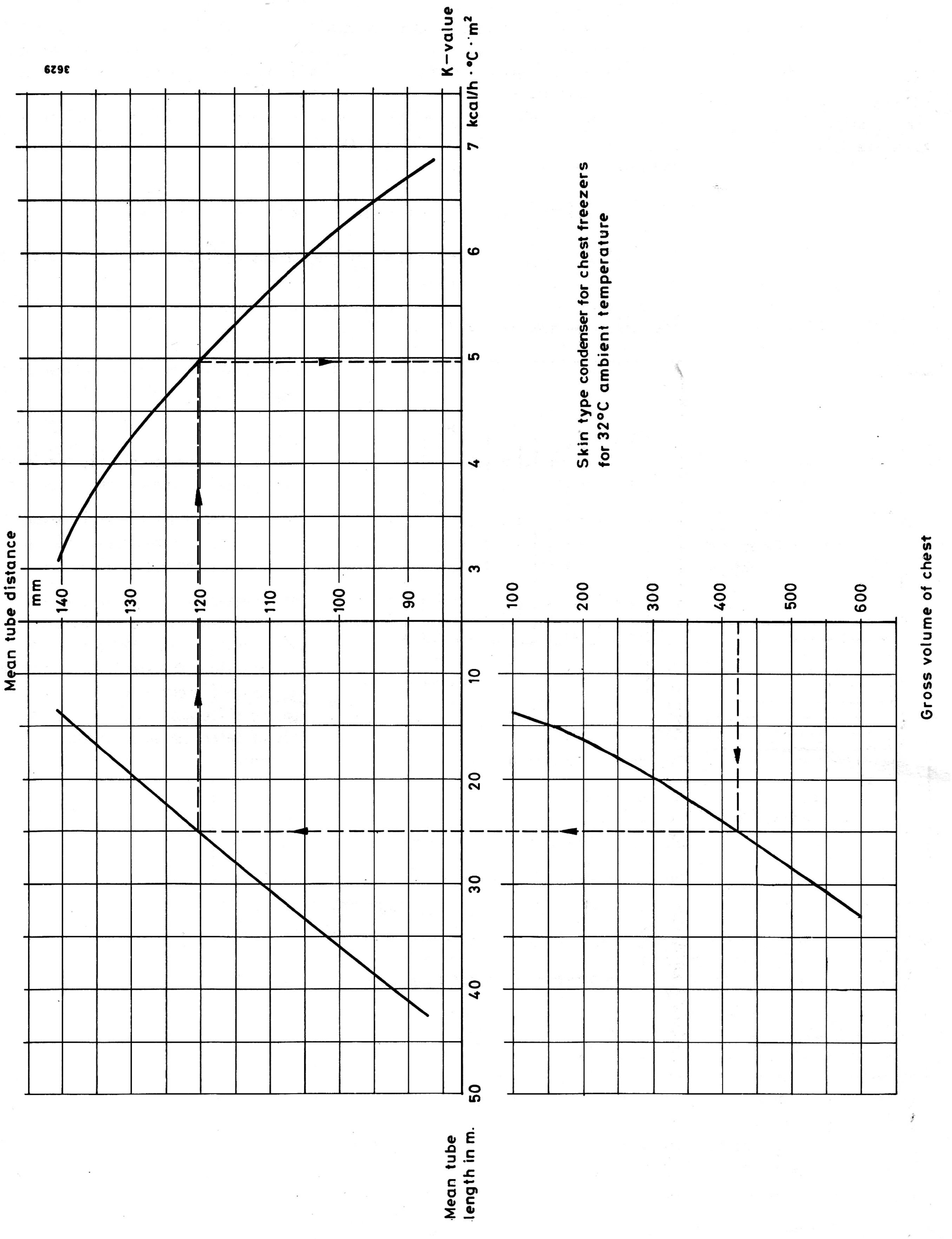
Q_k = Condenser capacity (kcal/h)

G = Quantity of water (kg/h)

c = Specific heat content (kcal/kg x °C)

t_1 = Temperature at inlet (°C)

t_2 = Temperature at outlet (°C)



3629
K-value
kcal/h · °C · m²
Mean tube distance
mm
140
130
120
110
100
90
3
4
5
6
7
100
200
300
400
500
600
10
20
30
40
50
Skin type condenser for chest freezers
for 32°C ambient temperature
Gross volume of chest
Mean tube
length in m.


MERCER
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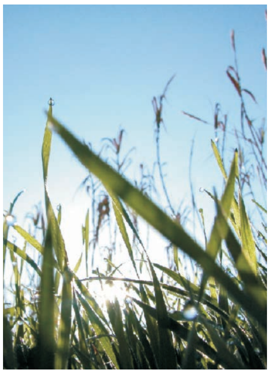
OUR IDENTITY

At Secop we are committed to our industry and are genuinely passionate about the difference we are able to make for our customers. We understand their business and objectives and the challenges of today's world of refrigeration and cooling systems.

We work in a straightforward way, being open, direct and honest because we want to make things clear and easy. Our people are committed to increasing value for our customers and constantly strive for better performance, knowing that our own progression and success is dependent on theirs.



OUR JOURNEY SO FAR

SECOP

1956
Production facility and headquarters in Flensburg, Germany founded.

1958
Start of production for PW compressors.

1970
Introduction of SC compressors.
The birth of a standard-setting platform in the light commercial market.

1972
Introduction of FR compressors.

1977
Introduction TL and BD compressors.

1990
Introduction of NL compressors.

1992
Introduction of PL compressors.

1993
Start of production with natural refrigerant R600a (Isobutane).
Production facility in Crnomelj, Slovenia founded.

1999
Start of production with natural refrigerant R290 (Propane).

2002
Production facility in Zlate Moravce, Slovakia founded.

2005
Introduction of GS compressors.

2008
Production facility in Wuqing, China founded.

2010
Introduction SLV-CNK.2 and SLV-CLK.2 variable speed compressors.
Introduction BD1.4F Micro DC compressor.
Introduction of DLX and NLU compressors.

2013
Introduction of the XV compressor – opening a new chapter in refrigeration history.
Secop acquires ACC Fürstenfeld, Austria.

2015
New generation of energy-efficient propane compressors.
New variable speed platforms for household and light commercial applications.

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