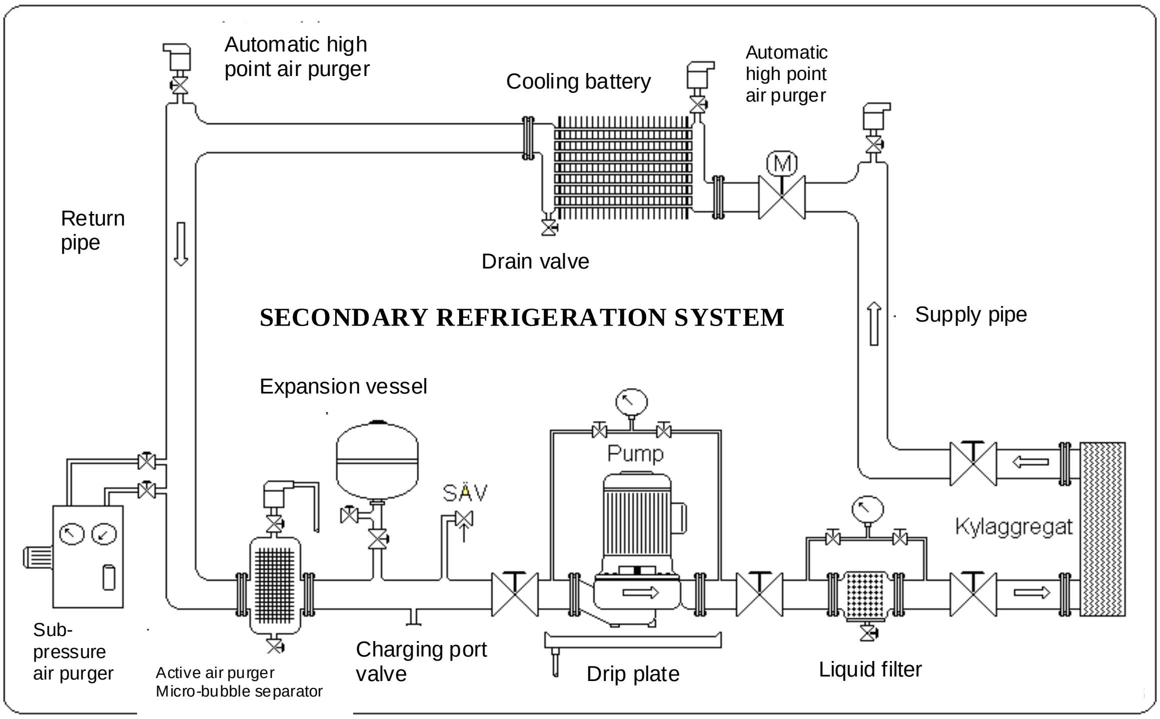
Handbook on indirect refrigeration and heat pump systems



Placement of components in an indirect refrigeration system (from a publication of the Secondary Refrigerant Team)

Produced within Effsys2 P2 with funding from Energimyndigheten and cooperation from KTH and a large number of companies and persons

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SUMMARY

Most refrigeration systems are constructed as direct systems, i.e. systems with direct expansion (DX) but indirect systems with secondary working fluid */ circuits have long been used for systems where there are many places to be cooled or where long pipes are required. Examples of systems with many places to be cooled are larger slaughter houses and dairies. Long pipes are common among else at air conditioning plants, artificial ice rinks and ground source heat pumps with long collector tubes in the ground.

Indirect systems have more and more replaced systems with direct expansion (DX-system) within commercial (supermarket) refrigeration, where the secondary working fluid transports heat from cooling rooms or cabinets to achieve the right temperature of the products. Indirect systems are even used at freezer applications but more and more with carbon-dioxide as phase changing secondary working media.

Indirect systems have more and more come in focus as requirements of more tight constructions are sharpened in order to minimize refrigerant leakage from plants as well as the adjusted legislation to secure the phasing out of various types of CFC- and HCFC- refrigerants.

This publication has been produced in cooperation with over twenty companies and persons (See Acknowledgements). Various sources such as books, reports, magazine articles, company information have after agreements been considered.

The chapters deal with the following topics: 1. Direct and indirect systems; 2. Indirect system solutions, 3. Components, 4. Secondary working fluids, 5. Corrosion in indirect systems, 6. Environmental aspects, 7. Projecting and dimensioning, 8. Design and control of system, 9. System construction and installation and 10. Optimization of indirect systems.

An expectation of all who have been involved in the work to produce this handbook or publication is that it will be an aid to understand and work with indirect systems.

*/ *secondary working fluid* often named secondary refrigerant, secondary coolant, brine

ACKNOWLEDGEMENTS

- The Swedish state power board has financed much of the work through the Effsys 2 program.
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- Authors and editors of various types of books, reports, magazine articles, company information have allowed the use of this material in this handbook.

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1. Direct and indirect systems (Å. Melinder)

1.1. Direct systems

Most refrigeration systems are constructed as direct system, i.e. systems with direct expansion (DX), where the refrigerant is used to directly transport heat from the space to be cooled to the space where heat is released. Besides the refrigerant a direct system has four main components, evaporator, compressor, condenser och expansion valve. The main principle is that heat from the cooling object or heat source is taken up in the evaporator and released in the condenser.

The refrigerant changes from liquid to vapour or gas in the evaporator that is placed in the space to be cooled. The pressure (and temperature) of the refrigerant gas is increased in the compressor and the gas then goes to the condenser where the refrigerant is cooled and condenses to liquid. The liquid then goes to the expansion valve, a throttling device where the pressure of the refrigerant is reduced, and then back to the evaporator and the process starts al over again (See Figure 1.1).

A separate cooling machine with compressor is often used for each evaporator and cooling unit. In other cases several or all evaporators and cooling units in a system are connected to one or more compressors splaced in a central machine room. The refrigerant is then often transported through long tubes, requiring relatively large refrigerant charge. (KBL1 1999).

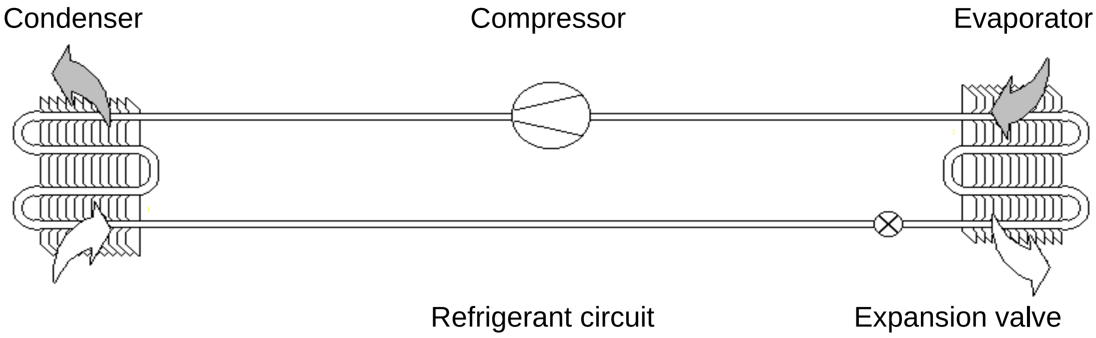


Figure 1.1. Direct system

1.2. Indirect systems with secondary fluid circuit

1.2.1. Introduction

A secondary working media or fluid (secondary refrigerant, secondary coolant, brine) here called secondary fluid is in an indirect system (IS) used to transfer or remove heat from the cooling object or heat source to the evaporator of the plant. Another secondary fluid can be used to remove the heat rejected in the condenser of the plant. It is here called secondary fluid for the condenser side.

Indirect systems with secondary fluid circuits have long been used for systems with many units to be cooled or where long tubes or are needed. Today there are many indirect systems that have been in use for several decades and that have functioned very well all this time. Exemples of systems with many units to be cooled are larger butcheries and dairies where direct systems often have proved to be more expensive and more complex. Examples of applications with long tubes or pipes are artifical ice rinks, air conditioning plants and ground source heat pumps where there are long collector tubes.

1.2.2. The principle of an indirect system

Indirect refrigeration systems are divided in two types of circuits, a primary refrigerant circuit and one or two secondary circuits, one on the (cold) evaporator side and one on the (warm) condenser side. Besides components of the direct system there are secondary fluids, heat exchangers, tubes and circulation pumps for these secondary circuits. (See Figure 1.2).

One of the secondary fluids transports heat from heat source or cooling object to the evaporator. Indirect systems have more and more replaced systems with direct expansion (DX-system) in commercial supermarket applications, where the secondary fluid transports heat from the cooling room or cabinet to ensure the right temperature of goods. Indirect systems are also introduced at freezer applications and carbon dioxide is more and more used as phase changing secondary working media or fluid.

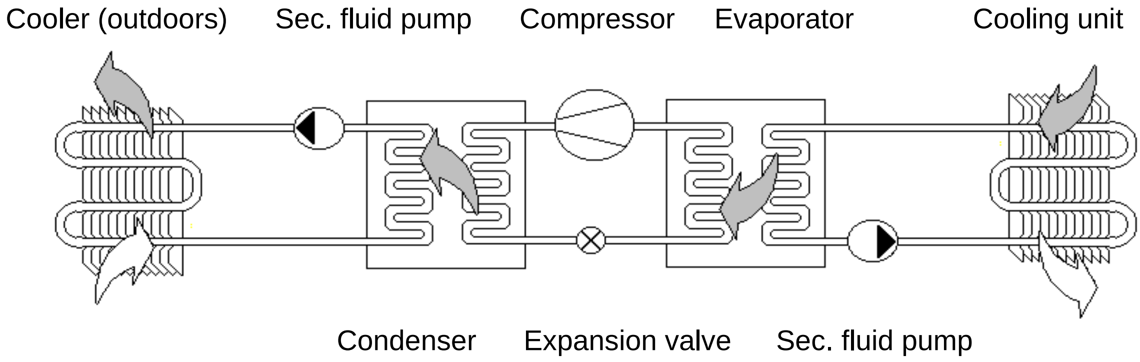


Figure 1.2. Indirect refrigeration system

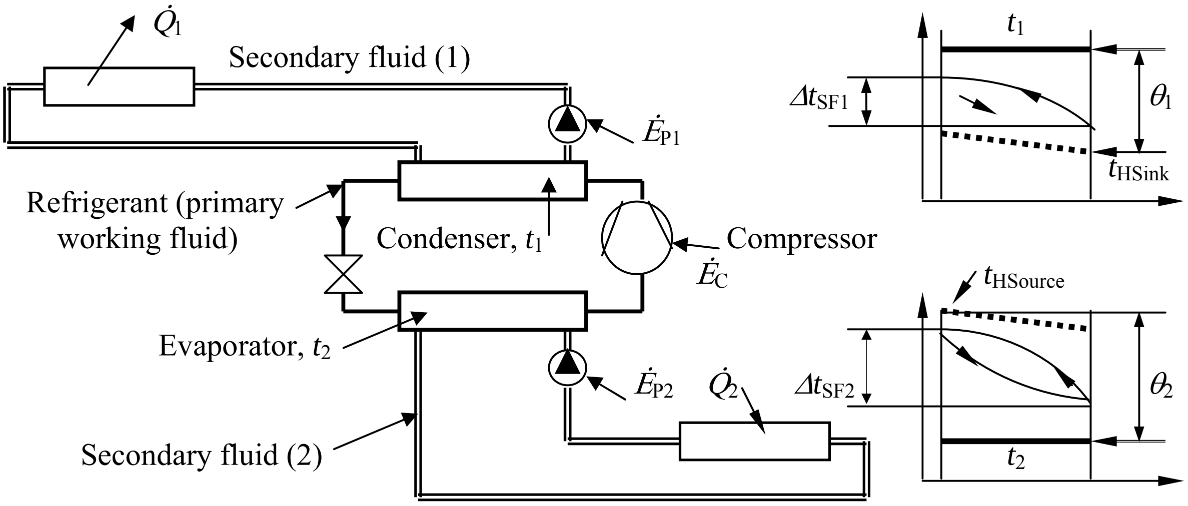


Figure 1.3. Complete indirect system with nomenclature and temperature profiles

Explanations of nomenclature in Figure 1.3:

$\dot{Q}_1$ = released heat capacity; $\dot{Q}_2$ = cooling capacity; $\dot{E}_C$ = compressor power; $\dot{E}_P$ = pumping power; t_1 = condensing temp.; t_2 = evaporating temp.; t_{HSink} = heat sink temp.; $t_{HSource}$ = heat source temperature; θ_1 = temperature difference between condensing temperature and heat sink; θ_2 = temperature difference between heat source and evaporating temperature: Δt_{SF1} and Δt_{SF2} are temperature changes of secondary fluids (1) and (2).

Indirect refrigeration systems with secondary fluid circuits are often constructed with compact refrigeration units where compressor, evaporator and condenser are placed in the machine room. This type of system makes it possible to drastically reduce the refrigerant charge (down to 5-15%). Since these units are placed in the machine room this arangement contributes also to a controlled use of natural refrigerants.

1.2.3. Reasons for increased use of indirect systems

Indirect systems have gained relevance as demands of tighter constructions have been sharpened in order to minimize refrigerant leakage from the plants.

One reason that has contributed to the increased use of indirect systems with secondary fluid circuits is that legislation has been designed to secure the phase out of various types of CFC- and HCFC-refrigerants. The refrigerant proclamation from the Swedish Environmental Protection Agency (Naturvårdsverket) toward the end of the 1980-ies tightened up the use of refrigerants. The general awareness of how refrigerants affect the global environment has increased since then. The awareness of problems that come with natural refrigerants, for instance increased risk of fire or of panic due to strong smell.

The main principles of the refrigerant proclamation can be summed up in the following action program:

1. Change-over to other refrigerants
2. Change-over to indirect systems
3. Better and tighter constructions
4. Better care and maintenance
5. Handling and care of used refrigerant

Areas of application are construction, manufacturing, installation, rebuilding, maintenance and control of refrigeration and heat pump plants containing refrigerants and/or secondary working fluids consisting of CFC-, HCFC- and HFC- compounds as well as handling of such refrigerants and secondary fluids (Ses Svensk Kylnorm, 1991, Faktablad 6).

1.2.4. Background to the trend toward indirect systems

In 1987 the so called Montreal protocol was established and many nations agreed to lessen the use of among else CFC- refrigerants, or "freons". This was the start of a global process that changed the view of and the use of syntetic refrigerants in refrigeration and heat pump systems.

In 1989 the refrigerant proclamation from the Swedish Environmental Protection Agency was put into effect in Sweden with the purpose to prevent discharge of refrigerants that affect the ozone layer. The concept ODP is used to express the Ozone Depletion Potential of a refrigerant.

In 1997 the Kyoto protocol was established where decisions were made about the substances affecting the so called green house effect that contributes to the global warming. This agreement has as its aim to wind up the syntetic refrigerant köldmedierna CFC, HCFC and in time the newly added HFC refrigerants, partly through a transition to natural refrigerants uch as ammonia, (NH_3), carbon hydrates (HC) and carbon dioxide (CO_2).

1.2.5. Advantages and drawbacks with indirect systems

Advantages with indirect systems

Indirect systems with a secondary fluid circuit make it possible to reduce refrigerant charge drastically, to 5-15% of the charge in a DX-system, by using a compact factory made design of the refrigerant circuit. Natural refrigerants can be used in a controlled way.

No tubing of the refrigerant circuit need to be performed on site. All welding and souldering of the refrigerant circuit can be made under factory conditions which reduces the risk of refrigerant leakage.

In Sweden this has led to changes and additions to the refrigerant proclamation about substances that affect the global heating and maximum refrigerant charges in various systems. The concept GWP is now used to express the contribution of a refrigerant to the Global Warming Potential.

These global environmental decisions in the Montreal och Kyoto protocols, form the basis for many of the great changes taking place in the refrigeration and heat pump sectors. National laws and statutes differ somewhat from country to country but the clear trend is to reduce refrigerant charge in the systems and a move toward natural refrigerants. This highly affects the choice and design of system technology with a clear trend toward indirect systems.

Drawbacks with indirect systems

Drawbacks with indirect systems are the increased cost for secondary fluids, pumps, tubes and extra heat exchanger for the secondary circuits.

An indirect system also requires an extra pump and extra pump work for the secondary fluid to be pumped arount to the various objects to be cooled.

The indirect system adds an extra heat exchange and temperature difference between refrigerant and secondary fluid. This may contribute to a somewhat lower evaporating temperature and pressure.

Increased use in spite of drawbacks

Indirect systems are in spite of these drawbacks more and more replacing traditional direct systems because of the earlier mentioned advantages.

An indect system gives great flexibility as to areas of application so that the refrigeration or heat pump unit can be produced in larger series. Indirect systems facilitates installation and the installation costs are thus normally much lower with an indirect solution than with on-site-built DX-systems.

It is easier to make adjustments in the piping system on the secondary fluid circuit than on the condenser or evaporator circuit. The system is more flexible, easier to move or to add more refrigerant units.

Modern technology with speed-controlled pumps that work momentarily against the real flow need are other energy saving action that can be used vid indirekta system.

Varying refrigerating needs can with indirect plants easier be combined and can often be dimensioned for a smaller total power. A reserve for a future extension can alternatively be installed.

It is much easier to make use of the condenser heat (heat recovery) from a secondary coolant circuit than from several direct condensing systems.

Small leakage in the secondary fluid system cause less risk for operation stop than leakage in the refrigerant circuit.

1.2.6. Do indirect systems give increased energy consumption?

Indirect systems require somewhat larger power than a direct system as the secondary fluid pumps and the extra heat exchange between the refrigerant and the secondary fluid can cause a somewhat lower evaporation temperature and pressure.

Theoretically this should in itself lead to higher energy consumption but experience from the field show that this is not always the case. What are the reasons for this?

It turns out that the total energy consumption of a well built indirect system in practice often becomes somewhat lower over the year than that of a DX-system as the indirect system better can use variations in ambient conditions over the 8760 hours of the year than what a direct system can do.

Less frost and better use of heat exchanger surfaces (i.e. cooling batteries) are factors that often give lower energy consumption than traditional direct systems.

Modern pump technology with speed-controlled pumps working momentarily against the current flow demand is another energy saving device used.

(Aittomäki 2003, (KBL1 1999), KBL3 2002, Lindborg 2003, Lundqvist 2000, Melinder 2003, Svensk Kylnorm 1991, Ure 2003, Yang 2001).

2. Indirect system solutions

2.1. Various types of systems

Some examples of indirect systems are airconditioning plants, heat pumps, commercial refrigeration or supermarkets, artificial ice rinks, larger slaughter houses and dairies.

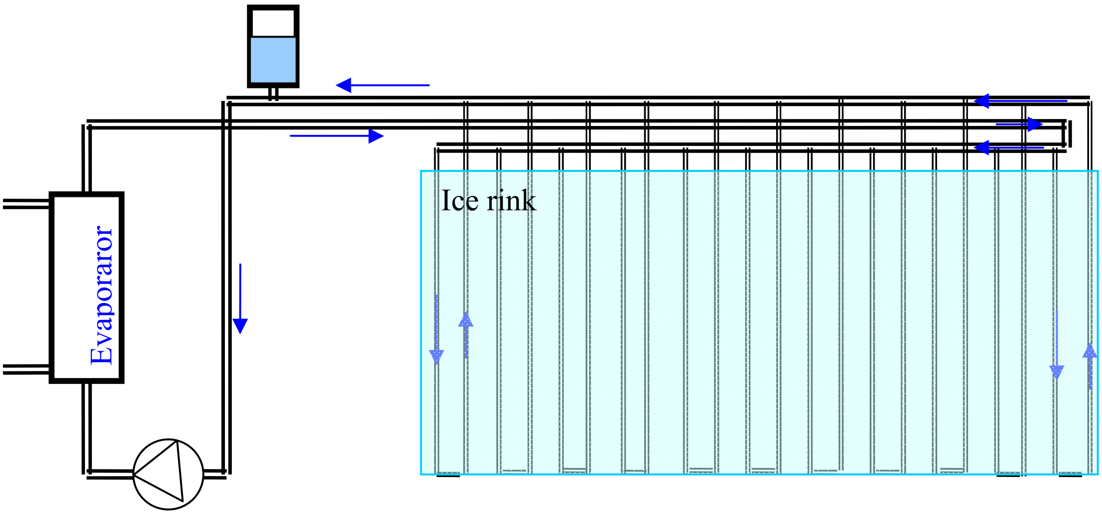
Indirect systems have long been used for systems where there are many places to be cooled where direct systems often have proved to be more expensive and more complex. Examples of systems with many places to be cooled are larger slaughter houses and dairies. Long connecting pipes are common among else at air conditioning plants, artificial ice rinks and ground source heat pumps with long collector tubes in the ground or rock. Applications such as commercial or supermarket refrigeration have more and more moved toward indirect systems due to the need to minimize the refrigerant charge and leakage.

Examples of important types of systems are given in the following sections. First, an example of design for an artificial ice rink is shown. After that, various types of system solutions for heat pumps as well as for supermarkets. A separate section deals with systems with carbon dioxide, CO_2, as secondary fluid.

There are several ways to arrange indirect systems. One klassification with definitions of used nomenclatures have been colleced in section 2.6 and Appendix 1. The material there is taken from Svensk Kylnorm, 1988 and B. Ekman, 2003.

2.2. Artificial ice rink (E. Granryd)

The following basic principal schedule can serve as a model for arrangements at an artificial ice rink. It is urgent to get as even temperature as possible in the icerink and many parallel tubes below the ice are needed to cover the surface. The scheme indicated in Figure 2.2.1 can be used to get as equal flow velocity as possible in the various tubes.



Figur 2.2.1 Tube arrangement for even flow distribution in an ice rink

2.3 Domestic heat pumps (E. Granryd)

A very large part of villas and other domestic housing in Sweden are heated with various types of heat pumps. At times they are combined with other heat sources.

The following picture taken from a brochure from the Swedish consumer agency shows the principal design of a ground source heat pump (Figure 2.3.1). Figures 2.3.2-4 show principal sketches for an ambient air heat pump, a bedrock heat pump and a lake water heat pump, all common types of domestic heat pumps in in Sweden.

Domestic Heat Pump with
shallow ground heat source

Figure 2.3.1. Ground source heat pump

Air to Air Heat Pump

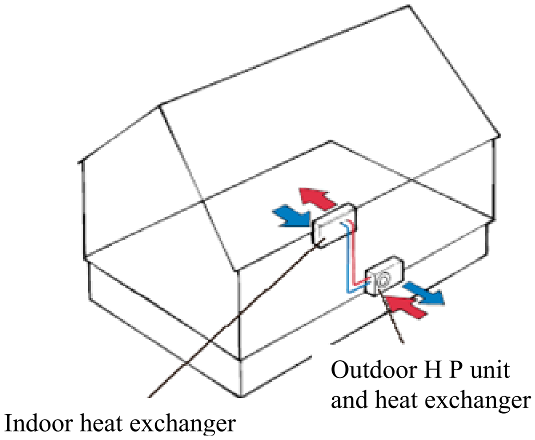


Figure 2.3.2. Ambient air heat pump

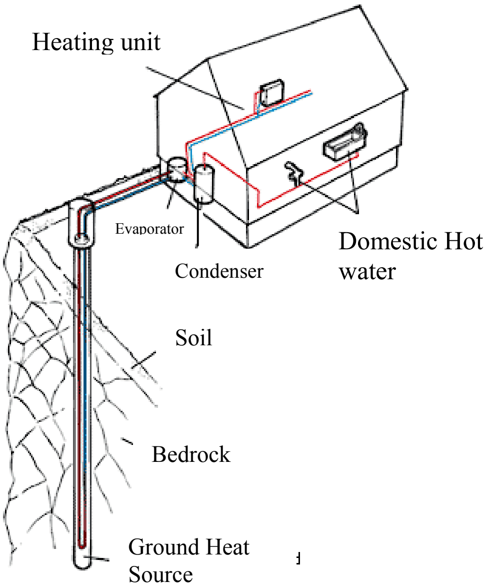


Figure 2.3.3. Bedrock heat pump

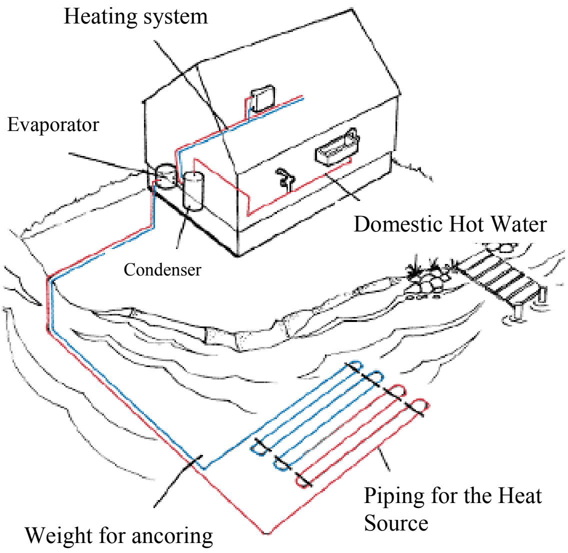


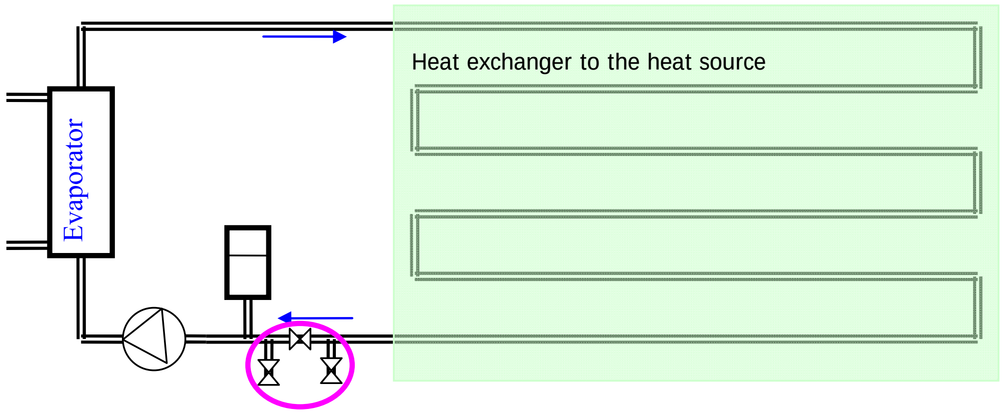
Figure 2.3.4. Lake water heat pump

A simple basic scetch of indirect systems for heat extraction at heat pumps is shown in Figure 2.3.5. The figure illustrates a surface ground source heat pump system, but the same basic solution is used for other forms of heat sources. For example bedroch heat takes heat with the aid of vertical holes in which (usually) two tubes are put sown as a U-tube.

There are several secondary fluids to choose between. Often water solutions with glycol (where it is permitted considering envoronmental aspects) or ethyl alcohol.

Only one loop is generally used in villa heat pumps in order to avoid possible problems with uneven distribution. The typical length is some 300 m and the dimension is 35 mm inner, 40 mm outer diameter. Typical flow velocity is about 1 m/s.

Typical for this type of secondary fluid system is that the heat exchanger (the tube system) has comparably large inner volume. At the start of a system it is important to get out all air so that the system is fully filled with liquid. För att åstadkomma detta brukar man använda A separate air separater that is dismantled after the charge is completed is used to accomplish this. See detail in the figure.



Filling detail:
When filling fluid to the secondary system it is beneficial to use a separate (open) container which simultaneously serves as an air separator during the filling process. By means of the three valves as shown the container can be removed after the filling process is finished.

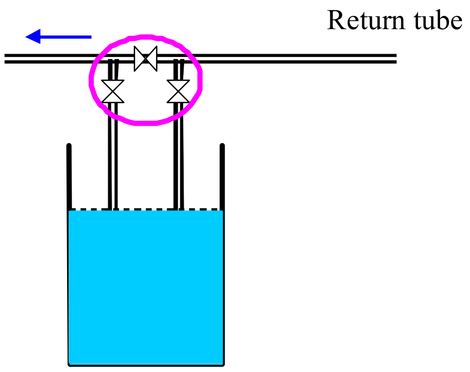


Figure 2.3.5. Principal sketch for indirect system at a heat pump and charging detail

2.4. System solutions for supermarket refrigeration (J. Arias)

2.4.1. Introduction

The purpose with refrigeration systems in supermarkets is to be able to store chilled and frozen food and make them available for the customers. Chilled food is best kept at temperatures of 2 to 8°C while frozen goods should be kept at about -18°C. The corresponding evaporation temperatures are usually about 15 degrees lower than the desired product temperature. A number of system solutions used today have emerged. Principal sketches are here shown for the following system solutions for supermarket refrigeration (Arias 2005):

- Direct system
- Completely indirect system
- Partially indirect system
- Cascade system with secondary fluid circuit
- System with heat recovery
- System with “floating condensing” (See 2.4.7)
- System with “floating condensing” and heat recovery
- System with “floating condensing” and heat pump

2.4.2. Direct system

The most common solution for refrigeration systems in grocery stores is the direct system, Figure 2.4.1. The refrigerant circulates from the machine room, where the compressors are placed, to the cooling cabinets in the store, where the refrigerant evaporates. Direct systems often use long tubes between the compressor, cooling cabinets and condenser on the roof. This can lead to very large refrigerant charges.

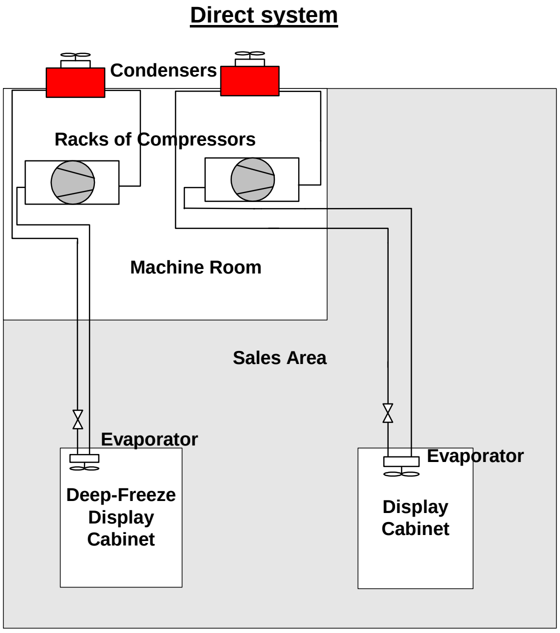


Figure 2.4.1. Direct system

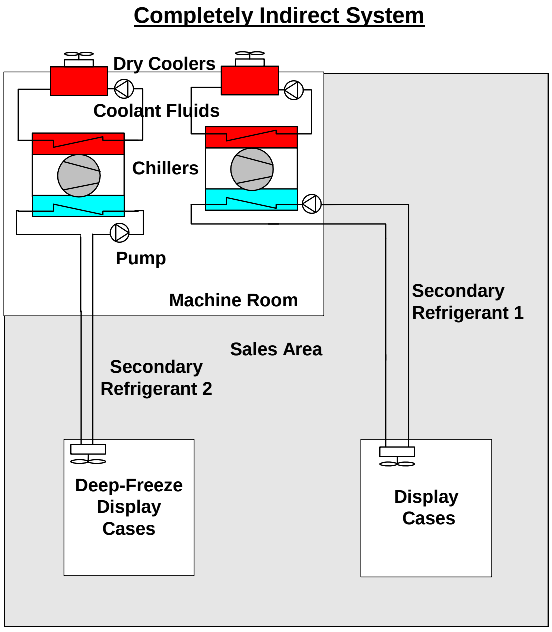


Figure 2.4.2. Completely indirect system

The most common type of direct system in a large store is the so called multiplex-system. Such a system consists of several compressors working in parallel and connected to a common tube, both on the low pressure side and on the high pressure side. The refrigerant charge in such a centralized refrigeration system is typically 4 – 5 kg/kW cooling capacity.

Another direct system often used has only one compressor connected to one or more separate cooling cabinets or freezers.

2.4.3. Completely indirect system

Indirect refrigeration systems have been introduced in supermarkets and grocery stores to decrease the refrigerant charge and to reduce the risk of refrigerant leakage. There are several forms of indirect systems. One of these is the “completely indirect systerm”. A principal scetch of that type of system is shown in Figure 2.4.2. In the example shown here there are two fully separate systems for cooling cabinets and freezers respectively. Different secondary fluids may be used due to the different temperature levels for cooling and freezing. In cabinets for chilled products there are typically temperature levels of the secondary fluid between -8°C and -4°C. Hence, the cooling machine has an incoming medium of -4°C and the temperature change is 4°C. The corresponding temperatures for freezers are -32°C and the secondary fluid is in the freezer cabinet heated to -29°C.

Various water-based secondary fluids with additives such as potassium formate, potassium acetate, glycols, alcohols or chlorides are used. For freezer applications it is of special interest to use CO_2 as secondary fluid as the phase change between liquid and gas cab be utilized in the freezer.

On the warm side there are also various system solutions with one or more refrigerant circuits to transport heat from the condenser(s) in the machine room to one or more secondary coolant chillers where heat is released to a heat sink.

If heat is released to outdoor air then it is advantageous to let the secondary coolant temperature vary with the outdoor air temperature – so called floating condensing (see further 2.4.7). A large part of the compressor work can be saved by utilizing the lowest possible condenser temperature. (However, too low temperature can not be used, in order for a satisfying functioning of conventional expansion valves in the refrigerating system).

2.4.4. Partly indirect system

The most common “partly indirect system” used in Sweden is what is illustrated in Figure 2.4.3. The condenser heat is released in the dry coolers placed on the roof of the store. For feezers a direct refrigeration system is used on the cold side while the cooling cabinets use an indirect system on both the cold and the warm side.

A variation on the partly indirect system is a distributed system on the warm side (Baxter 2003). It consists of several small refrigeration systems directly connected to the the cooling cabinets or freezers with liquid cooled condensers. The secondary fluid on the warm side is circulated between the condensers and (one or more) air coolers on the roof or for heat recovery.

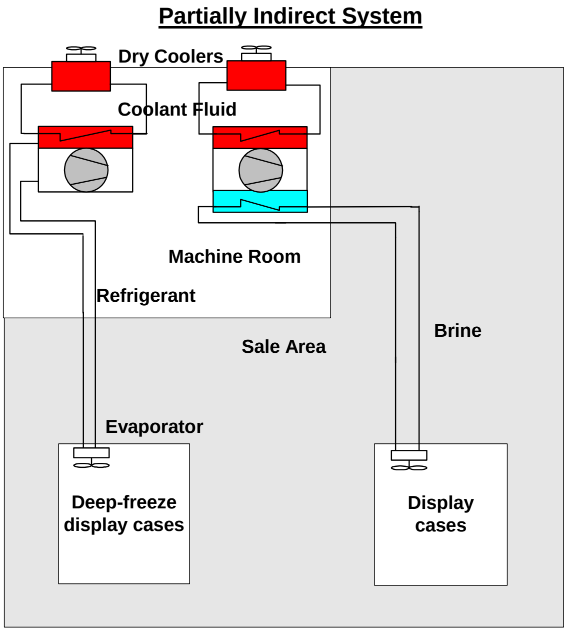


Figure 2.4.3. Partly indirect system

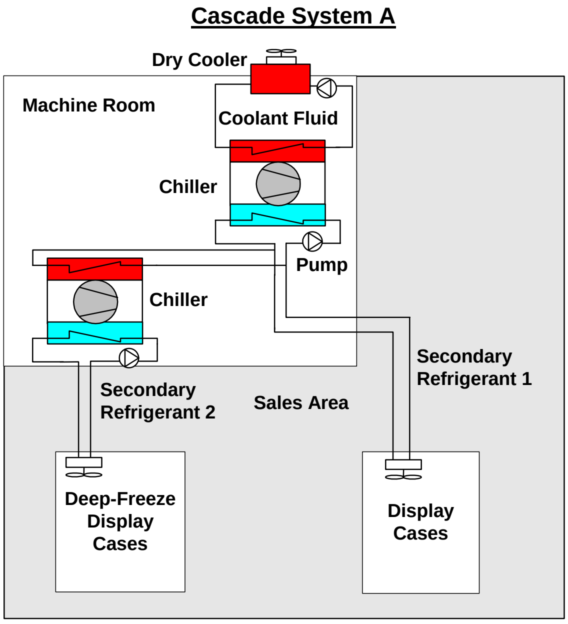


Figure 2.4.4. Cascade system with secondary fluids for cooling and freezing

2.4.5. Cascade systems with secondary fluid circuits

The cascade system, Figure 2.4.4, gives the advantage that it is possible to avoid the large pressure ratios in the freezer compressors that appear in a fully indirect system for freezer purposes (as in Figure 2.4.2). The cascade system works with two temperature levels and two secondary fluid circuits, ”1” and ”2”. Typical temperatures for the cooling cabinets that are cooled by secondary fluid 1 are -4 to -8°C. Secondary fluid 1 is also used to cool the condenser in the refrigeration system of the freezers. Secondary fluid 2 is used for the freezers where the temperatures usually are -32 to -29°C (same as in Figure 2.4.2).

The condensing temperature in the refrigeration system of the freezers can be about 0°C due to the low temperature of secondary fluid 1 cooling the condenser. This causes the coefficient of performance for the freezer system to be high. The drawback is that the power needed to cool secondary fluid 1 which increases the need of compressor power in the high temperature refrigeration system.

2.4.6. System with heat recovery

There are many possibilities for heat recovery – to heat the air in the store or adjacent space, to heat (or preheat) tap warm water, to heat water for a radiator system in the store (lokalen) or a local heating system for nearby houses (Compare. Figure 2.4.5).

Traditional refrigerating systems in grocery stores release large amounts of heat from the condensers to the surrounding but today most stores in Sweden use heat recovery from the condensers. Drawing benefit of heat from the condensers for heating of the store or the building also increases the total efficiency of the system.

A drawback with heat recovery systems is that they often require using a higher condensing temperature than other systems, increasing the energy demand of the compressors. However, when considering the whole system, including the benefit from the recovered heat, the total energy demand may be considerably reduced and a greater energy saving can be achieved.

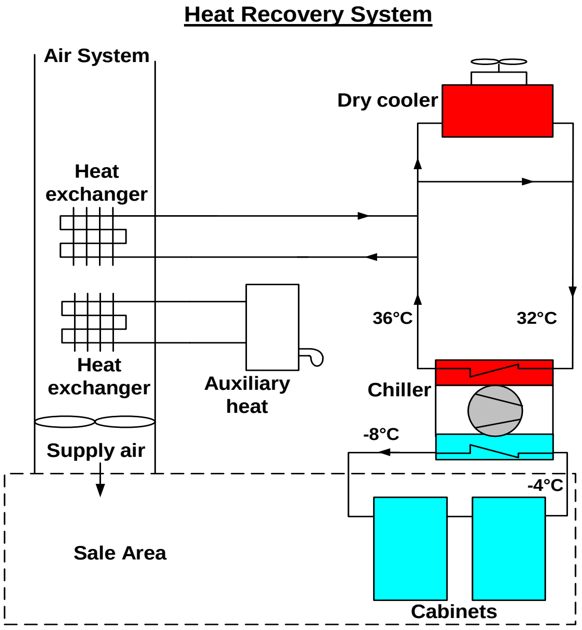


Figure 2.4.5. System with heat recovery

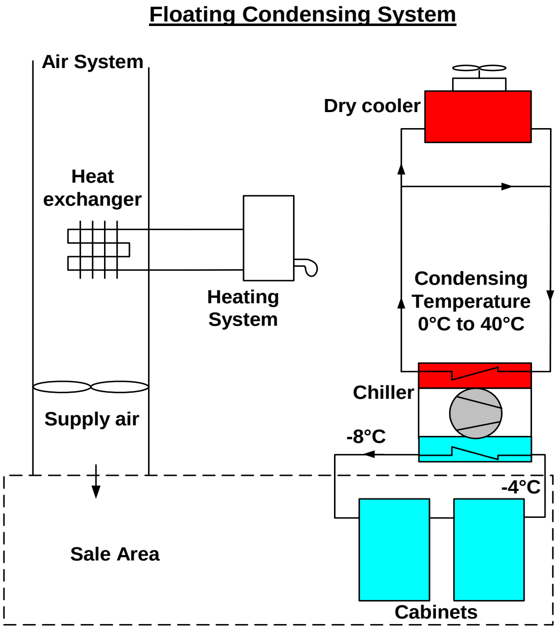


Figure 2.4.6. System with floating condensing

2.4.7. System with ”floating condensing” */

In a system with “floating condensing” the condensing temperature changed with the anbient temperature. At low anbient temperatures the condensings temperature can be considerably reduced. This decreases the energy used for the compressor which in turn increases the coefficient of performance. (Compare Figure 2.4.6).

*/ Systems with "floating" condensing are such where the condensation temperature is not controlled but permitted to be as low as the heat sink permits. This saves energy for compressor operation.

2.4.8. System with ”floating condensing” and heat recovery

A newer system with both “floating condensing” and heat recovery has been developed in Sweden in order to increase the energy efficiency in grocery stores. The need for refrigeration for cooling rooms and cabinets as well as the need of heating for the building are included. When the need of heating is reduced, one or more refrigerating units can be operated at lower condensing temperature. (Compare Figure 2.4.7).

2.4.9. System with ”floating condensing” and heat pump

A new system with both floating condensing and heat pump has also been developed in order to increase energy efficiency in stores. (Compare. Figure 2.4.8).

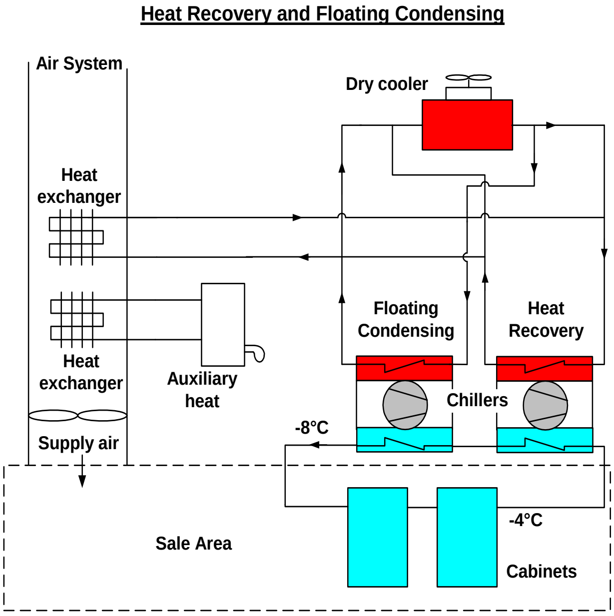
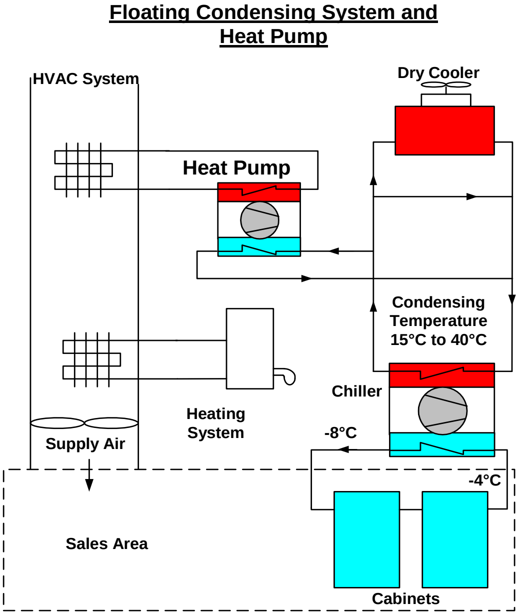


Figure 2.4.7. System with floating condensing and heat recovery



Figur 2.4.8. System with floating condensing and heat pump

2.5. Indirect system with CO_2 as working medium

(L. Rolfsman, E. Granryd)

2.5.1. Introduction

Carbondioxid, CO_2, as refrigerant [1] has long been used in refrigeration technology. It was already used from the end of the 19-th century until the 1940-ies. The interest for CO_2 was revived when CFC media were phased out during the late 1980-ies. The use of CO_2 as secondary fluid is of later date. An indirect system with carbondioxide, CO_2, as secondary fluid was installed in an ICA grocery store in Lund in the beginning of the 90-ies. Since then various types of secondary fluid systems with CO_2 have been built around the world. Only that type of indirect systems will here be dealt with. The use was from the beginning centered on temperatures suitable for storage of frozen grocery. These applications are still the most common but several other applications have been added since then. Today there are ice rinks with indirect CO_2 systems built in Sweden and other countries. Indirect systems with CO_2 at temperature levels suitable for storage of chilled grocery are common in new Danish stores and to some extent also in Norway. Systems with CO_2 for cooling of computer halls have started to be used in Great Britain.

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[1] Characteristic for CO_2 is that the temperature in the critical point is as low as 31°C and the pressure there is 74 bar. The saturation vapor pressure is also high, (35 bar at 0°C). The triple point is -56°C at 5,2 bar.

Due to the low temperature in the critical point the vapor does not condense at temperatures above 31°C – there is no distinction between vapour and liquid. For refrigerant cycles where heat is released at temperatures near the critical temperature the throttling loss in the expansion valve is comparatively large. The cycle is improved by maneuvering the expansion valve to increase the pressure on the warm side of the cycle considerably higher than the critical pressure.

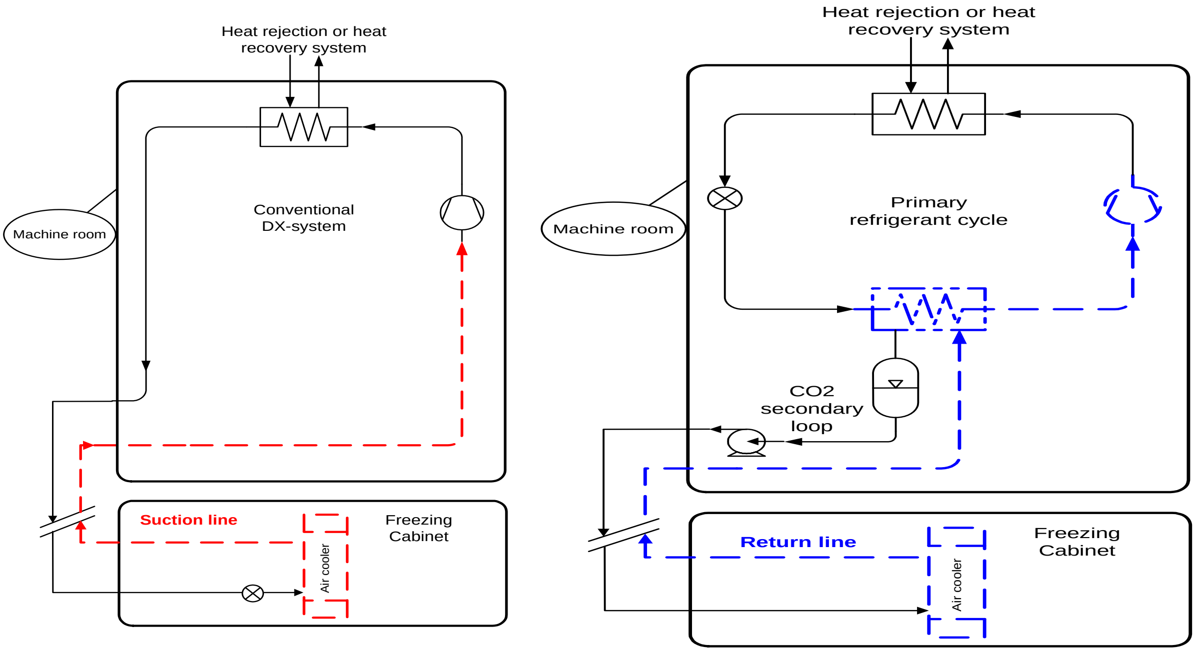
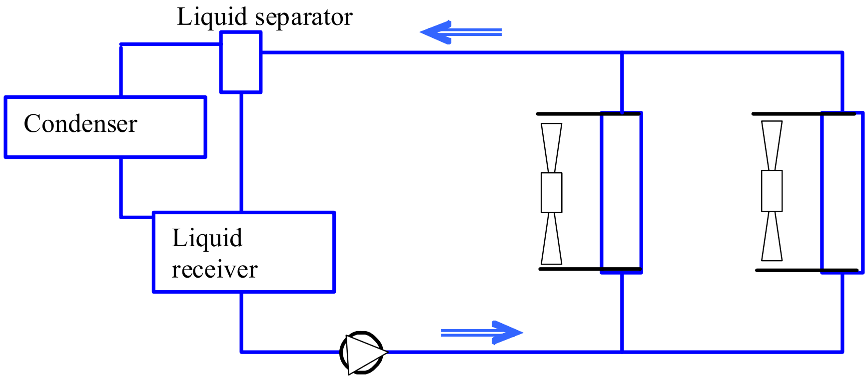


Figure 2.5.1. System with direct expansion (left) and indirect system with CO_2 as secondary fluid (right) (Sawalha).



Figur 2.5.2. Principal scheme for tube drawing at a two-phase circulation system with CO_2.

Liquid is pumped out to the various cooling objects where the carbon dioxide is evaporated. The amount of CO_2-liquid evaporated in each cooling objects depends on how large liquid flow that is used. In the return line there is two-phase flow of both liquid and gas. The vapor is led to the condenser and the liquid can directly be returned to the liquid tank through a simple liquid separator (not shown in the figure). The heat exchanger called ”condenser” in the figure is serving as evaporator in the primary refrigerant circuit.

The difference between an indirect system with single-phase secondary fluid and a system with CO_2 is that CO_2 is used as two-phase medium. From the liquid tank shown in Figure

2.5.2 CO_2 liquid is pumped out to each evaporator/cooling place where some of the liquid evaporates.

In the system it is the saturation pressure for the liquid, at the temperature in the liquid tank, that determines the pressure. Apart from the influence of pressure drop the temperature is the same in the various parts of the system. CO_2 liquid has considerably lower viscosity than all comparable 1-phase media.

For two-phase systems the flow is determined by the cooling demand as well as by the desire to reach boiling on all the surface of the evaporator. The amount of liquid circulated compared to the evaporated or boiled amount is given by the circulation number, *n,* that is the number of kg liquid circulated divided with the number of kg vapor formed). The choice of circulation number should be determined by an optimizing calculation (pressure drop – heat transfer). However, in practice, the surplus of liquid out from the evaporator can be chosen below 50% of the evaporated amount, corresponding to a circulation number, $n < 2$, For many smaller systems in stores the surplus is considerably higher mainly because sufficiently small liquid pumps have not been available.

2.5.2. Defrosting

Incomplete (wet) evaporation is an advantage from a defrosting view point. The constant temperature in the evaporator of the CO_2 side gives a more evenly distributed frost layer and can reduce the defrosting time. The frost growth can also be lower as the evaporation temperature can be kept somewhat higher.

For defrosting of CO_2 evaporators it is especially important that a “dry boiling time” is included in the operating sequence. The reason for this is to avoid big pressure variations in the systemet and great heat losses at the defrosting. The way to proceed is to close the liquid supply and let the fan continue to operate until the evaporator can be viewed as emptied of CO_2. A normal time for this can be 15 minutes. After that heat can be added for the defrosting.

Two different defrosting systems are used: One uses electric heat rods in the evaporator while the other uses a separate one-phase secondary fluid system with separate coils in the evaporator.

The size of the evaporator determines what system is best in each separate case. The secondary fluid used for defrosting can preferably be heated with heat from the condenser of the cooling system or from the oil cooler.

A normal construction temperature for the secondary fluid system for defrosting is +25°C. The supplier of the evaporator should verify dimensioning data.

Electric defrosting of large evaporators is unpractical, implying great need of electricity and is also sensitive to crack down in the electric rods.

Stop defrosting is possible for room temperatures >2°C. The liquid supply is then closed and the fan can continue to run.

2.5.3. Advantages

Systems with CO_2 give two special advantages.

Heat transport by use of the heat of vaporization gives considerably smaller flow in the secondary fluid circuit for a certain cooling demand.

In practise it means considerably smaller tube dimensions and less pumping power as well as a smaller temperature change.

Tube dimensioning

The diameter for the liquid tube can be chosen to give the velocity ≈ 1 m/s.

The return tube with 2-phase flow is most sensitive to dimension correctly even if CO_2 is less sensitive than other media. Liquid will be transported in vertical tubes at rather low velocities as the gas density is high.

For fully correct dimensioning it should for each plant be performed as an economical optimization with technical limitations. Gas velocities around 3-6 m/s can be useful Ffor smaller systems. Larger diameters can have larger velocities. Higher temperatures gives higher pressure and thereby higher gas density. Higher gas density makes liquid transport by dragging more effective.

Heat transfer

CO_2 has good heat transfer properties giving somewhat smaller temperature differences. The phase change means that the temperature change on the CO_2 side in the heat source exchanger is neglible. The smaller flow gives a smaller tube dimensions in the coils compared to single phase.

For ice rinks there is an advantage with more even ice temperature over the whole surface. However this difference may hardly be noticeable in practice. (Compare Figure 2.3.6).

No oil in the system

When choosing valves the supplier should be aware that there is no oil in the system. Certain common valves can stop functioning witout oil.

Having no oil is a great advantage if packed grocery would be prayed at a leakage. (For other systems, the grocery must in general be destroyed because of contamination if there is a leakage of common secondary fluids or of common refrigerants in direct expansion systems).

Price of media

CO_2 is considerably cheaper than both refrigerants and single phase secondary fluids. The latter both due to price and quantity. Different suppliers give various names of the quality to be used. Inform the supplier for what what the CO_2 shall be used. Today the supplier knows what an indirect system with CO_2 means.

The amount of water should especially be kept low. It must be possible in all condensers to remove non condensable gases.

Choice of refrigerant in the primary circuit

There is great freedom in choice of refrigerant in the primary circuit. A solution with indirect system makes it possible to use refrigerants such as ammonia, NH_3 och propane that for security reasons can not be used in a direct system.

2.5.4. Drawbacks

Pressure

What makes CO_2 different as refrigerant and even more different as secondary fluid is its vapor pressure curve. The pressure is 25 bars just below -10°C and 35 bars at 0°C. The whole system must be dimensioned to stand pressures that will occur. According to the coming version of EN 378 the constructer can choose suitable construction pressure (PS in PED). Suitable pressures are for freezing 25 bar (testing pressure 36 bar) and for cooling 35 bar (testing pressure 50 bar).

Earlier it was difficult to find components on the market but not today.

A special problem with CO_2 is to handle what can happen at an operating stand still when the temperature in the system can rise. If the temperature increases the pressure in the system rises and there is risk that the security valve opens and the charge is lost.

For freezer installations with freezer rooms the coils can serve as storage vollumes for liquid at the low temperature. The coils should be fed from the bottom and have magnetic valves normally closed when there is no operating signal. The exit connections from the coils should be open so that evaporated gas from the liquid receiver can condense if the plant of some reason is out of operation (temporally). This will keep the pressure down until the coils are filled with liquid.

The same principle can be used for cooling rooms if storage temperatures are low enough to correspond to a pressure below the construction pressure (PS)

Some active principle has to be choesn in all other cases. A small glycol accumulator that during normal operation was cooled by the suction line was built at the first plant in Lund in the south of Sweden. The container could condense the CO_2-charge during power failure.

Other possibilities are for instance:

- controlled blow off at a pressure just below the highest premitted

- a small refrigeration unit with separate feeding from an auxiliary generator or petrol engine.

- the first plant in Lund was equipped with a small glycol accumulator that during normal operation was cooled by the suction line and thereby could condense the charge during power failure.

Two-phase flow

Many who construct single phase systems lack knowledge of two phase flow in tubes. The constructer needs an understanding of two phase flow.

Leakage

The leak rate is higher for gases than for liquids. The system needs to be built as a usual refrigerant system. The choice of materials is no problem especially if there is no oil in the system.

A warning should be given for leakage of CO_2 into ammonia systems. A white water soluble powder is formed that is disintergrated at +60°C. It can clog the oil system in ammonia compressors. No really good warning device is available.

Refill valves in CO_2/ammonia systems should especially be clearly marked and the charging should be done by competent persons.

Humidity

Water in CO_2 systems is different than in other refrigerant systems. Acid is formed that start corrosion and probably also copper plating. In a CO_2 secondary system driers should be instaled in a part flow circuit to the pump (for instance in the circuit used for low capacity operation). The humidity will freeze out in the evaporator due to the lower solubolity in gas phase.

The drying agent shall be meant for CO_2. Certain zeolites can dissolve and be devastating in the system.

Personal security

It is dangerous to breathe in high concentrations of CO_2. The border value for continous work is 5000 ppm. At values above 50000 ppm (5%) sensitive persons can be seriously affected. In small rooms it is advisable to have CO_2 warners.

Knowledge

During the construction of indirect CO_2 systems a partly different kind of expertise is required compared to work with single phase system, similar to the type of knowledge required at common refrigerant systems.

Energy usage

At a comparison of energy usage for an indirect system with a direct (DX) system the indirect system is generally expected to have higher energy consumption due to the temperature difference in the heat exchanger connecting the primary and secondary circuits. This ought to cause somewhat lower evaporation temperature and coefficient of performance, COP_2. However, CO2 has very favorable heat transfer characteristics and the work to run the CO_2 pump need only be 1-2% of the primary circuit compressor work. The low pressure drop with CO_2 and the low volume flow also contribute to minimizing the energy consumption. Calculations show that the total energy consumption is equal to or somewhat lower than a DX-system with R404A.

(ASHRAE Refrigeration 2006, Sawalha 2008).

2.6. Definitions and classifications of system types

Refrigeration plants are grouped in various classes depending on the method used to take up heat from the medium to be cooled, i. e. groceries. For the direct and the fully indirect or partly indirect systems the following basis for classification may be used. (This classification follows the references: Svensk Kylnorm 1988, Ekman 2003).

2.6.1. Direct system

Direct systems (DX) are systems where both evaporator and condenser without any in between medium (secondary fluid) are in direct contact with the medium to be cooled and heated respectively.

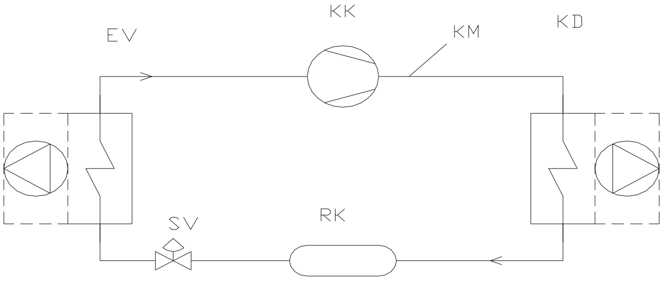


Figure 2.6.1. Principal sketch of direct system

2.6.2. Indirect system

Indirect systems (IS) are systems where either evaporator or condenser (or both) are not in direct contact with the medium to be cooled.

Fully indirect system (FINS)

Fully indirect systems refer to systems where both evaporator and condenser condenser are connected to the source and sink by an 'in between medium' (a secondary fluid) for heat exchange.

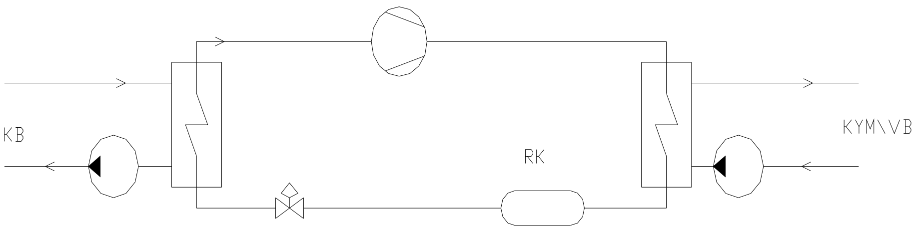


Figure 2.6.2. Principal sketch of fully indirect system

Partly indirect system (PINS)

Partly indirect systems refer to systems where either evaporator or condenser are connected to the source or sink by a secondary fluid for heat exchange.

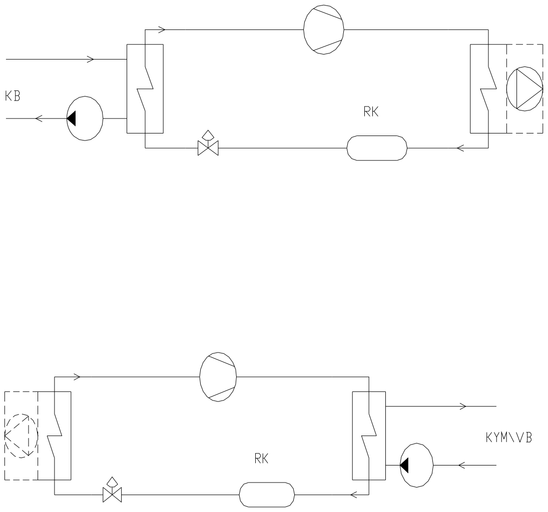


Figure 2.6.3. Principal sketch of partly indirect system

Secondary fluid systems are classified according to below mentioned groups.

Open indirect system

A system in which the evaporator cools a secondary fluid that, in a spray chamber or similar arrangement, has direct contact with the medium to be cooled.

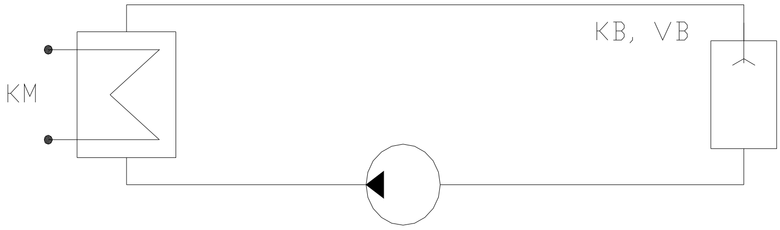


Figure 2.6.4. Principal sketch of open indirect system

In a ventilated open indirect system the evaporator is placed in an open vessel or in a ventilated container, causing a corresponding effect.

Closed indirect system

System where the evaporator cools a secondary fluid that is in heat exchange with the medium to be cooled though a closed system.

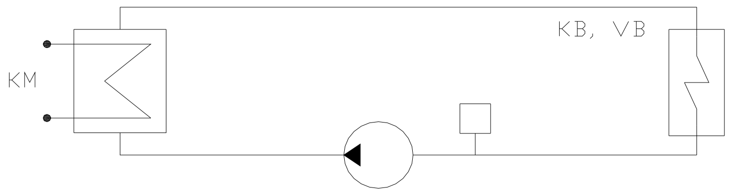


Figure 2.6.5. Principal sketch of closed indirect system

In a ventilated closed indirect system the heat exchanger is placed in an open vessel or in a ventilated container, causing a corresponding effect.

Double indirect system

System similar to a *ventilated closed indirect system*, besides that the cooled secondary fluid also passes a second heat exchanger. This is then placed in a corresponding way as the heat exchanger for a *indirect system* and cools a second secondary fluid circuit in which a spray chamber or similar has direct contact with the medium to be cooled.

2.7. Other possible system types

Basic definitions are given in Ekman 2003 and Svensk Kylnorm 1988 after which some of the most common system connections for the secondary fluid side are accounted for. The two different main types are systems with constant or varying flow in the secondary fluid circuit, respectively. The classification is also divided for cases with one or more units in the circuit. However there is no claim made for the classification to be complete as various system variations can be possible. The list below covers most of possible standard solutions.

- Two circuit system with varying flow and accumulator - One unit.
- Two circuit system with varying flow and accumulator - Several units.
- One circuit system with constant flow - One unit.
- One circuit system with constant flow - Several units.
- Cooling object
- Variable flow in the primary circuit (two circuit systems)
- ON/OFF regulated with two way valve (cooling coils, cooling cabinets, etc.)
- Modulated two way valve (cooling coils, cooling cabinets, freezers, heat exchangers)

- Constant flow in the primary circuit (one circuit system)
- ON/OFF regulation with three ways valve alt. double valves (cooling coils, etc.)
- Modulated three ways valves (cooling coils, freezer condensers, heat exchangers)
- System without heat recovery
- System with heat recovery
- System design for the classification
- Open resp. closed cooling tower – Secondary fluid condenser
- Closed recooler - Secondary fluid condenser
- System with heat recovery
- Open cooling tower - Secondary fluid circuit with condenser and heat exchanger for heat recovery
- Closed cooling tower - Secondary fluid circuit with condenser connected to the heating system
- Open cooling tower – Double condenser (Secondary fluid /Heat recovery)
- Closed cooling tower - Double condenser (Secondary fluid /Heat recovery)
- Closed recooler - Double condenser (Secondary fluid /Heat recovery)

(Ekman 2003, Svensk Kylnorm 1988).

3. Components in indirect systems and their function

3.1. Various types of components

3.1.1. Introduction

An indirect secondary fluid system contains all or several of the following main components: circulation pumps, heat exchangers (evaporators and in cooling units), air purging devices, expansion vessels, defrosting equipment.

A number of the following components can also be found in an indirect system: shaft sealings, filters, valves, gaskets, joints, splices, valve sealings, tubes or pipes for secondary fluid system, drip plates, measuring devices, manometers, etc.

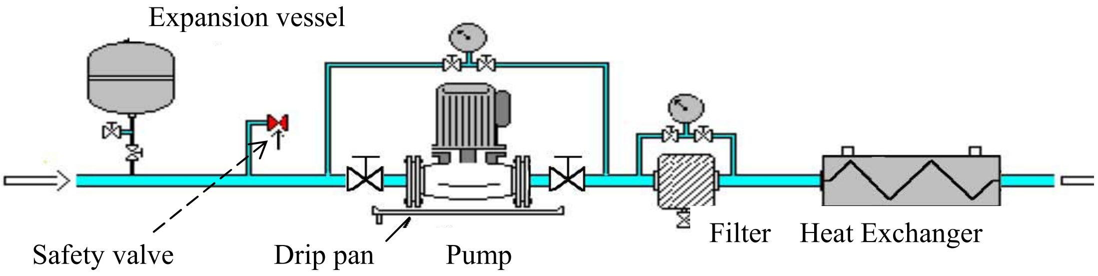


Figure 3.1.1. Some of the components in a secondary fluid system

Placing of the various components isconsidered as well as what is important to think of in order for the various components proper functioning in the system. The pump unit is a central part of a secondary fluid system and will be considered first of the components. Air purgingdevices and expansions vessels are also treated more extensively. Heat exchangers and most other mentioned components are dealt with to some extent. (KBL 2000)

3.1.2. Placing of components

Expansion vessels should be mounted at the suction side of the pump and liquid filters on the pesssure side of the pump. Pumps should be equipped with closing valves and fully cover drip plates.

A sufficiently high static pressure should be ensured on the suction side of the pump in order to prevent gas bubbles to appear in the pump.

Micro bubble air purgers and vacuum degasers are mounted in the return pipe. High poinkt air purgers are mounted on the high points in the system and are equipped with closing valve in the system.

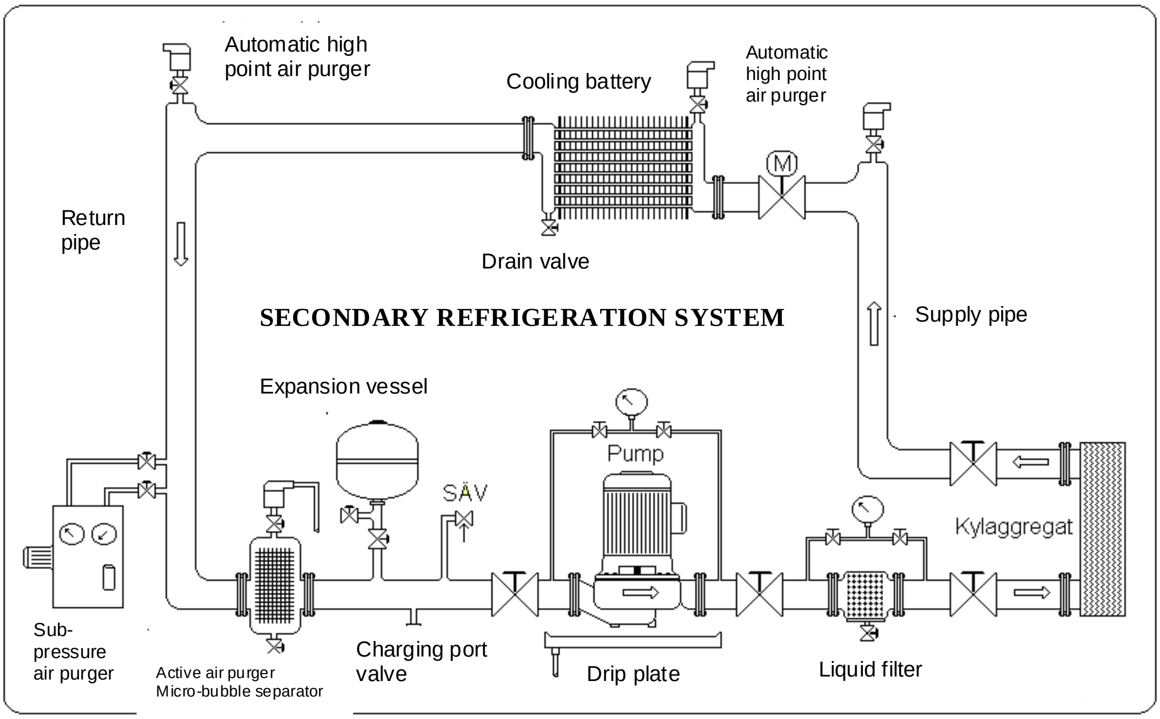


Figure 3.1.2. Placement of components in an indirect refrigerating system

3.1.3. Function of components – important to consider

Pumps - check regularily that the pump with fully covering drip plate and splash protection functions well. Rinse at once away possible spill.

Air purger devices - check regularily that the device functions well and does not leak as this is important for satisfying operation. Air cause poorer heat transfer and lower pump capacity in the system and it increases the risk for corrosion and leakage. Air purging valves shall, at intervals, be dismanteled ad cleansed, preferably with water. Air purging is most effecient when the system is warm. [Se även Köldbärarlaget tidigare publicering.]

Expansion vessel pre-pressure - should be checked to ensure that the pump has suffiently high static pressure. Make sure that the device has correctly set pre-pressure in pressure isolated state. The equipment should of course be free of leakage. The pressure of the system should be checked regularily.

Valves - check regularily that the valves function and are tight. Rinse away possible spill instantly.

Filter – place filters at the pressured side of the pump. Choose suitable mask size and clean the filter regularily.

Measuring device, manometers, etc. – Check that the equipment functions and is free of leakage.

(KBL2 2000, KBL4 2003)

3.2. Pump units in secondary fluid systems

3.2.1. Introduction

As circulation pump units are a central part of an indirect secondary fluid system they will be considered first and most extensively of the components. Various pump types are first treated followed by dimensioningor of pumping power and pressure drop i the system.

Cirkulation pumps are today more and more used in refrigeration, due to the change to indirect systems. Water based secondary fluid are largely used in this type of systems. Density and viskosity of these secondary fluids have a great impact for the dimensioning of the pumps. Wrongly dimensioned pumps can cause extra great energy consumption and create other problems in the system.

High density causes the energy consumption of the pump at the dimensioning point to increase in proportion to the higher densitety. Pump diagrams that usually are given in catalogues from the pump manufacturers are based on water at +20°C.

High viscosity causes the liquid to become more viscous and impairs the hydraulic efficiency of the pump. At high viskosity there is also a need to correct for flow and head pressure. Experience from heat systems where water usually is pumped need to be increased with knowledge about secondary fluids to be able to dimension pumps in these systems.

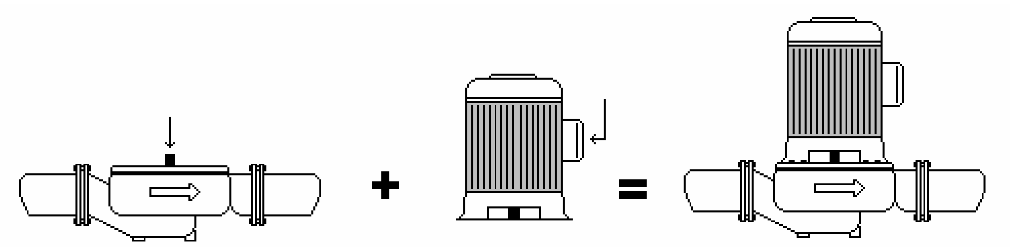
Flow and pressure drop must first be calculated in the system. The pump can then be dimensioned from these values. Dependent of type of secondary fluid chosen and these specific thermodynamic properties these values will be the basis for correcting flows and pressures for determining choice of pump. Hence, the pump should be dimensioned after secondary fluids chosen. (Bredberg 2001)

3.2.2. Types of pumps in secondary fluid systems

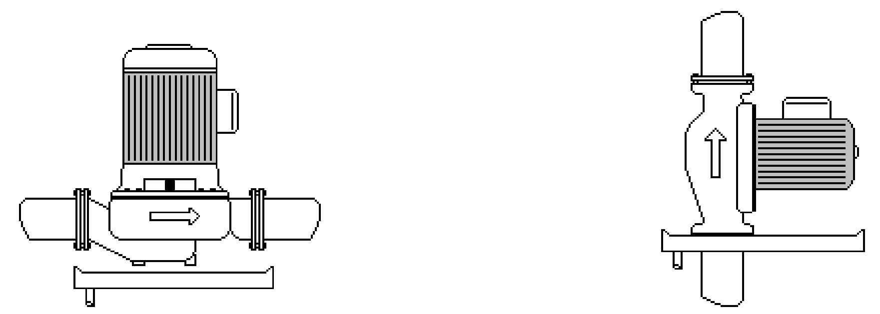
A pump with mounted power source (motor) is a pump unit. In daily speech we denote a pump unit as a pump. The power of the pump unit is the sum of the power to the power source (motor), P_1, and the power for the pumping, P_2.

Two main types of pumps, dry and wet pumps, are used at indirect secondary fluid systems. The difference between these pump types is whether the motor of the pump is air cooled or cooled by the fluid it shall pump. Dry pumps have the advantage of not adding unnecessary heat energy to the fluid that then has to be cooled off by the refrigerating unit. Wet pumps however have the advantage of not needing any shaft sealing and its risk of leakage. But they can not be used at operating temperatures below -10ºC.

A variation is so called magnet driven pumps, where the transmission between the motor and the pump takes place with magnets which gives a dry pump without shaft sealing. This type of pumps is more expensive but an excellent alternativ especially in the lower temperature range.



Pump types (dry resp. wet)



Dry pumps (motor air cooled) Wet pumps (motor cooled by the liquid)

Figure 3.2.1. Dry and wet pump types

In the dry pump the shaft lead through is made water tight with a shaft sealing, which is one of the critical parts of the pump. The shaft sealing is a detail that wears out in the pump and demands regular control and possible replacement. How often depends on operation conditions and outer influence. Normally a shaft sealing should have a small leakage, depending on its construction. A wet pump does not have this type of shaft sealing. (Bredberg 2001, Moberg 2002)

3.2.3. Circulation pumps withd speed regulation in secondary systems

Circulation pumps with continous speed regulation are used more and more. Pumps with dry (fan cooled) motor should be chosen. Pumps of column tube type (wet) should not be used in larger plants due to unfavorable heat addition to the secondary fluid resulting lower efficiency.

Speed-controlled pumps improve technical, operational and economical properties of the plant. Installation, operation and maintenance are simplified.

Valve design in the system can be reduced and simplified during construction. Start up time is reduced as the pump continously can be adjusted to system data. As the capacity of the pump continuously adapts to the momentary needs of the plant and are supervised electronically, the plant becomes more secure and the operation more simple.

Speed-controlled pumps improve the system also from operational economic and environmental concern. The added cost for a speed-controlled pump is neglible in relation to the savings in operational costs. The savings in valve design quickly compensates the added cost for this type of pump. (KBL2 2000)

3.2.4. Placement of pumps

In order to achieve largest possible static inlet pressure and more secure deairing, it is recommended that the pumps are mounted on a low level in the plant.

When there is a need for a reserve pump the pumps in a secondary fluid system under 0°C should be of single design. Twin pumps shall then not be used.

The circulation over a refrigeration unit (the evaporator) should be constant, which means that a charging lamp is mounted that only circulates over it.

The system is in this way divided up in two parts; a loading circuit and a distribution circuit.

At the dimensioning of circulation pumps in secondary fluid systems one needs to consider that the pumping curves in the catalogues of the pump manufacturers are given for pumping of water at +20°C. When pumping of any other medium than water the pump curve should be corrected.

Dimensioning of pump and mounting of correct type of shaft sealing in the pump can be performed by the pump manufactorers if the following information is given:

Head pressure, volume flow, type of secondary fluid, concentration, operation temperature, ambient temperature as well as if the pump should get surface treatment (finish), which is recommended in secondary fluid systems.

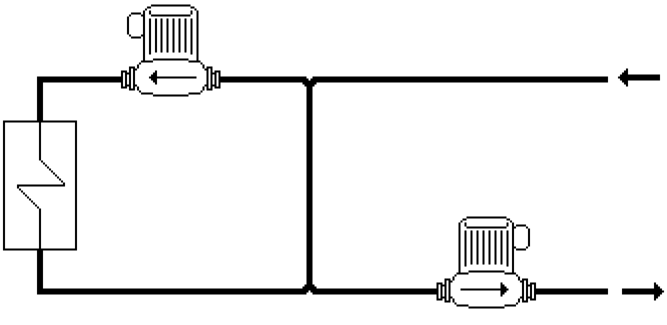


Figure 3.2.2. Charging and distribution circuits

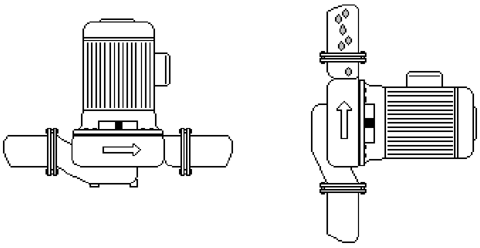


Figure 3.2.3. Placement of pumps

Mounting of circulation pumps in secondary fluid systems should be done with minimizing the risk of air pockets in the pump in view. One way is to mount the pumps in a rising tube as air bubbles do not easily gather in the space around the shaft sealing in the pump causing break down due to poor sealings. Circulation pumps should not be mounted in the high points of the system. (KBL2 2000)

3.2.5. Pump dimensioning

Pump curves in the pump manufacturers catalogues are given for pumping of water at +20°C temperature. When pumping any other medium than water the pump curve should be corrected. When pumping of any other medium than water the pump curve (operating point) should be corrected and consideration should be taken of any change in power consumption. Attention should also be given to the density and viscosity at the operation temperature.

Dimensioning of pump is carried out by the pump manufacturer on the basis of given information.

Desired volume flow, *Q*, and head pressure, *H*, should be chosen within the area of operation of the pump. When dimensioning outside normal area of operation the risk of cavitation and the formation of steam (vapour).

A pressure curve, NPSH, shows the lowest needed opening pressure to the suction side of the pump so that the liquid will not evaporate. To prevent the formation of steam and deposit of gas the minimum pressure of the suctionside of the pump should be secured.

When inquiring about pump dimensioning the follow values should be given:

Head pressure	kPa eller mvp
Volume flow	kg/s, 1/s eller m^3/h
Type of secondary fluid	% eller fryspunkt °C
Density	kg/m^3
Viscosity	mPas, cP eller cSt
Operation temperature	°C
Ambient temperature	°C
Surface treatment	Type

To think of when inquiring about pump dimensioning:

Specified value should be compensated for chosen media and system resistance.

Specified value should be compensated for chosen medium in the system.

Specify freezing point or concentration of chosen medium.

Specify density at the operation temperature.

Specify viscosity at the operation temperature.

Specify lowest operation temperature.

Specify ambient temperature for the pump motor.

Specify how the pump shall be surface treated.

Dimensioning of pump and type of shaft sealing after the above given tasks are performed by the utförs av pump manufacturer. (KBL1 1999)

3.2.6. Shaft sealing at secondary fluid pumps

Introduction

A mechanical shaft sealing consists of a rotating and a fixed, stationary sealing part (this sealed surface are so tight against each other that the slot in between gets a molecular dimension). This means that a few drops of the secondary fluid ooze or leak through the slot every hour, but this function is needed as the liquid froms a lubricating film between the sealing surfaces and is necessary to lessen the friction and heat generation between the sealing surfaces.

Normal and abnormal leakage at shaft sealings

A mechanical shaft sealing always leaks a little. Whe the lubricating liquid comes out of the shaft sealing the water evaporates. During pumping of i.e. glycol water solution the water evaporates, while the glycol can be seen as leakage on the outside of the pump. When the secondary fluid is a salt it is the salt crystals that are left on the outside of the shaft sealing, which can be viewed as a fault, but is fully normal.

Abnormal leakage appears when the shaft sealing surfaces are damaged. As an example it has been confirmed that copper, as a result of a galvanic process, can come out on the rotating part and form small "copper islands" that create unevenness in the sealing surface causing larger leakage. Hydraulic lack of balance and pressure blows can also cause damage on the sealing. It is veryimportant that the right material is used in the sealing surfaces and carbon graphite impregnated with various bonding agents has proved to be suitable on the rotating part and silicon carbide has proved to be advantageous on the fixed part. An alternative to carbon graphite that is an electrically conductive material, is glass fiber armed Teflon. This material is not electrically conductive and does not attract copper iones in the same way as carbon graphite.

Type and material of shaft sealing should be chosen considering the type of secondary fluid used. The pump manufacturers have great knowledge of this and can help the designer when choosing pump. It can be appropriate to consult with the pump manufacturer about suitable choice of sealing for chosen secondary fluid. There are shaft sealings adjusted for various types of secondary fluids.

At low temperatures condense appears on the pump house. Drip plate should always be mounted in secondary fluid systems. (Bredberg 2001, Moberg 2002)

3.3. Heat exchangers

3.3.1. Various types of heat exchangers

A number of types of heat exchangers are used in indirect systems. On the warm side in e.g. a grocery store secondary fluid coolers cooling the condenser (Comp. Figure 2.4.2 – 2.4.8). The transfer between the air and the secondary fluid usually takes place with so called fan air coolers, where the air flows between finned cold tube coils. The evaporator has as task to transfer haet from the secondary fluid on the cold side to the refrigerant during evaporation. Hence, it is also called liquid cooler.

Liquid coolers of various types are used to cool the secondary fluid on the cold side. Since long shell and tube evaporators of vaious sizes have been used and for small sizes coaxial evaporators tht seldom are used today.

In later years these types of evaporators have largely been replaced with very compact plate heat exchangers. The secondary fluid flows on one side of the plates and the refrigerant boils on the other (See Figure 3.3.1). (Granryd 2005)

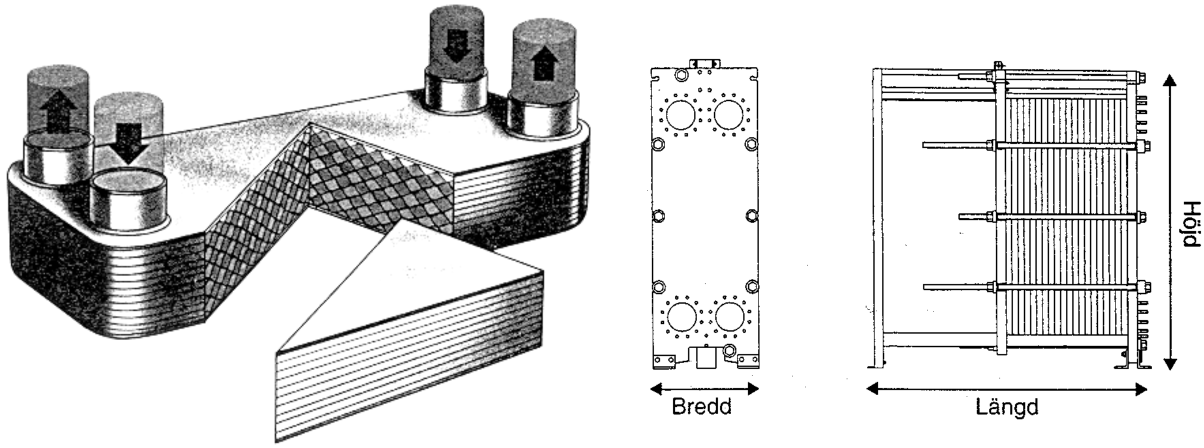


Figure 3.3.1. Compact plate heat exchangers (Bredd= Width; Längd = Length; Höjd = Height)

The advantage with compact plate heat exchangers are evident from the way a villa heat pump looks today. Figure 3.3.2 shows such a unit with a compressor (black) and a small secondary fluid pump (red-black) and both refrigerant and secondary fluid tubes. The compact plate heat exchangers on both condenser (behind the compressor) and evaporator (to the right) are small and contributes to a compact unit that (surrounded by a thin) easily can be placed in a villa.

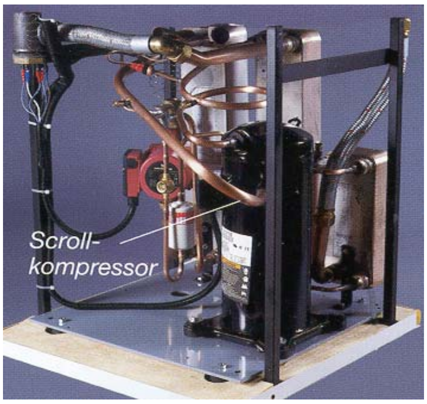


Figure 3.3.2. Villa heat pump unit

3.3.2. Cleaning of plate heat exchangers

The complex channel structure of plate heat exchangers creates heavy turbulence already at low flows. If the water is extremely hard a deposit layer can appear if also the temperature is high.

Plate heat exchangers that can be opened can be cleaned manually. A total control of all the heat exchanger parts can at the same time be made. The plate heat exchanger is opened and the sealings are removed from the plates. Depending on the type of fouling various agents of cleaning are used.

Soldered plate heat exchangers can be cleaned by circulating a weak acid solution, for instance 5% phosphoric acid. During regular cleaning 5% oxalic acid can in stead be used. For the cleaning to be effective the heat exchanger should be back washed and the flow should be 1.5 times higher than the normal operating flow. After the cleaning the heat exchanger shal be washed with plenty of water to remove any remaining acid. If the media contains particles larger than 1 mm the heat exchanger should be protected with filter using a gauge size of about 1 mm. (AMA-nytt 1999)

3.4. Air purging in secondary fluid systems

3.4.1. Air in secondary fluid systems (KBL2 2000)

The corrosion triangle

Air consists of: 78% nitrogen gas N_2; 21% oxygen gas O_2 and about 1% noble (inert) gases, incl. argon. As air consists of various gases the term gas (rather than air) ought to be used. But, for simplicity we use the term air and air purging or de-aeration in this publication.

Various types of problems can appear in poorly air purged systems. For instance reduced pump capacity, reduced heat transfer, regulation problems, erosion and corrosion och leakage at seals. Of these corrosion causes the most serious problems.

Corrosion in the systems is manily caused by oxygen in the air. Since oxygen is one of the elements essential for the corrosion process, corrosion increases in poorly air purged systems.

Oxygen is one of three important factors in the so called corrosion triangle that contributes to corrosion. The other factors besides oxygen are electric conductivity and materials that are corrosive in the system.

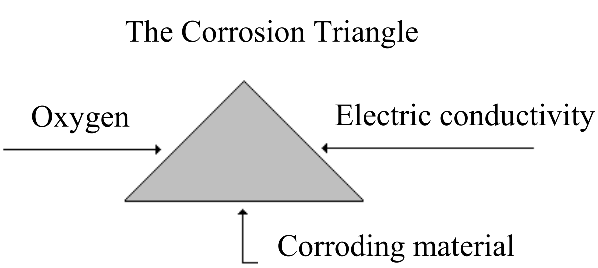


Figure 3.4.1. The Corrosion Triangle

Air in different forms in a system

Air may in a system occur as free air (air pockets, air bubbles) and as bound air (dissolved in the liquid). It is easier to remove the fre air than the bound air in the liquid. It is important to keep the oxygen concentration in the secondary fluid on a low level. The amount of air that a liquid can bind depends mainly on pressure and temperature. A high temperature and low pressure cause the liquid to bind less air. A low temperature and high pressure causes the liquid to bind more (AMA-nytt 1999, KBL2 2000).

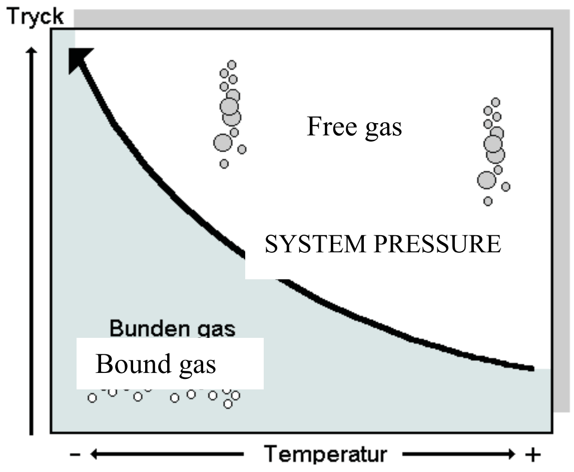


Figure 3.4.2. Ability of liquids to bind gas.

Different ways of removing air

Various ways of removing air from liquids are chemical, physical och thermal air purging:

In chemical air purging, various substances are used to bind oxygen (for example hydrazine, sodium sulphite, etc). Some of these substances are harmful to health and some are banned.

When using these binding agents it is important to check their concentration in the heat transfer fluid, to avoid the opposite effect or other problems.

In physical air purging, Henry's Law is applied. This law states that the solubility of gases in liquids decreases with decreasing pressure. The pressure reduction causes bound air to move into the free gas zone from which it can be released.

Thermal air purging takes place first at 30 °C and above, with bound air becoming free air.

Other factors affecting air purging

Low surface tension of the liquid increases the risk of foaming. High viscosity causes the air to rise more slowly in liquids. Pressure and temperature affect the solubility of air and therefore the air purging (Henry's Law). It is much more difficult to remove the air in a system at low than at high temperatures. The density of the liquid does not affect the solubility of the air in the liquid.

Recommendations for air purging

Circulate the system at high temperature (room temperature) for a while after filling it. Air purging is speeded up at high temperature. Be careful to get out all air to avoid future operation disturbances. Protective glasses should be used during air purging. (KBL2 2000)

3.4.2. Air purging and the functioning of various air purgers (KBL1 1999)

Introduction

Various methods of air purging are covered in the figure below. At systems with med water based anti freeze agents it is very important to purge the system completely from air. Air pockets or air present (oxygen) increases the risk for corrosion and break down or consumption of corrosion inhibitors. When using automatic air purgers, a valve that can be closed when the system is air purged or need tobe cleaned, should be mounted between this and the system.

1. Manual top purger
2. Automatic purger
3. Active deareator
4. Sub atmospheric deareator
5. Pump

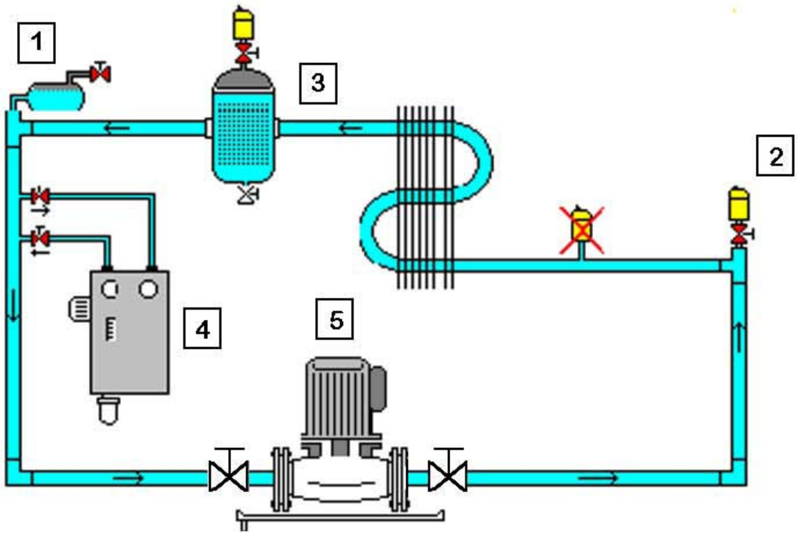


Figure 3.4.3. Various methods for air purging (KBL1)

What type of air purger and where it should be placed in the system depends on the design and size of the system. To be able to choose right type (method) and to mount it at the correct place in the system there is need to know how the various methods function.

Installation of air purgers

In order to obtail a well functioning plant is is important that the air in the system can be separated from the media in suitable points.

De-aeration valves (automatic or manual) are in view of this mounted on the tube system (preferably after each object) in "high points" at a 90°-curve. A T-shaped piece is mounted replacing the 90°-curve where one of the tube pieces in which the de-aeration valve is mounted is directed upward. This means that this tube will function as a volume chamber for collecting air that will be taken care of by the de-aeration valve. As an alternative this mounting can be made on a horisontal straight tube (See figure 3.4.4.).

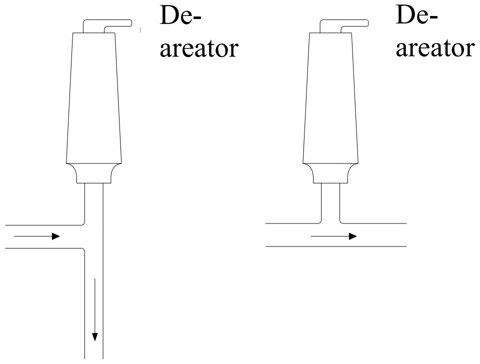


Figure 3.4.4. Installation of deareator valves

Some kind of central way of air purging should besides the above mentioned be mounted in the plant. This should be some kind of standard solution available on the market, for instance Spirovent. See installation guidelines from each supplier.

One way to manually deaerate a system centrally is to draw a part to the mixing vessel during a certain time (≈ 20% of the total flow). Any air in the system will bubble up to the surface. Secondary fluid drawn ill then returned to the main system with the aid of a pump. Drawing and returning of the secondary fluid should be done on the suction side of the secondary fluid pump.

Control and possible manual separation of air in the system should take place with even intervals. It is especially important to deaerate after interference in the system or after recharging secondary fluid.

High-point air purgers

Free air bubbles that circulate in the system are collected in certain high points when the circulation stops and can then be removed with automatic air purgers. When air is gathered under the de-aerators the liquid is pushed aside and this is registered by the float of the de-aerator. When the liquid level has reached a certain level the de-aerator opens and let air out to the surrounding.

As the name implies, this type of air purger must be installed at high points in the system. Their operation is based on the fact that free air rises. During charging of the system, the air rises to the high points. When the system has been charged and is in use, it is difficult for the free air bubbles to leave the liquid naturally. The air bubbles circulate with the liquid until the circulation stops. Only then can the bubbles rise to the high points.

De-aeration can take place manually with a ball valve. Manual air purgers involve a lot of work, since they have to be opened and closed manually several times, and there is a considerable risk of poor air purging. When choosing high point de-aerators you should choose the automatic type. In these there is a float which opens and closes the air purger automatically. This means that there is little work involved and good air purging is ensured.

After some time, automatic high point de-aerators (float valves) tend to leak because of various factors, such as dirt in the valve and deposits on the float. This leakage may cause major problems outside the system. In addition, leaky air purgers sometimes allow air to enter the system. To overcome this problem, a shut-off valve can be fitted between the system and the high point air purger. (See Figure 3.4.6).

You may ask what is the use of a high-point de-aerator when it is isolated from the system? As already explained, it is difficult for air bubbles to leave a circulating liquid naturally. Consequently, a high point air purger (float valve) is not much use when the liquid is circulating, and should normally be closed with the shut-off valve mentioned above.

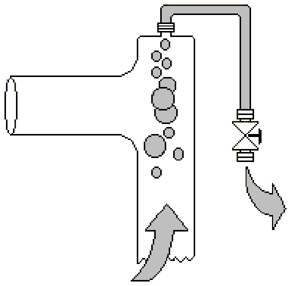


Figure 3.4.5. Manual highpoint air purger

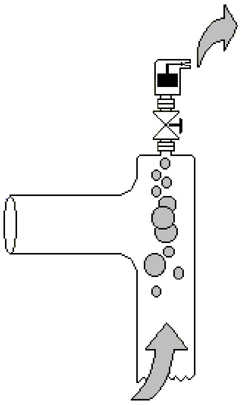


Figure 3.4.6. Automatic high point air purger

Start and stop of circulation pump is a method used for de-aeration. When air purging is required, the circulation pump is stopped to allow the air to rise to the high points. The shut-off valves are opened when there is a need for high point air purging, for instance during servicing. We have now achieved two things: good high point air purging and a non-leaking valve.

Active air purger/micro-bubble separator

When the system operates air bubbles can be caught by a micro-bubble separator (See Figure 3.4.7.) consisting of a presurized vessel containing a net that in combination with the surface enlargement reduces the mean velocity of the fluid and catches the air bubbles. The bubbles then rise to the top of the presurized vessel where an automatic air separator releases the collected air. Också i detta fall bör A manual closing valve should also in this case be installed before the automatic air separator. This type of air purger must be installed where the pressure is lowest and the temperature highest in the system. A suitable location is in the return line from the refrigerated units/objects but before the expansion tube.

If you choose to install a common air purger for the entire system, it must be positioned so that the entire flow of liquid circulates past the air purger.Any remaining air bubbles which were not separated when the system was charged are now circulating in the liquid. These bubbles pass through the micro-bubble separator where they are trapped and discharged via a float air purger.

Sub-pressure air purger or vacuum degasser

The methods above removed free air. To remove bound air it is advisable to use a sub-pressure air purger (pressure reduction). Sub-pressure air purging through pressure reduction should be used in order to remove also the bound air. Such a sub-pressure air purger or vacuum degasser is used to lower the level of dissolved oxygen and uses the fact that the solubility of gas is reduced with lower pressure (Comp. Figure 3.4.8). A partial flow of liquid from the system passes through the sub-pressure air purger, where the pressure on the liquid is reduced and the bound air is given off as free air and removed.

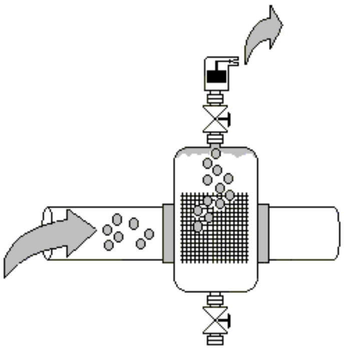


Figure 3.4.7. Active air purger, micro-bubble separator

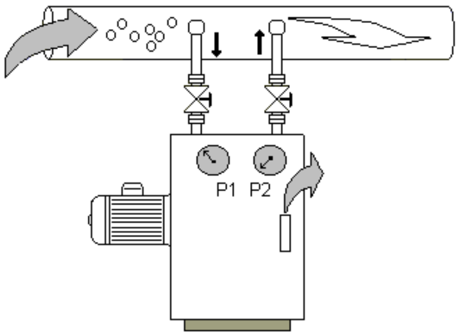


Figure 3.4.8. Sub-pressure air purger, vacuum degasser

A partial flow of liquid from the system passes through the sub-pressure air purger, where the pressure on the liquid is reduced and the bound air is given off as free air and removed. After that this liquid volume is released in the system again.

Another way of removing air from the system is to apply a vacuum to all or parts of the system before and during charging with heat transfer fluid.

Summing up air purging advice

The following facts should be considered when designing and assembling secondary systems:

It is easier to remove the free air that takes the form of air pockets and air bubbles in the system than to remove the air that is bound in the liquid.

It is therefore important as early as when charging the system with heat transfer fluid, to make sure that the free air can escape from the system via high points. When the system is charged from a low point, the air rises to the high points. Here, high point de-aerators are used.

When the system is full and pressurised, and the heat transfer fluid has begun to circulate, small amounts of free air will circulate with the liquid. This air comes from places in the system from which air was not displaced during charging (e.g. heat exchangers, fittings, etc.). These free air bubbles circulate with the liquid and are removed by installing an *active air purger* in the return pipe of the system.

To obtain a low air concentration in the liquid and so reduce the risks of corrosion or other problems, it is recommended to install a sub-pressure air purger in the system.

The equipment and methods to be chosen in order to create an air-free system depend on the design and size of the system. However, high point air purgers should always be installed in systems, since it is important to get rid of as much air as possible at the initial charging. Air purging is most effective at higher temperatures. The refrigeration unit must not be running during air purging. De-aerate the system thoroughly before putting it into service.

Consult the supplier of the air purging equipment about the location/installation requirements of each air purger and the compatibility of the air purger with the heat transfer fluid in question. Protective glasses should be used during air purging.

Air in the systemet is removed with effective air purgers placed where the static pressure of the system is low. Sub-pressure air purging is a good method. Automatic high point de-aerators shall be provided with a valve, so they can be closed off from the system when there is need.

(AMA-nytt 1999, Ekman 2003, KBL2 2000)

3.5. Closed secondary fluid systems with expansion vessel

3.5.1. Closed secondary fluid systems

The need to remove air from the system in order to prevent corrosion and to obtain right system function has already been considered. A system that has been freed from oxygen is according to recommendations to view it as a system with less than saturated corrosion risk. Nature strives for equilibrium with similar acid level both inside as well as outside the system.By keping a pressure in the closed system higher than outside, we get a system where air can not leak in and lessens the risk that the liquid is saturated with oxygen. The expansion tank with sufficient volume and prepressure shall be connected just before the suction side of the pump. The pump shall be ensured a sufficiently high static pressure on the suction side so that deposits in pump do not appear.

It is desirable to keep overpressure in all the system during operation. Usually the system is started at room temperature and the temperaturee is lowered successively. The liquid will then have lower volume and this volume reduction must be compensated for by adding more liquid.

3.5.2. Expansion vessel

In order to keep a constant overpressure at varying temperatures and when gas leaves the system (air purging) an expansion vessel is installed. This can be a closed, prepressured type, provided with a membran or a blower of syntetic rubber. The vessel consists of two chambers, of which one is meant for the system fluid and the other chamber contains compressed nitrogen gas, alternatively air. Pressurized vessels should periodically be inspected for security reasons.

In larger systems the pressure in the vessel can be maintained by a compressor or a pump. A compressor vessel with rubber bellows has even greater possibility to maintain an overpressure due to the compressor. An open system with a pump that keeps the pressure is a solution that automatically keeps the system pressure. There is a risk of air settling of the fluid but this can be prevented with a pressure free membrane. It is recommended that expansion vessels in secondary fluid systems are over dimensioned when there is no continuous system supervision.

When starting up and when the system keeps room temperature, the vessel should contain maximum liquid volume in order to be able to add liquid to the system when the temperature is lowered. In a system with prepressured expansion vessel it is important that the prepressure is checked and adjusted regularly. Control and adjustment can only be made when the vessel is empty of liquid. A correctly designed closing and drain valve should be mounted in connection to the expansion vessel. Adjustment of prepressure can then be done without larger draining of the secondary fluid. For dimensioning of the volume and choice of materials of the expansion vessel it is advisable to contact the supplier.

(When calculating the size of a closed expansion vessel, the expansion volume of the system is first calculated. This depends of amount of liquid, coefficient of volume expansion (varies stongly with type of liquid) and temperature span. After that consideration is given to the lowest and highest permitted pressures in the system, as this determines the size of the gas cushion. This volume gives, besides the expansion volume, the total volume of the vessel).

Static pressure

When the system is filled with secondary fluid the hight of the liquid column and the density of the liquid will make up the pressure that can be read on a manometer.

Example: The total hight from the lowest to the highest point in the system is 10 meter. The liquid consists of a mixture of organic salt and de-ionized water. The density of the liquid is by the manufacturer given to 1200 kg/m^3. The manometer P1 placed in the lowest point in the system will thn show: 1.18 bar. The pressure in the highest point of the system P2 will then be: 0 bar. However, if we desire an overpressure in the system of 0.5 bar; even at high point P2 thr manometer P1 will show: 1.18 + 0.5 = 1.68 bar. The above given values are valid when the circulation pump is closed.

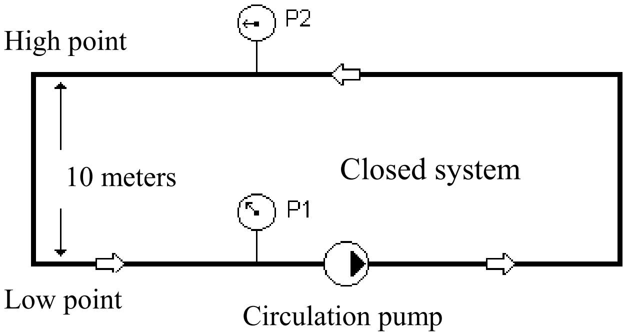


Figure 3.5.1. Static pressure

Prepressure in expansion vessel

It is important that the installer knows the importance of prepressure for correct function of the plant. It has been recommended to have a low position of circulation pumps and also that the expansion vessel should be placed in conjunction with the suction side of the pump. If the expansion vessel is placed according to this recommendation in the earlier example where we want an overpressure in the highest point P2 of 0.5 bar, the needed prepressure in the vessel P3 1.68 bar,. Membran vessels are usually delivered with a filled prepressure of 0.5 bar.

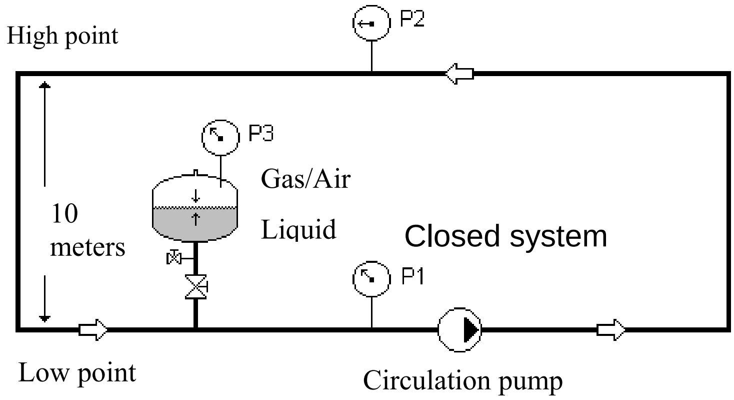


Figure 3.5.2. Prepressure in expansion vessel

Operation disturbances

In a prepressured and de-aerated avluftat (de-gassed) system with closed high point de-aerators can in principle no air enter. However, during interference / service of the system, air can enter. If the flow velocity of the liquid in some part of the system is too high (over 1.2 m/s) the static pressure in this point will decrease drastically. This happens also at abrupt changes of dimensions. In the worst case vacuum can appear in these points during operation and air can here enter the system. If the circulation pump is placed high in the system, there is risk that air will precipitate in the paddle wheel. This may result in less capacity of thin a e pump, noice and pump breakdown. The de-aerator of the plant will also not function correct way.

Check that the vessel is connected to the suction side, increase the system pressure and the prepressure until the air and the disturbance disappears. during operation, that the manometer (P1) the suction side of the pump shows needed inlet pressure. This pressure can be obtained from the manual/operation instruction of the pump.

Dimensioning of expansion vessel

The task of the expansion vessel is to "take care" of the variation in volume that may appear in systems when the operating temperature varies. The dimensioning must be performed with a certain care in order for the plant to get a satisfactory function as to correct amount of fluid and suitable system pressure at varying operation cases.

Closed expansion vessels are recommended for smaller systems. Larger systems may have very large closed expansion vessels. An alternativ can then be to choose an open expansion tank in combination with a pump that maintains the pressure and a suitably adjusted bypass valve. At lowering temperature the pressure level will decrease, which can be sensed by a pressure sensor that start the pump that maintains the pressure and that pumps in liquid from the open expansion tank (normally the existing mixing vessel). With rising temperature the pressure level in the system increases, causing the bypass valve to open and let the surplus return to the open vessel. If the plant has the same type of fluid on the "cold" and "warm" sides a common expansion vessel can can be a good alternative. The expansion vessel is usually placed on the suction side of the primary pump where the system pressure normally is lowest. (Ekman 2003, KBL4 2003)

3.6. Defrosting

3.6.1. Introduction

Humidity in the air will condense as frost or ice on the finned coils if the surface temperature is below 0°C. Frost growth on the finned coil surfaces may improve heat transfer rate depending on what type of goods that is stored in the cooling room. Goods with high water content that are unpacked increase the need for defrosting. A certain amount of frost on the surfaces can improve the heat transfer but if frost growth is allowed to increase too much then the capacity of the finned coil will gradually decrease because of poorer heat transfer.

This deterioration is mainly due to reduced air flow through the air coil as a result of inceased pressure drop through the air coil. Hence, the air coild should be deforsted with even intervals in order to get he plant to function well. (Aittomäki 2003, Ekman 2003, Granryd 2005)

3.6.2. Defrosting methods in secondary fluid systems

Introduction

Some defrosting methods that are used in indirect systems with secondary fluids are:

- Self defrosting with the aid of the surrounding air for temperatures over about +2°C. Fans hasten the defrosting speed and can be on or off depending on how long time the defrosting may take inview of demands on temperatures and product quality.
- Electrical defrosting rods that goes through the heat exchanger (often in a similar way as the secondary fluid tubes) is a common method. The fins are first warmed up and then the cold secondary fluid in the tubes will be heated up leading to longer defrosting time (about 5-10 min.) than with direct expansion coils.
- Defrosting with the help of warm fluid is a rather quick method. The liquid can be heated with electrical heating element, heated secondary fluid from a tank, from the condenser of the freezer or from secondary fluid of the warm circuit. The temperature of the liquid shall not be too high. At freezer applications it can be +6 to +8°C while for cooling applications it can be somewhat warmer, i.e. +10°C.

(Aittomäki 2003, Granryd 2005)

Stop defrosting of secondary fluid systems

Defrosting often takes place after pre-set defrosting times. A breaking function calls for a defrosting cycle that stops the liquid cooler causing heat to be added to the secondary fluid system. The pumps continue to maintain a circulation through all the cooling objects in order to defrost all objects the same way. The defrosting thermostat senses the incoming secondary fluid temperature and closes at pre-set end value and then the liquid cooler re starts.

An increase in the recommended value of the secondary fluid during a predermined time can also be an alternative in place of stopping the defrosting over the thermostat.

Quick defrosting of secondary fluid circuit

Defrosting takes place after pre-set defrosting times. A breaking function calls for a defrosting cycle and opens two motor valves between the warm and cold circuits causing the secondary fluid to be mixed with the cooling fluid circuit. A prerequisite is that it is the same type of fluid or alternatively a heat exchanger can be mounted between the circuits. A minimum limitation is made of the cooling fluid temperature to + 20°C by closing one of the motor valves. The defrosting thermostat senses the incoming secondary fluid temperature and stops at pre-set exit value at which the liquid cooling unit restarts.

At an alternativs quich defrosting method a defrosting pump and a heat absorbing pump work continually during defrosting, start delayed with 90 sek. The two motor valves open simultaneously and the cold secondary fluid is mixed with the heated fluid. The defrosting thermostat feels the return temperature of the secondary fluid and closes at pre-set value, at which the pumps stop and the two motor valves close after the delayed set time. During the During the delayed time the fans should be in operation. Defrosting normally takes place 1 time/24 hours. (DEM, K. Svensson, Frigotech)

3.6.3. Classification of defrosting system

To accomplish a defrosting energy be added to the coils either from the cooling room or from an outer energy source. Added energy shall initially heat up the coils, including the amount of secondary fluid that is inclosed, to 0°C. After that additional energy has to be added corresponding to the heat of melting for the frost/ice on the coilsurfaces. In certain cases the whole secondary fluid system is heated up during a defrosting. See below.

Defrosting should in principle be stopped when the last melting water has ben drained away via the drip plate. It is usually not possible and the coils are often heated up a few degrees above zero. If the coils are heated by an outer heat source it can in unfavorable cases reach temperatures much higher than the cooling room, which leads to an increased cooling demandand poorer energy efficiency.

It is also imptorant that the available heat source can give sufficient power to give a reasonable defrosting time. An upper limit is ca: 40 minutes.

Another very essential question is wherther the coils shall be defrosted individually, or if the whole system shall be defrosted together or if a combination of both alternatives is to prefer. There is no unequivocal answer to the question but choice of defrosting system depends on the type of plant. At the end of this part there is a discussion of choice of various system types in applicable cases. (Ekman 2003)

3.6.4. Individual defrosting

If hte separate coild shall be defrosted separately the defrosting methods below can be a possibility:

- Stop defrosting (energy taken from cooling objects)
- Electric heat defrosting (electric rods mounted in the coils).

In the case of stop defrosting the secondary fluid through the coils is stopped (magnet valve) when a defrosting is initiated. The fan of the air cooler is in operation during the defrosting cycle and "warm" cooling room air is circulated through the coils. The room temperature should not be below about 2°C. Otherwise defrosting times may be too long. This method is superior as to energy efficiency when energy is taken from the cooling room. Regarding the control of defrosting, see point 3.6.5.

Electric heat defrosting is from an investment cost point of view more expensive than stop defrosting. Electric rods are mounted in the coils that usually are prepared for this purpose. A difference with stop defrosting is that the method can cause the surface temperature of the coils to be very high locally. This can mean that the melting water formed on the coils will boil, or evaporate to the surounding. Heat losses to the surounding (the cooling room) increase the cooling need and impair the energy efficiency. The cost of energy for the electric heat will also be added. (Ekman 2003)

3.6.5. Joint defrosting

At joint defrosting the coling unit is usually stopped and heat is added to the secondary fluid system.

This presentation is limited to defrosting with central shunt (3-ways valve), that is all cooling objects requiring defrosting, are defrosted simultaneously.

Defrosting with central shunt shunt is suitable for two circuit systems with ackumulator and for and for single circuit systems if you can not permit the coling unit to be run through by warm secondary fluid. The principle arrangement for central shunt is evident from Figure 3.6.1.

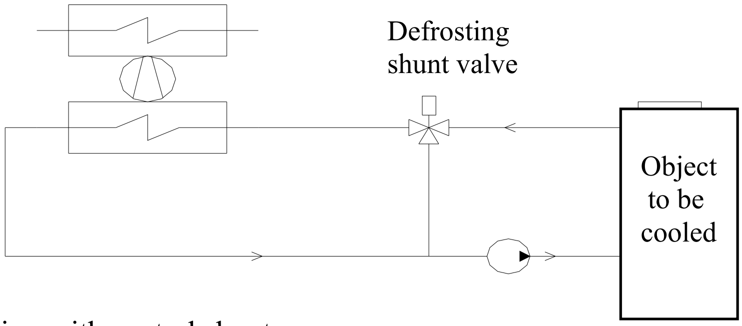


Figure 3.6.1. Defrosting with central shunt

The heat sources that can be vara in line for joint defrosting are:

- Electric heat (immersion heater in the secondary fluid system).
- Heat energy from the freezer condenser connected to the secondary fluid system.
- Heat energy taken from the coolant side either in stored form or directly if the cooling and freezer unit consists of two separate units.

Defrosting of the system is directed by superior controlling. The unit is stopped and the 3-way valve is put in defrosting position. To obtain defrosting of the cooling coils there must be a suitable heat source connected to the secondary fluid system.

When choosing heat source available heat within the system should first be used. With system is here meant cooling and possibly freezer unit.

Heat from freezer condensers are primarily chosen if such are found and secondarily heat from the secondary fluid for the condenser side of the cooling system.

In the first mentioned case the freezer condensers are connected to the secondary fluid system. The coupling can be viewed as a cascade connected two stage plant with the secondary fluid as middle circuit. During the defrosting cycle the freezer unit has to be run compulsory. This can mean that for a short while there may be a lowered temperature in the freezer rooms. Observe that the coupling also affects the avaiability at a crackdown (See chapter 7.1). If the condenser power is small some support can be given with electric heat.

If there is no freezer plant then the heat can be taken from the "warm side" of the cooling unit. The system solution means that required energy must be stored up during ordinary operation. One solution is to install a heat accumulator in the pipe system at one distribution port of the shunt valve see Figure 3.6.2. The accumulator is charged over a middle circuit.

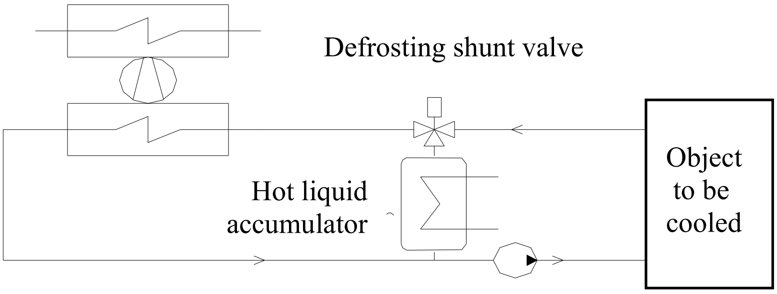


Figure 3.6.2. Principal arrangement with defrosting over a heat accumulator

The heat source can either be outgoing secondary fluid from the condensers or produced with aid of hot gas heat exchanger. Higher temperature levels can be obtained in the latter case which lessens needed accumulator volume. Hot gas heat exchanging leads to longer charging time.

If a separate freezer unit is installed heat energy can be taken from its outgoing secondary fluid from the condenser side which directly can be heat exchanged with the secondary fluid system of the evaporator side. Accumulator is not needed in this case. However, the freezer unit must forcibly be kept running during the defrosting periods.

As a last alternative an electric immersion heater is installed in the secondary fluid system (see further under choice of system). (Ekman 2003)

3.6.6. Control of defrosting

An optimally designed defrosting system should be ska vara operated by demand. Demand operation means that the equipment continuously supervises the frost formation on the coils. When a "critical" frost growth has been formed on the surfaces of the coils the defrosting cycle is initiated. When a "critical" defrosting means here that the defrostning intervals shall be chosen to obtain lowest possible energy consumption for the system as a whole. The defrosting time shall also be adjusted to avoid unnecessary over temperatures.

Constructing a system according to what is mentioned above is fully possible but implies in most cases an unreasonably high investment cost. It is only very large indirect systems that fully can carry the cost for demand operation. Usually there is a compromise between optimal function and investment cost.

The simplest form of defrosting automatic mechanism is to install one or more defrosting clocks. These are tuned after practical tests for the prov för "worst case". The defrosting clock operates start and stop of the defrosting cycle. If load and cooling object varies this will in certain cases result in a number of unnecessary defrosts. The same applies to vid adjustment of the length of the defrosting time. One way to improve the automatic mechanism is to install a thermostat that stops the defrosting. The bulb of the thermostat is surface mounted on a representative place on the coils. A thermostat can be used to stop the defrosting for several sets of coils.

In defrosting systems operated by demand there are some alternatives to choose from. One way can be to measure the pressure drop on the air side of the coils with a differential pressure sensor. Increased pressure drop indicates increased frost growth. Another alternative can be to measure the temperature difference between the air temperature and outgoing secondary fluid temperature with a simple differential thermostat. Increased temperature differences lead to more defrosting.

The problem is in the first case to find pressure sensors that can measure small pressure sets with enough accuracy to a reasonable price. At temperature difference meaurements partload operation can lead to smaller differences. A result can be that no defrosting is initiated, although there is a need for it. It should be mentioned that the partload case in itself lessens the coil load and a larger defrost growth may be acceptable.

If demand operated defrosting is chosen (as mentioned above) a thermostat should be mounted that senses the surface temperature and stops the defrosting (Ekman 2003).

3.6.7. Choice and control of defrosting system

If the plant has a few coils that need defrosting and the temperature levels in the cooling room is above about 2°C then stop defrosting should primarily be chosen. If a room is to cold stop defrosting then electric defrosting is chosen for this alternative. Electric defrosting is always chosen for coils with natural convection.

Defrosting with central shunt can be a good alternative if stop defrosting can not be applied. This mainly aplies to larger systems.

Choice of heat source should primarily be the cooling fluid from freezer condensers that are connected directly to the secondary fluid system (cascade connection). Note that this alternative often means that emergency cooling must be installed to the freezer condensers in case of a crack down.

If there is a separate freezer plant then cooling fluid from heat condensers should be the heat source. The choice means that there is no need to during defrosting.

If there is no freezer plant then heat will be taken from the warm side of the cooling unit and be accumulated.

Electric heat from a immersion heater in the secondary fluid should be avoided as this is the worst from energy saving perspective. (Ekman 2003)

3.7. Other components

Besides the components that have been considered there are various types of filter, valves, sealings, gaskets, joints, splices, valve sealings, drip plates, measuring devices, manometers, etc. Of course there are also various types of tubes or pipes for secondary fluid system.

3.7.1. Filter

Filters should be mounted on the pressure side of the pump. Filters with suitable mesh size (max. 1.0 mm) should in systems with plate heat exchangers be mounted before the heat exchanger. Clean the filter regularly.

Systems with operaton temperature below 0°C should preferably be welded or soldered. Couplings should be chosen considering the risk for frostburst.

Pumps should be provided with closing valves and fully covering drip plate.

It is important that there are no loose particles in the fluid. The particles destroy sealings in valves, pumps, etc. and as a result cause leakage. A dirt filter should be installed to avoid particles in the system. The mesh size of the filter should preferably be below 50 µm (0,050 mm).

3.7.2. Measuring devices, manometers

Manometers mainly used to measure pressure changes over i.e. dirt filter. Many cheaper manometers can contain components of less noble materials, such as tin that should be avoided. It is for this reason very important to check that the materials in the manometers are compatible with the secondary fluid. Check that the equipment functions and is does not leak.

3.7.3. Valves

Various types of valves are used to be able to regulate and close various flows. In the context of secondary fluid two types of valves are most commonly used, throttle valve and ball valve. The throttle valve is used with larger tube dimensions and is then wedged in between two fins. The ball valve is used with smaller tube dimensions and is preferably mounted fully welded or fully soldered with silver plumb to lessen the risk for leakage. Check regularily that the valve functions and is compact. Rinse immediately away any spill.

(KBL4 2002, Moberg 2002)

4. Secondary fluids as secondary working media

(Å. Melinder, B. Bredberg; M. Feltwall; J. Ruchatz)

4.1. Introduction

4.1.1. Name and function of the fluids

The term secondary fluid is here used for fluids that take up and transport energy from cooling objects or heat source on the cold evaporator side in an indirect system. On the warm condenser side in a cooling system the fluid is here termed secondary fluid for the condenser side. This fluid transports heat from the condenser to heat recovery or to liquid coolers that usually are placed outdoors. At air conditioning systems and heat pumps water is often used.

The terms of the fluids in English is either connected with one of the areas of application or function: *Secondary refrigerant* (IIR), *secondary working fluid* (IIR) and *secondary coolant* (ASHRAE, USA) all suggest liquids for an indirect or secondary system. *Heat transfer fluid* and *antifreeze* indicate one of the functions of the fluids. The German, French and Swedish terms, K*älteträger*, *Frigoportaire* and *köldbärare*, all indicate that they transport cold. *Brine* means salt solution but is often (not fully correctly) used also for other types of secondary fluids. In this book we use the term secondary fluid as explained above (See also 1.2.1).

4.1.2. What types of secondary fluids are used today?

Water is excellent as secondary fluid as long as the temperature is above 0°C. A freezing point depressant additive to water is needed for secondary fluid temperatures near or below 0°C. The most common basic additives are ethylene glycol, propylene glycol, ethyl alcohol (ethanol), calcium chloride, potassium acetate and potassium formate but glycerol, betain, potassium carbonate and sodium chloride are used in certain applications. Water solutions of methyl alcohol (methanol) and ammonia are also used for a number of special applications but these types are not treated much in this publication.

We wil mainly cover water based secondary fluids and only briefly mention syntetic oils and phase changing media such as ice slurry and carbon dioxide.

A number of commercial products are marketed under various names. It is important to know what basic chemicals they contain and this should be evident from the product information sheet for the product.

(AMA Nytt 2005, Melinder 1997, Melinder 2007)

4.2. Desired properties of a secondary fluid

The secondary fluid should be chosen considering the thermal properties at the operation temperature, environmental and health aspects as well as choice of materials in the system and corrosion. Generally speaking there is no secondary fluid that is ideal for all applications. There is a need to find out what demands on secondary fluid and choice of materials each type of system design requires. The following properties should be considered when choosing secondary working:

4.2.1. Thermophysical properties

A secondary fluid should have sufficient freezing protection and should have good thermophysical properties, high density, specific heat and thermal conductivity but low viscosity at the operating temperatures. The secondary fluid is then able to:

- transport large refrigeration capacity through pipes or eller tubes with a given volume flow
- give good heat transfer leading to a small temperature difference for the heat transfer in evaporator or cooling place or heat source, and give small pressure drop so that a pump with small or reasonable power can be used.

4.2.2. Other properties and aspects

The following aspects should also be considered when deciding what type of secondary fluid to be used in a plant:

- Corrosion aspects and material compatibility, how the fluid affects metals and other materials in the system. It is important that the fluid does not cause material problems in form av of corrosion. The tendency for corrosion should be carefully considered. Corrosion aspects are treated in chapter 5.
- Environmental aspects, toxicity, health aspects, biological degradation in ground and water, flammability, effect on outer environment. Environmental aspects can be of crucial significance for choice of secondary fluid type and consideration should be given to the total environmental impact of the product. The effects of the products influence on the immediate environment, the working environment, outer environment and also health risks. Environmental aspects are considered in chapter 6.
- Control and maintenance requirements, handling regulations and transport aspects. Need of protective equipment (See also chapter 9).
- Economic aspects: Purchase price and its influence in an LCC perspective. Demands on special constructions and material as well as maintenance and control costs. (Compare also chapter 10 on Optimization).

It is important to examine the information and security sheets of the product carefully with regard to these aspects.

4.2.3. Some problems with various types of secondary fluids

Some problems with the various types of secondary fluid are here briefly mentioned.

- Water freezes to ice at or near 0°C. Water and the following water solutions are corrosive in the presence of oxygen if effective corrosion inhibitors are not added.
- Ethylene glycol is very toxic for humans (A glass of mixed glycol could kill an adult).
- Propylene glycol, glycerol, betain and ethyl alcohol have high viscosity, especially at lower temperatures.
- Flammability risk determines safety demands on the use of ethyl alcohol (ethanol) in higher concentrations (≥ 30 % by weight).

- Many today hesitate to use salt solutions, both chlorides and potassium salts, due to worry for corrosion problems. Chloride salts are strongly corrosive to steel products while potassium salts are corrosiva to zink, soft soudering and some also to aluminum.

  (Melinder 1997, Melinder 2003, Melinder 2007a)

4.3. Choice of secondary fluid and additive concentration

For choice of secondary fluid and for technical calculations of an indirect refrigeration or heat pump system it is important to know oncentration and freezing point temperature as well as data for density, volume expansion, viscosity, specific heat, and thermal conductivity.

4.3.1. Freezing point and additive concentration are connected

The freezing point is the temperature at which ice particles begin to form in equilibrium and should be chosen below the normal operating temperature of the fluid so that it without difficulty can be pumped around in the system. If the normal operating temperature is -10°C the freezing point can be chosen to t_F = -15°C or -20°C. A lower freezing point than necessary should not be chosen as it impairs the properties of the secondary fluid.

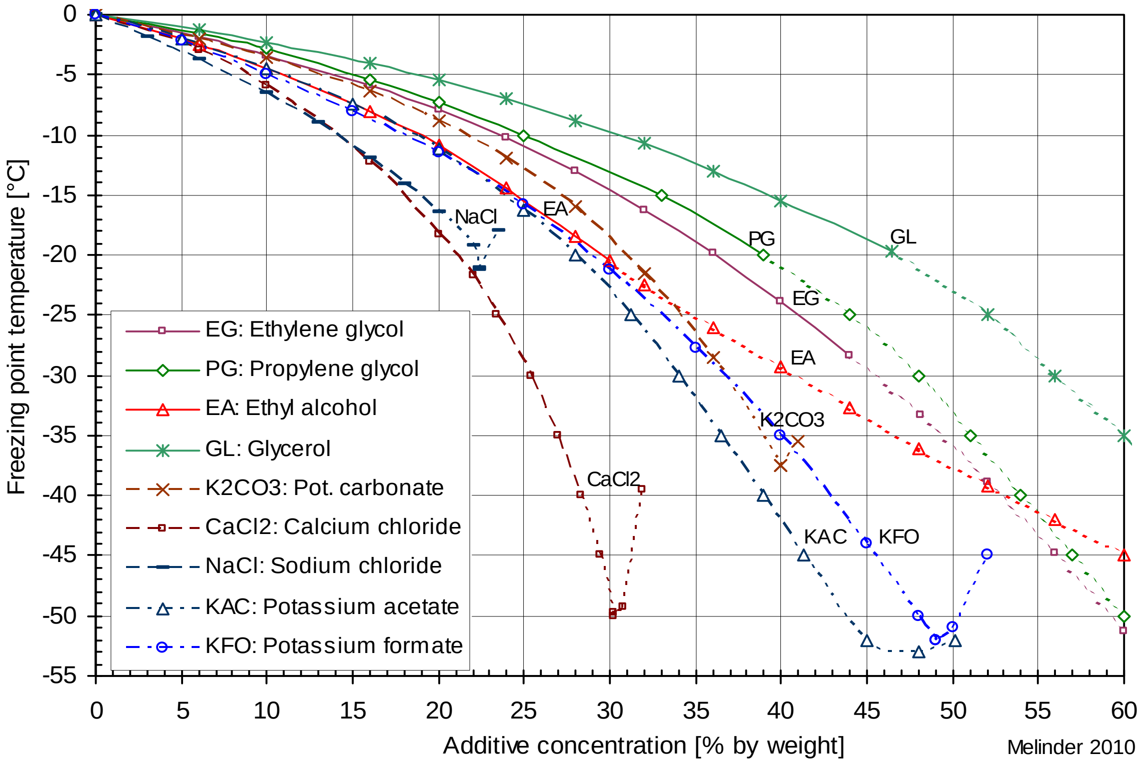


Figure 4.1. Freezing point as function of the additive concentration

Figure 4.1 shows the freezing point as a function of the additive concentration for a number of secondary fluids. The curve is drawn to the eutectic point for certain salts. Water solutions do not usually freeze to a solid mass near the eutectic temperature but none of these secondary fluids can without risk be used at an operation temperature below -40°C.

A concentration higher than the eutectic shouldbe avoided as the ”freezing point” curve then quickly rises as salt chrystals are formed already at these higher temperatures. The risk for forming of solid ice in tubes or heat exchangers is small with water solutions even 30% concentration of ethylene glycol (t_F = -15°C) is recommended in ASHRAE 2009 in order to avoid damage of the system.

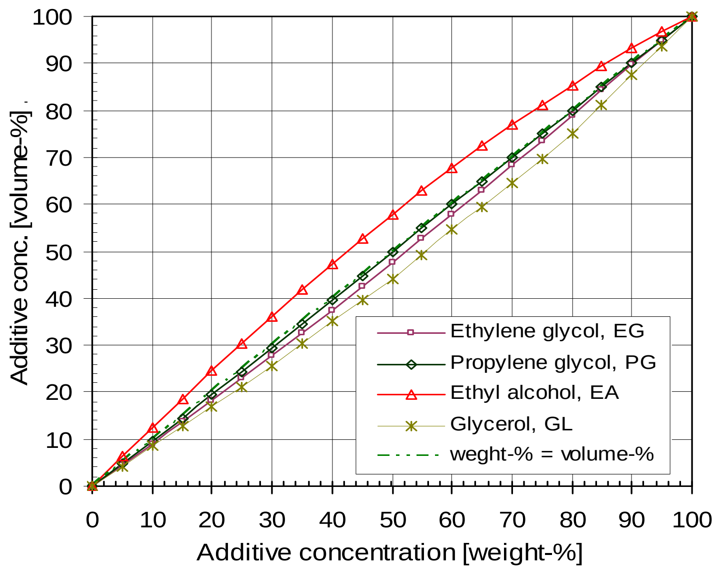


Figure 4.2. Additive concentration in volume-% as function of weight-%

When a water solution of glycol, alcohol or glycerol shall be mixed it is the additive concentration by volume [rather than by weight that is asked for.

Figure 4.2 shows the relation between volume and weight concentration for glycol, ethyl alcohol and glycerol. The curve for ethyl alcohol isa above the straight line which is because of ethyl alcohol has lower density than water.

4.3.2. How can additive concentration and freezing point be determined?

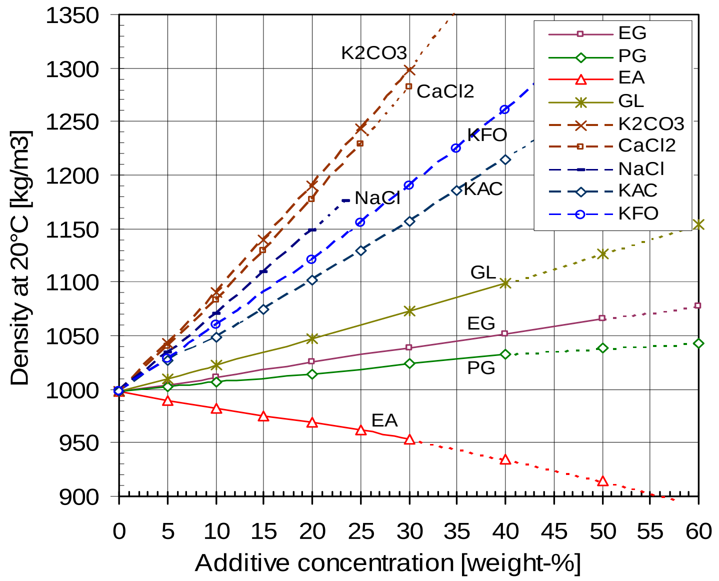


Figure 4.3. Densitety at +20°C as function of additive concentration

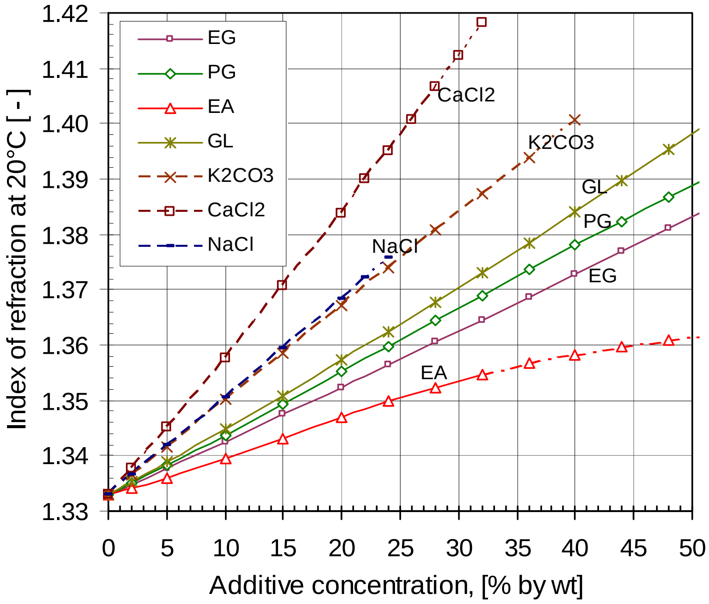
The concentration of a known type of water solution is often determined by measuring the density, by weighing a known volume of the liquid with an aerometer or densimeter or with refractive index or gas cromatografy.

Figure 4.3 gives the densitety at +20°C as function of the additive concentration for the various types of secondary fluids.

Only ethyl alcohol of these water solutions has lower density than water.

4.3.3. Determining the additive concentration with refractive index

Figure 4.4 gives refractive index at +20°C as function of additive concentration.



A glycol meter where the additive concentration and freezing point of a glycol is determined using the refractive index of the liquid.

Can a glycol meter be used for another type of water solution? Yes, what really is measured is refractive index and it should principally be possible to convert a glycol concentration to that of another water solution with the aid of Figure 4.4 where refractive index is given as function of additive concentration for water solutions. If, as an example, the meter shows 40% ethylene glycol it corresponds to≈16% calcium chloride $CaCl_2$.

Figure 4.4. Refractive index at +20°C as function of additive concentration

(ASHRAE 2005; CRC 1986-87, Mellan 1977, Melinder 1997, Melinder 2007a)

4.4. Suitable temperature spans for various secondary fluid types

(Å. Melinder, B. Bredberg)

What is a practical span for using a secondary fluid for the cold or warm side is mainly determined by its freezing point curve and its viscosity. Other thermal properties such as flash point och corrosion rate at different temperatures can influence. The secondary fluids found in Figure 4.1 to Figure 4.3 are given with **bold** text in the table.

Table 4.1 gives the practical span for use, i.e. operation temperature, for the different water solutions used as secondary fluids. The freezing point is often chosen 10ºC below the operation temperature. A short explanation is first given to the table for water and the various water solutions. It can be good when needed to find out if the commercial secondary fluid products functions well during longer time at higher temperatures than the room temperature.

Water freezes near 0°C and can usually be used down to +2 ÷ 4°C.

Glycols: Certain glycol products are suitable for temperatures up to and at times over +100ºC. Even if the theoretical area of operation of glycols is from -55ºC to +120ºC there are the following practical limitations: At high temperatures > +80ºC there is risk for breakdown of the product which among else affects inhibitors and hence gives risk for corrosion. The high viscosity limits the use at low temperatures.

Ethylene glycol – water can be used down to the operation temperatures at about -30ºC and propylene glycol – water to about -20ºC. However, well dimensioned heat exchangers, pumps and tubes are needed for good function and acceptable energy exchange.

Ethyl alcohol (ethanol): The flash point limits the use of ethyl alcohol – water, for temperatures above about 25ºC, as well as for high concentrations > 30 weight-% .

Glycerol: The high viscosity limits the use of glycerol – water down to operation temperatures of about -20ºC as for propylene glycol. The same should be true of betain – water. It is good to find out if commercial products function well for a long time at higher temperatures than 40ºC.

Chloride salts: The eutectic point (see Figure 4.1) and the steep freezing point curve near this point limits calcium chloride – water to about -40ºC, sodium chloride – water to about -10ºC while lithium chloride from this point of view can be used even somewhat below -50ºC.

Organic potassium salts: Figure 4.1 suggests that the freezing point curve near the eutectic point is not a steep as for the chlorides. Usually the freezing point is recommended ju to be chosen about 10ºC below the operation temperature limitng this temperature to -40ºC or somewhat lower for potassium acetate and potassium formate. Commercial products, incl mixtures of these salts, at times give somewhat lower operation temperature.

Table 4.1. Practical span for using secondary fluids (for evaporator and condenser sides)

Practical operation temperature

Water solutions	100°C	80°C	60°C	40°C	20°C	0°C	-10	-15	-20	-25	-30	-35	-40	-45	-50°C
Water	+	+	+	+	+	+									
Water solutions															
Ethylene glycol	+	+	+	+	+	+	+	+	+	+	+				
Propylene glycol	+	+	+	+	+	+	+	+	+						
Ethyl alcohol				+	+	+	+	+	+	+					
Glycerol				+	+	+	+	+	+						
Calcium chloride				+	+	+	+	+	+	+	+	+	+		
Sodum chloride				+	+	+	+	+							
Pot. carbonate				+	+	+	+	+	+	+	+				
Pot. acetate		+	+	+	+	+	+	+	+	+	+	+	+	+	
Pot. formate				+	+	+	+	+	+	+	+	+	+	+	
Betain				+	+	+	+	+	+						
Methyl alcohol				+	+	+	+	+	+	+	+	+	+	+	+
Ammonia					+	+	+	+	+	+	+	+	+	+	+
Litium chloride				+	+	+	+	+	+	+	+	+	+	+	+
Two phase media															
Ice slurry						+	+	+	+	+	+	+			
Carbon dioxide								+	+	+	+	+	+	+	+

Two phase media: Ice slurry has its great potential at temperatures around -5 to-10ºC but has been tested down to -35ºC. Two phase carbon dioxide is best suited at freezer temperatures, about -30ºC when the pressure is not too high.

4.5. Dimensioning of freezing point (B. Bredberg)

Dimensioning of freezing point of a secondary fluid is determned by the lowest temperature the system is exposed to. In a cooling system the evaporation temperature is the lowest temperature and the freezing point of a secondary fluid must be dimensioned after it, not after the operation temperature. Most evaporators and condensers are today designed as compact plate heat exchangers and their function may be affected if ice crystals are formed in the fluid (Comp. Figure 4.5).

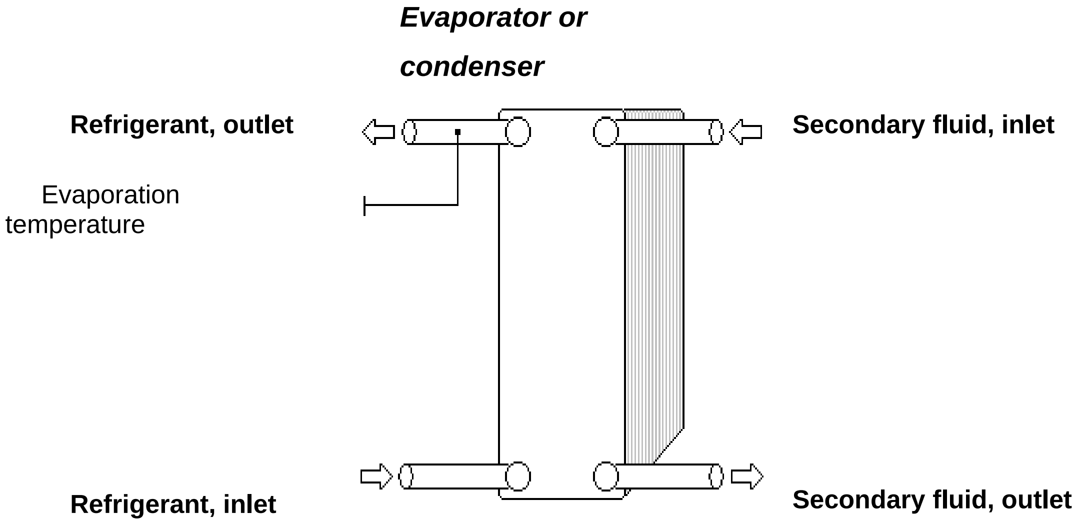


Figure 4.5. Plate heat exchangers as evaporator or condenser

Exemple 1: Dimensioned operation temperature -8°C which normally gives an evaporation temperature of -13° to -15°C. A security marginal of 5°C for variations in the system should be calculated, giving a dimensioned freezing point of -20°C.

Exemple 2: For a liquid cooler placed outdoors the freezing point for the secondary fluid on the condenser side should be dimensioned after the lowest outdoor temperature. The physical properties of the secondary fluid should be determined from the dimensioned freezing point (concentration) and the operation temperature.

Ice crystals are formed in the fluid if the freezing point is dimensioned too high or if the concentration på of the secondary fluid is not obtained. These ice crystals increase the viscosity and impair the flow which leads to lower cooling power. The cooling mashine will work at too low temperature which brings the risk of stop via low pressure breaker.

It should be mentioned that there is little risk if ice crystals are formed in the fluid. An ice slurry is formed and it ay even improve properties! The risk is rather that the ice crystals get stuck on the evaporator surface and block the flow.

However the system will not break due to freezing, the evaporator does not break due to increase in volume which can happen with pure water. Compare car engine with or without glycol.

4.6. Practical advice for use of secondary fluids

4.6.1. General

- When using concentrates intended to be mixed with water, it is important to mix with the right water quality. Contact the supplier of the secondary fluid for information about water quality needed for the product.
- Always readt the Product information for the fluid chosen before using the secondary fluid. Check the secondary fluid acording to the guidelines of the supplier.
- If using secondary fluids with inhibitors for applications within the food sector, use either ready-mixed solutions from the supplier or mix with deionized water.
- It is important that the secondary fluid products contain a suitable inhibitor package that gives good corrosion protection.
- From the marking of the wrapping, product information and security data sheets, it should be evident what secondary fluid that has been used as base product. The identification should be easy, because, as an example, it is not suitable to mix ethylene based products with propylene based products.
- It is strongly recommended that every plant has an”identity card” where it clearly is evident what products have een used, freezing protection or concentration, date of charging the system and who did it. The main reason is the thermo physical properties in the system, freezing protection, viscosity etc., will change when mixing. Various types of corrosion inhibitors can give unwanted effects when mixing, such as deposites and/or gel formations that affect or prevents circulation.
- Information about type of secondary fluid and concentration should clearly be given at the plant.
- Inhibitor content varies both in amount and quality and can hence be directly damaging for pumps, sealings and construction details in the plant.
- All secondary fluids should be treated as chemicals and not consciously be let ous in the drain. Restrictions regarding each secondary fluid is provided by the the supplier. During Vid drainage/destruction of secondary fluid from the system the local environment and health authority should be contacted. Used secondary fluid is to be classified as industrial refuse.
- Motor coolant glycol should not be used in this type of system due to unsuitable inhibitor composition which can cause pump crackdown. The most common motor coolant glycol are based on ethylene glycol but even propylene glycol (so called ecological glycol) but these are formulated for other environments and temperatures.

4.6.2. To think of when using ethyl alcohol

Alcohol can be used as energy carrier basically within he same areas of use as glycol, with the reservation for limits regarding working temperatures due to the inflammable properties of the alcohol. In collectors for surface ground, rock, and surface water heat sources mainly ethyl alcohol are used.

Ethyl alcohol or in daily tal ethanol can be produced in two ways– chemically via cracking of eten and via fermentation and destilling of organic materials such as cellulose, molasses, grains etc. The ethanol that is produced in the latter way is in daily speech called kallas bio-ethanol as they are produced from renewable sources. When bio-ethanol is used as energy carrier it is usually called (in Swedish) "köldbärarsprit". It has both advantages and drawbacks. To the advantages belong their environmental properties (bio-ethanol), during production aswel as possible leakage. Ethanol is moreover easily biodegradable in nature.

Facts about the flash point

The flash point, t_{FL} [°C], is the lowest temperature at which a liquid can be ignited by an (open) flame. When using ethanol in an indirect system, the safety regulation in Sweden says that the highest ambient temperature in cooling systems should be at least 5°C below the flash point of the ethyl alcohol concentration. Safety regulations are strict mass concentrations above 30%.

Liquids that easily set on fire are in Sweden classified from Class 1 to 3 based on the following flash point values:

1:	$t_{FL} < 21°C$	$c_A > 70$ %
2a:	$21°C \le t_{FL} < 30°C$	% $70 \ge c_A > 30$ %
2b:	$30°C \le t_{FL} < 55°C$	% $30 \ge c_A > 10$ %
3:	$55°C \le t_{FL} \le 100°C$	% $10 \ge c_A \ge 0$ %

Class 1 and 2a are strictly regulated. The flash point for pure (100%) ethyl alcohol, $t_{FL} = 12°C$ (Svensk Kylnorm 1992).

Use of methyl alcohol (methanol), should be restricted in similar ways.

Figure 4.6 gives flash point, t_{FL} [°C], for concentrations of ethyl alcohol -water.

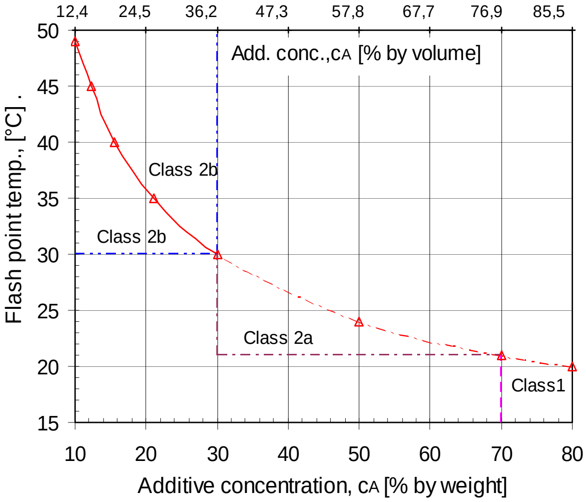


Figure 4.6. Flash point for ethyl alcohol-water

Fire hazard

- Among the drawbacks are that the product is may be a fire hazard which affects storage and transport possibilities. Concentrated bioethanol (95%) has fire class 1 and is classified as dangerous goods during transport. A common mixture is 30 weight-%, which gives a freezing point of about -20ºC and is the limit for a lower fire class, 2B.

- Regading storage, always consult the Rescue Services Administration and the explosives inspection agency to find out local regulations bestämmelser (See facts about flash point).

- The following advice regarding safety during handling can generally be proper regarding preparation of flammable liquids:

- No open fire. All spark generation shall be switched off. Electric motors, heat boilers, etc. shall be decoupled. Care for good ventilation. Work outdoors preferably.

- During dilution in mixing vessels add water first and first then the concentrated solution.

To avoid the formation of static electricity – see that the concentrated secondary fluid ethanol is not poured from a high place down in the mixing vessel. Always ground the equipment.

The SVEP-norm

Svenska Värmepumpsföreningen (SVEP) has developed a norm for how a secondary fluid ethanol to be used in collectors to energy wells shall be. The SVEP-norm has in Sweden come ato be the norm for the in Sweden market.

Quality demands for ethanol used for freezing protection in secondary fluids in collectors in ground, rock, and surface water to the cold side of heat pumps. Only 95 – 99.5% bioethanol or alternatively a technically produced ethanol may be used as base. The ethanol shall be denatured according to Swedish norm.

Quality demands that the drilling contractor andh/or heat pump installer is responsible for: The ethanol shall be denatured with 10% denaturering agent consisting of Isopropanol and N-butanol in accordance with the permission of the Food and Drug Administration in Sweden. Denaturering content over 10% can not be accepted as the ground water quality can be affected negatively at possible leakage**.** A colour fit for human consumption shall be added.

Laws to heed: During transport, storage, handling and installation of ethanol as secondary fluid in a heat pump setting the following laws should be heeded:

For transport: Rules ADR-S, SRVS 2002:1 of the Swedish Rescue Services Agency.

För storage and handling: Rules SÄIFS 1996:6, SÄIFS 1997:3 and SÄIFS 1997:3 pl. 4 of the explosives inspection agency in Sweden.

För installation: Rules SÄIFS 1997:9 of the explosives inspection agency in Sweden.

4.7. Properties for technical calculations of indirect systems

For technical calculations of indirect refrigeration and heat pump systems it is of importance to know the concentration and freezing point temperature ans well as data for density, volume expansion, viscosity, specific heat and thermal conductivity. Data for these properties are needed to determine the cooling capacity that a certain volume flow can transport, heat transfer at heat excgangers och pressure drop in the system.

Data for technical calculations and a more detailed description of the physical properties of various secondary fluids are given in the IIR publication**:** ”Thermophysical properties of liquid secondary refrigerants – Tables and diagrams”. Author: Åke Melinder KTH Stockholm (Melinder 1997).

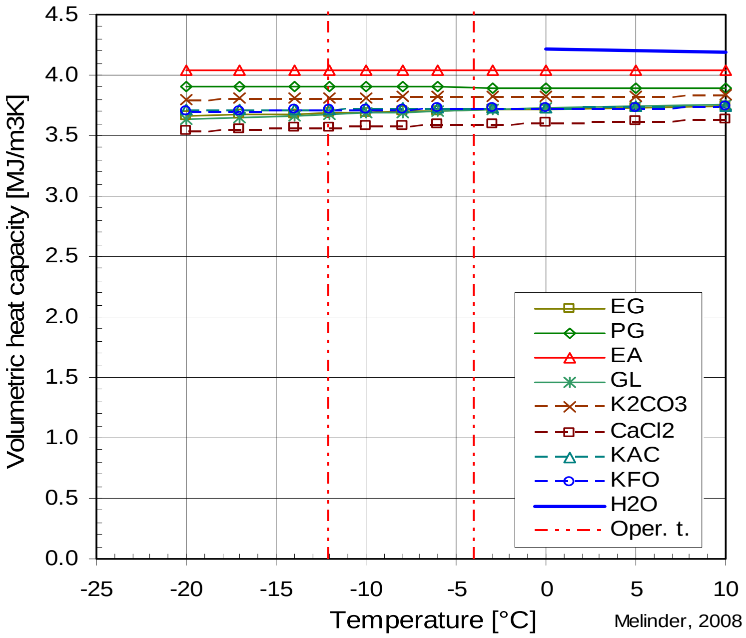
A new IIR publication in English with the title: ”Properties of secondary working fluids for indirect systems” contains updated data and a more detailed description of the properties of those fluids It will be distributed by IIR, Paris and the Swedish Society of Refrigeration.

4.7.1. What cooling capacity can be obtained with a certain volume flow?

The heat power or **cooling capacity**, $\dot{Q}_2$ [W] that a certain volume flow rate, $\dot{V}$ [m^3/s], can transport is given by

$$\dot{Q}_2 = \dot{V}(\rho \cdot c_p)\Delta t \qquad (4.1)$$

where $(\rho \cdot c_p)$ is the volumetric heat capacity of the fluid and Δt is the temperature change between in and outlets.



Figur 4.6 7 shows that the volumetric heat capacity, ρc_p [MJ/m^{3}K], for various secondary fluids with concentrations giving the freezing point t_F = -20°C. The different secondary fluids are well gathered. (For terms, see Fig. 4.1) Ethyl alcohol that is closest to water gives for a certain volume flow about 10-15% larger cooling capacity than ethylene glycol and most salt solutions do. Nearest ethyl alcohol is propylene glycol and potassium carbonate.

Figure 4.6. Volumetric heat capacity as function of the temperature

4.7.2. Heat transfer and pressure drop in a secondary fluid circuit

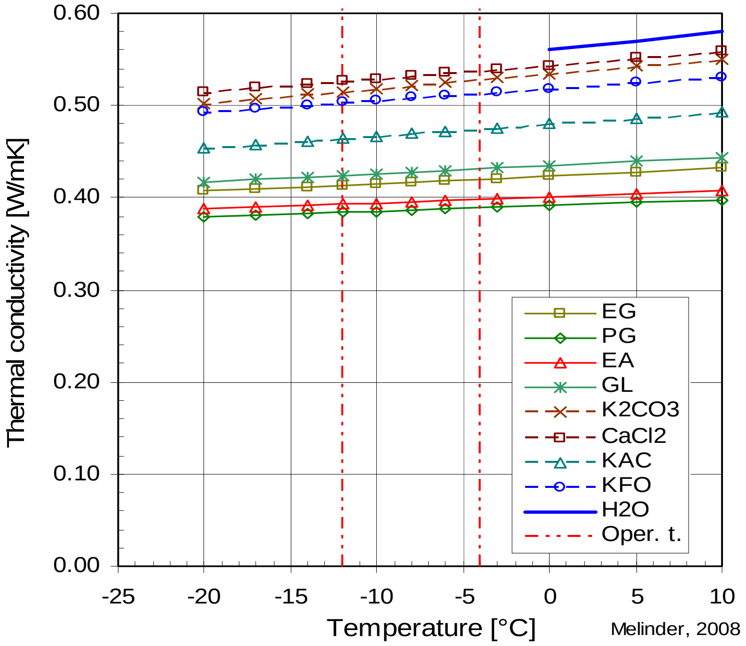


Figure 4.7. Thermal conductivity as function of temp.

High values of thermal conductivity and specific heat are desirable as it contributes to good heat transfer giving smaller temperature differences at evaporator and at cooling object or heat source.

Figure 4.7 shows thermal conductivity *k* för various secondary fluids with concentrations giving the freezing point t_F = -20°C. (For terms, see Fig. 4.1)

Most salt solutions have only a little lower values than water but about 20-30% higher values than glycols, ethyl alcohol and glycerol.

A high viscosity makes it impossible to keep the flow turbulent in a conventional heat exchanger and at transition to laminar flow impairing heat transfer so the temperature difference at the heat transfer wall surface becomes too large.

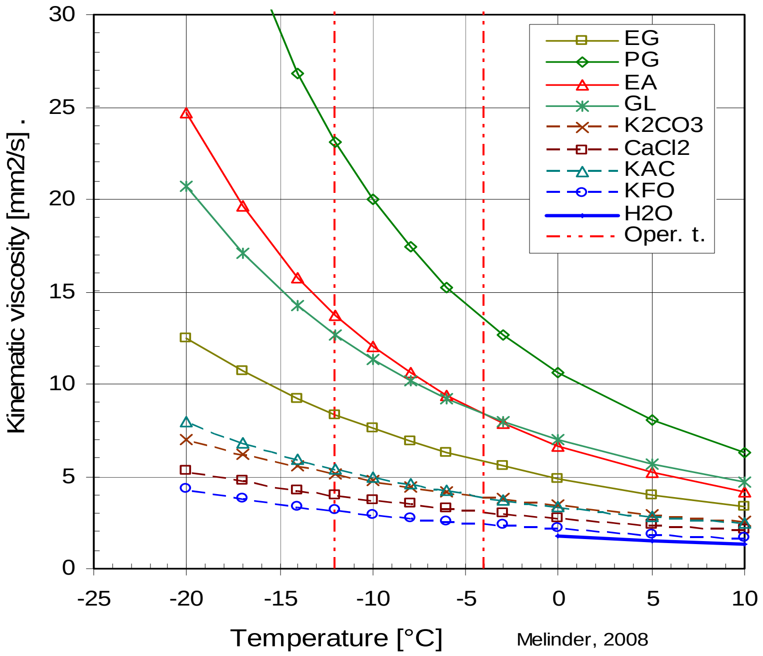


Figure 4.8 gives kinematic viscosity as function of the temperature.

The concentrations for the various water solutions have here been chosen to correspond to a freezing point t_F = -20°C. (For terms, see Fig. 4.1)

Note that the viscosity for propylene glycol at the operation temperature -10°C is almost three times as large as for ethylene glycol and 4-6 times the viscosity of the salts.

Figure 4.8. Kinematic viscosity as function of temperature

The viscosity is of special interest as ot stongly affect the type of flow in heat exchanging machinery as well as pressure drop at the flow intubes. It is important not to have unnecessarily high concentration of the freezing point depressant additive. Many troublesome operation disturbances, specially in heat pump technlogy, are connected with increased viscosity at lower temperatures giving laminar flow in the evaporator at the operation cases where there is low temperature in the heat source. (Melinder 1997, Melinder 2007).

4.8. Summary of various properties of the secondary fluid types

When deciding which secondary fluid to use in a plant one must consider thermophysical properties and other aspects of the fluids. The secondary fluid shall give desired freezing protection, be able to transport and transfer heat or (cold) and not give too large pumping power. The secondary fluid should not cause material problems in the form of corrosion. It is important that the products contain an inhibitor package that gives a good corrosion protection. It should be acceptable from an environmental aspect, non toxic and not dangerous to handle. All secondary fluid types have certain good properties but also one or more negative sides, as can be seen from the following list.

A number of different commercial secondary fluid products are marketed in various product names, which can the identification of the products more difficult. It is important to know what chemicals they contain and this should clearly be evident from the product information for the product in question. A number of commersial products of the various secondary fluid types on the Nordic market are here given within square brackets [...].

4.8.1. Water

It has very good thermophysical properties but freezes at or just below 0°C. In view of the good properties of water one should not use more of the freezing point depressant additives than what is necessary för a safe function. Water and the various water solutions are corrosive in the presence of oxygen if effective corrosion inhibitors are lacking.

4.8.2. Glycols

Only glycols are normally used at high temperatures, such as in solar collectors. Most commercial products have good corrosion protection as secondary fluids, but this is not true of glycol without inhibitors or products for the car industry. Formulated glycols are well developed products that have long lifespan and low total cost for a secondary system, thanks to acceptable thermophysical properties, low corrosion activity, low purchase cost and properties that are easy to handle. A limitation with glycols are applications with low work temperatures where the viscosity blecomes too high. From an environmental aspect the products are acceptable, but are not produced by renewable materials and can at great spillage form ecological and biological impact.

Ethylene glycol (EG). Rather good thermo-physical properties for cooling applications. Classed as health risk as EG doesn’t taste unpleasant, but is highly toxic to consume for humans. (Comp. Chap. 6) [Antifrogen N, Dowcal 10, Dowtherm SR-1, Dowtherm 4000, Glythermin 10, Glythermin NF, KB-MEG, Tyfocor, Zitrec MC, etc.]

Propylene glycol (PG). Not suitable for freezer temperatures; very high viscosity at low temperatures. Not classed as health risk as it is not as är toxic as EG for humans. [Antifrogen L, Dowcal 20, Dowcal N (food stuffs), Dowfrost, Glytherm 20, Glythermin L, KB-MPG, Tyfocor L, Zitrec LC, Zitrec FC (food stuffs), etc.].

4.8.3. Alcohols and some other organic secondary fluids

Ethyl alcohol (EA). Ethyl alcohol, ethanol, is classified as fire hazardous farlig from about 30 weight procent due to low flash point. Can not be used for freezer temperatures; high viscosity at low temperatures. Note that ii Sweden denaturing agents (10–20 % of substances such as isopropanol and n-butanol) which should affect their thermal properties. See also 4.7. [Brineol Bietanol, E-therm KBS Bio, Köldbäraretanol 95%, Svedol KBS Bio, Thermol, etc.]

Glycerol (GL). Very high viscosity at low temperatures. Swedish product is obtained from rape fields, viewed as environmentally friendly. [Biotherm, Vegocool].

Betain (BE). High viscosity at low temperatures. New Nordic product obtained from sugar beats, viewed as environmentally friendly. [Thermera].

4.8.4. Inorganic and organic salts

Chloride salts: Strongly corrosive in the presence of oxygen. (Corrosion inhibitors with cromates viewed as a health risk, especially inhalation during the mixing process). The freezing point curve is steep and the concentration sould not be too near the eutectic point.

Calcium chloride ($CaCl_2$). Good thermal properties with low viscosity even at low temperatures. Often used at ”artificially frozen” ice rinks [Brineguard, ”Kalciumklorid”].

Sodium chloride (NaCl). Can not be used below about –10°C. Eutectic temperature –21°C.

Potassium carbonate (K_2CO_3). Low viscosity. Corrosive with zink, soft solering and aluminum (if effective corrosion inhibitors are not used) due to high pH-value. Risk of damage at eye contact due to high pH-value. Protective glasses shall be used during deaeration. Eutectic temperature –37,5°C [Anticora, ”Pottaska”].

Organic potassium salts: Certain commercial products have rather high pH-value but lower than K_2CO_3. Protective glasses shall be used during deaeration. Corrosive against zinc, soft solder and possibly aluminum (if effective corrosion inhibitors are not used). Viewed as environmentally friendly. can be used for cooling- och freezer applications.

Potassium acetate (KAc). Rather good thermo physical properties [Antifrogen KA, Tyfoxit].

Potassium acetate/ Potassium formate (KAc/KFo) [Pekasol, Temper].
Potassium formate (KFo). Very good thermal properties with low viscosity even at low temperatures. [Antifrogen KF, Freezium, Hycool, Tyfoxit F].

Potassium formate/sodium propionate (KFo/NaPr) [Zitrec S].

4.8.5. Phase changing secondary fluids

Two phase media: Ice slurry based on various types of water solutions. Ethyl alcohol oand sodium chloride have mainly been used. For freezer temperatures potassium formate has been suggested. Properties, see above. (Melinder 2007).

Two phase carbon dioxide, CO_2, has come to use at freezer temperatures. Some information with data and properties for CO_2 as secondary two phase medium is given in the new IIR publication: ”Properties of secondary working fluids for indirect systems” by Åke Melinder, distributed by IIR och SKTF.

(Melinder 1997, Melinder 2003, Melinder 2007a)

5. Corrosion in secondary fluid systems

(AMA Nytt 2005; BFR R11:1986, Ignatowicz 2008, KBL4 2002, Melinder 2008)

5.1. Introduction

There is always a risk that some form of corrosion can occur in systems with water based secondary fluid. Corrosion can in certain case result in the possible holes in the tube walls, and corrosion products can also lead to damage of pump wheels and seals. By designing the systems to fit the chosen secondary fluid, the corrosion risk can be minimized or corrosion rate can be considerably reduced.

Corrosion processes are often of an electrochemical nature, which means that the reactions take place while releasing (oxidation reaction) or taking up (reduction reaction) electrons. In order for these types of corrosion to take place, electrons must easily be able to travel in between the places where these reactions take place. Metals are, as known, good electrical conductors, which is also why the metals have a great tendency to corrode.

In addition, it requires that the corrosive material is in contact with a conductive liquid. Liquids containing ions in solution are conductive, which is more or less true for all water based secondary fluids. Secondary fluid that consists of water mixtures of methanol or ethanol are the least conductive. After that come water mixtures of glycols, glycerol or urea. Most conductive are secondary fluids that consists of water solutions of salts, inorganic (i.e. calcium chloride, potassium carbonate) or organic salts (i. e. potassium formate, p. acetate).

It is also required that an oxidant is present for corrosion to take place. In secondary fluid systems the oxidant is almost exclusively oxygen dissolved in the liquid. Copper, for example, does not corrode at all if oxygen is not present and even the corrosion rate in steel can be reduced dramatically by removing oxygen from the system.

Therefore, it is very important to remove oxygen from the heat transfer fluid. First and foremost, it is necessary to remove the free oxygen gas contained in possible air bubbles in the liquid. This is done after filling the system through a high point air purges or micro-bubble separators. In addition, it is important that the bound oxygen, i.e. the oxygen dissolved in the liquid is removed. The higher the temperature and the lower pressure liquid has during air purging, the less oxygen can be dissolved in it. The bound oxygen can be removed with a sub-pressure air purge and by raising the temperature of liquid after charging so that the bound oxygen leaves the liquid. It is also important that the system has a certain pressure so that no new air is sucked in through gaskets in pumps or similar parts.

Common types of corrosion that may occur in the secondary fluid system are: general corrosion, galvanic corrosion, localized corrosion, such as pitting or crevice corrosion, flow-accelerated and erosion corrosion.

5.2. Types of corrosion

5.2.1. General corrosion

General corrosion occurs uniformly over the entire surface exposed to the conductive liquid. In certain cases, the corrosion process stagnates after a while as it forms a protective layer of corrosion products on the metal surface, which protects the metal from further corrosion - the metal is passivated. Different metals are passivated in different environments (type of liquid, pH, etc.) and sometimes the metal's ability to be passivated is improved by adding additional alloying elements to the metal. Examples of materials that are often passivated are: stainless steel, titanium, aluminum and copper. Since the pH of secondary fluid can determine whether a material is passivated or not, it is important to ensure that the secondary fluid has the right pH for the different materials present in the system.

General corrosion can be prevented or slowed down by adjusting the fluid environment. This may be done by adding substances to adjust the pH to an appropriate level or by adding so-called corrosion inhibitors. Corrosion inhibitors react with the metal surface so that it is passivated. Inhibitors inhibit the corrosion process by slowing down either reduction or oxidation reaction.

5.2.2. Galvanic corrosion

This type of corrosion will occur when two metals of different nobility come into contact in the presence of the conductive liquid. Thus, one should not link dissimilar metals that have a big difference in the corrosion potential. This depends as well partly on the fluid type. In systems with copper, being a noble metal, less noble metals such as zinc, magnesium and aluminum should therefore be avoided. This applies particularly for liquids with the high electrical conductivity.

Galvanic corrosion can also occur in an alloy composed of two metals with different nobility, e.g. dezincification of brass, where a selective release of zinc from the alloy occurs. What remains is a residue of porous copper having low corrosion resistance. Brass is commonly used for casting elements and pump wheels. Dezincification of brass can be avoided if the zinc content is not too high and the alloy also contains small amounts of other elements (arsenic, antimony, lead, aluminum and silicon). In secondary fluid systems only low zinc content alloys should be used.

While soldering the copper materials in secondary fluid so called brazing with silver solders should be used instead of soft solders containing tin. Thus, a special care should be taken regarding micro-bubble separators and pressure gauges (manometers), as these may contain soft solders.

5.2.3. Local corrosion

The examples of localized corrosion that may occur in the secondary fluid systems are crevice and pitting corrosion. Both types of corrosion arise when the oxidant (oxygen) concentration differs from place to place. The crevice corrosion is related to the stagnation of fluid motion, and the lower oxygen concentration compared to the other locations in the system. Due to the potential difference between two points the metal being in contact with the liquid starts to corrode. Metals that have the ability to form a protective layer on the surface are particularly vulnerable to pitting corrosion. The protective layer, which consists of various metal oxides, dissolves and pit formation starts. The same phenomenon can occur during deposition of the dirt particles that may occur in poorly cleaned system. Under the dirt particles the oxygen concentration can drop, leading to the increase in potential difference and corrosion. It is therefore important that after installation of the secondary fluid system cleaning it thoroughly before filling is advised.

5.2.4. Flow-accelerated and erosion corrosion

Some materials are sensitive to fluid flow. This is especially true for copper, which is a commonly used material in secondary fluid systems. The reason is that the corrosion products formed on the metal surface and creating the protective layer are water soluble. Increased flow velocities can wash away this protective layer, thus exposing bare metal surface and increasing the corrosion rate. If the fluid has a high electrical conductivity, which is the case of salt solutions, the protective oxide layer (corrosion products) may be porous and have lower adhesion to the metal surface, which can result in the risk of increases corrosion. Carbon steel and cast iron is also susceptible to erosion corrosion while titanium and stainless steel are completely insensitive.

Flow-accelerated and erosion corrosion is often used synonymously, but in the strict sense erosion corrosion implies to the presence of solid particles in the liquid. These particles by impingement of the metal surface or its protective oxide layer cause that the corrosion rate increases. Although, free air bubbles and cavitation can lead to the erosion corrosion and the risk increases with the increased liquid flow rate. Flow-accelerated and erosion corrosion often occur at the simultaneously as the first type of corrosion leads to the existence of solid particles in the liquid.

To avoid flow-accelerated and erosion corrosion, it is important that the secondary fluid system is very clean, vented and the flow rate is not too high. This applies particularly to heat exchangers having tubes with small dimensions and 180 ° bends at the ends. To keep the low content of particles in the liquid, a part of the fluid should be passed through a filter where they are removed.

5.3. Corrosion and inhibitors

5.3.1. Introduction

Corrosion processes have the electrochemical nature, thus two or more reactions take place either by releasing (oxidation) or consuming electrons (reduction). This requires that electrons can easily move between the locations where these reactions are taking place. The corrosion process is illustrated in Figure 5.1.

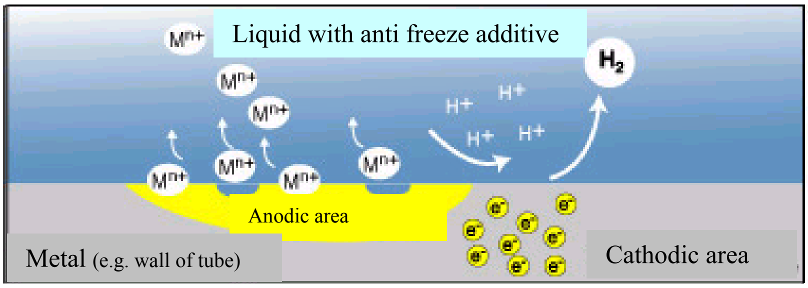


Figure 5.1. Corrosion process

Metals are good electrical conductors and have high tendency to corrode when they are in contact with an electrically conductive fluid. All water based secondary fluids contain ions in solution, thus leading to a greater or lesser electrical conductivity. The corrosion susceptibility of different types of secondary fluids varies widely. Salt solutions, both organic and inorganic, have significantly higher corrosion prone than other types of secondary fluids due to the fact that salt solutions have high electrical conductivity. Glycols and glycerin have lower corrosion tendency and alcohols have the lowest. Ethanol solutions are often sold without a corrosion inhibitor and can be more corrosive than the glycol solutions with a corrosion inhibitor.

A first stage of corrosion is the condition where the electrons in the cathodic area or metal atoms with a positive electrical potential accumulate in the anodic area before the electrons leave the metal surface. These shifts in the electrical potential are taking place in the material that is electrically conductive, and often do not initiate the corrosion process.

It also requires an oxidant is present for oxidation to take place. In secondary fluid system the dissolved oxygen in the fluid is the most common oxidant. The amount of oxygen that can be tied into the liquid is determined by pressure and temperature of the liquid. A high pressure increases the solubility of oxygen in the liquid. The dissolved oxygen in the liquid is removed more easily if the temperature of the liquid is increased. As the operating temperature of a secondary fluid system is relatively low, oxygen is rather tightly bounded to the liquid. The low temperature causes that the de-aeration process does not happen as spontaneously as in a heating system where the temperature is higher. Therefore, a secondary fluid with only water as cooling medium is more corrosion prone than a heating system with water only.

In order to reduce susceptibility to corrosion various inhibitors to secondary fluid are added. For secondary fluids which may be exposed to very high temperatures e.g. in the solar heating systems, inhibitors may have a temperature stabilizing ability. Otherwise, only inhibitors of the corrosion inhibiting type are used in the secondary fluids.

5.3.2. Corrosion inhibitors which build protective layer

Conventional corrosion inhibitors often form a protective film of a mechanical nature. This film prevents the transport of electrons and release of metal into the electrolyte, and works as long as the film is intact. Thus, the protective film can be damaged by mechanical stresses. This protective film also affects negatively the heat transfer between the metal wall and fluid. With time the traditional inhibitors can be consumed or broken down and it is necessary to refill system with a new secondary fluid and /or corrosion inhibitor. Thus, a new film is building up not only in the damaged areas but throughout the system, which further reduces the heat transfer. The function of the film-forming inhibitors is shown in Figure 5.2.

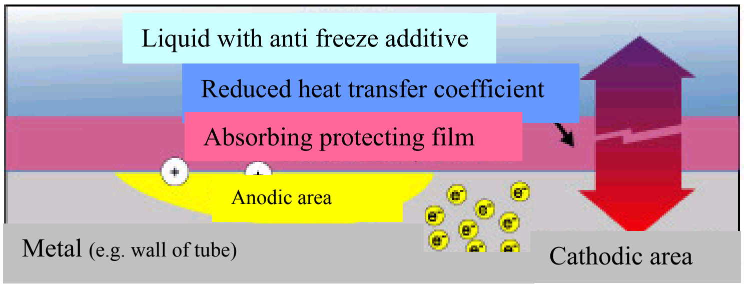


Figure 5.2. Film forming inhibitor

Other types of inhibitors are adsorbed on the metal surfaces due to chemical interactions already in pre-corrosion stage and produce a local temporary protective film. When this electric potential difference is balanced, the corrosion inhibitors return to the liquid and thus are not consumed.

5.3.3. Inhibitor composition

Det är svårt att få reda på vad inhibitorerna består av eftersom inhibitorsammansättningen It is difficult to know what the composition of inhibitor mixture is due to the fact that usually the commercial inhibitor name is the base to distinguish different brands of the same product from another. Inhibitors are usually only a small proportion (<<5%) of the secondary fluid and will usually need not be declared unless the substance are to be labeled under the chemicals legislation. They can therefore be a trade secret. Thus, everything is depended on the manufacturer in case of controlling the concentration of secondary fluids inhibitors, because no one exactly knows what it should contain and whether any of them is consumed.

Important: Therefore, it is important that the secondary fluid in the system is documented and listed on a sign at the filling site or at another suitable location and a container with a reference sample of the original filling liquid is stored.

Inhibitors may include products that are environmentally harmful and can negatively affect the sewage purification processes in the treatment plants. Due to the above reasons and the fact that often secondary fluid contains a high metal concentrations after having circulated in the pipelines made of metal, the consumed secondary fluids is not poured down into the drain.

Effect of inhibitors is covered more in further chapters. In Chapter 8, charging of secondary fluid system and system starting up is described. Chapter 9 deals with control of secondary fluids quality and of product characteristics. Analysis of the corrosion condition of the secondary fluid system and the importance of not only taking care about the symptoms is raised.

5.4. Tips how to avoid different types of corrosion

Select and assemble materials having metal composition as homogeneous as possible to avoid the galvanic corrosion. Please select a higher quality of steel or metal. Higher grades may be less susceptible to corrosion and thus has a longer life time.

Follow the fluid and other material supplier’s recommendations. Keep in mind that some materials are not suitable from the temperature point of view.

Construct and assemble the system so that it can easily be filled, vented and maintained. The presence of oxygen favors always the corrosion processes. High and local pressure drop can create air inlet passage /presence of oxygen and can be avoided by selecting the correct pipe sizes.

At the right flow rate the risk of erosion corrosion is reduced in the system (internal wear of the pipe wall). Contamination of the liquid with metal ions and corrosion residues can cause damage in the shaft seals in pumps.

5.5. Factors that affect corrosion

The aqueous solution in contact with common metals e.g. steel, copper, copper alloys, aluminum, etc. can cause corrosion. The corrosion rate depends particularly on: the oxygen availability, temperature, pH, electrical conductivity, material selection, salinity, particle presence, flow velocity etc. Inappropriate metal combinations give rise to the galvanic corrosion. Below is presented a brief explanation of the various factors of importance.

Oxygen: Generally, the corrosion rate increases with increasing oxygen content. A closed system is thus more favorable than an open one. If the system includes elements such as plastic tubing oxygen can diffuse through the plastic material and cause corrosion. The rate of oxygen transfer is likely to depend largely on the oxygen content in the tubes around and usually is relatively low.

Temperature: High temperatures give higher corrosion rate. Thus, secondary fluid systems will benefit from the low temperature. At the same time, however, the oxygen solubility increases with lowering temperature which is important in systems with oxygen access. In the solar panels system where the operating temperatures are higher, the de-aeration of the fluid is easier and the oxygen content lower.

Electrical conductivity: Galvanic corrosion will occur when two different metals of nobility are linked together in a liquid with high electrical conductivity. One should not link the material that has a big difference in the corrosion potential.

Salinity: High salinity results in higher corrosion rate. Of particular importance is the chloride ion concentration.

pH: A key factor in the corrosion rate is pH, the solution acidity. A pH value of 7 is representing a neutral solution. Corrosion increases generally with decreasing pH, but also a very high pH (highly alkaline) may cause corrosion of some metals.

Particulates: Particulates cause localized corrosion and increases the corrosion problems.

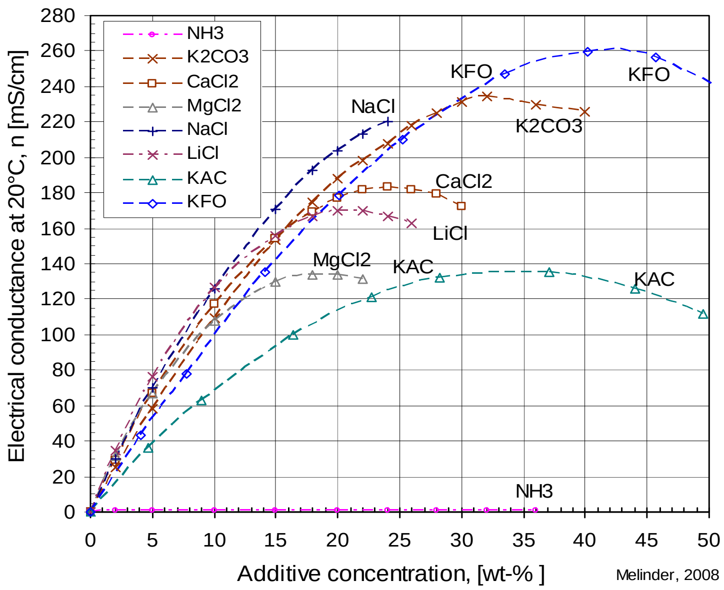


Figure 5.3. Electrical conductance of water solutions

Figure 5.3 shows the electrical or specific conductivity of the different types of aqueous solutions used as secondary fluids. This electrical conductivity or conductance strongly contributes to corrosion.

Only salt solutions have high values (ammonia like pure water have values close to 0). Note that the salt solutions with high additive concentrations are used in the refrigeration applications. Potassium acetate, KAC, has the lowest specific conductance values of these salt solutions (Melinder 2008).

A fluid's acidity or alkalinity is measured in pH scale and should be between 7 and 9. A pH of 7 means that the liquid is neutral. pH lower than 7 means that the solution is acidic and a value higher than 7 means that it is alkaline.
It recommended that the pH is tested every year as a pH lower than 7 to 9, may lead to the formation of the degradation products. On the other hand, a high pH, well above the 9, makes the liquid corrosive toward zinc, aluminum and even soft solders, if effective corrosion inhibitors are not used. There is also a risk of damage of skin and eyes, especially if they come into contact with liquid or spray droplets from the ventilation system.

Tabell 5.1 summarizes the pH values of aqueous solutions at +20 °C (Melinder 2008).

Tabell 5.1. pH-värden för vattenlösningar vid +20°C.

Type of additive	pH-värde
Ethylene glycol, EG	6 - 7.5
Propylene glycol, PG	6.5 - 8
Ethyl alcohol, EA	7
Glycerin, GL	≈ 5 - 7
Ammonia, NH_3 (1.7 %)	11.6
Potassium carbonate, K_2CO_3(50g	11.5–12.5
Calcium chloride, $CaCl_2$	8 – 10
Sodium chloride, NaCl	4.5 – 7
Potassium acetate, KAC (50g)	7-8.5
Potassium formate, KFO (7.5 %)	7-8.5

(Values are for 100 grams of additive per litre of water, unless indicated in the table).

5.6. Various types of inhibitors

Adding a corrosion inhibitor to secondary fluid prevents one or more mechanisms that can imitate the corrosion. This section provides a brief overview of several types of corrosion inhibitors and inhibitor mixtures that can be used in certain types of secondary fluids, such as ground heat pump system.

There are two important characteristics to consider when using inhibitor mixtures. One is the health and environmental aspects of two corrosion inhibitors (belonging to the so-called anodic inhibitors group), which cause an accelerated corrosion if they are added in too low concentration or consumed without being replaced. Nitrite and chromates inhibitors should be avoided from the environmental point of view in this kind of system.

Some of the anodic inhibitors that must be kept in a sufficiently high concentration are: nitrites, silicates and orthophosphates. In these cases where concentrated secondary fluids already containing the corrosion inhibitor package need to be diluted, it is important to observe the recommended concentrations of the corrosion inhibitors in order to fulfill the minimum levels.

There are several inhibitors that are used in so-called neutral solutions, ie. solutions having a pH of about 7- 9. This pH range is suitable for the secondary fluids normally used. Examples of such inhibitors are: nitrites, benzoates, borates, phosphates, silicates, chromates, benzotriazole, benzothiazole, amines, aldehydes, etc.

The following part provides a brief overview of a number of different inhibitor’s function, composition and finally gives examples of inhibitor mixtures as reported in the literature.
Borates: Salts of boron are used preliminary as a pH stabilizing agents. They have a high buffering capacity, meaning that they can neutralize large quantities of acidic or alkaline

additives without a significant pH change. The most commonly used is sodium tetraborate dehydrate often called ‘borax’. With borax as a base, a number of effective inhibitor mixtures can be produced which are effective for different combinations of metals.

Nitrite: Sodium nitrite alone is very effective corrosion inhibitor for steels and cast irons, but attacks tin solders. However, together with sodium benzoate a good protection is obtained for a number of alloys including tin.

Benzoate: has the advantage that it does not form insoluble compounds, like for example silicates, phosphates and carbonates. This reduces the risk of localized corrosion due to presence of contaminants. Sodium benzoate has been used for a long time as corrosion inhibitor for steel and a number of non-ferrous metals. Good corrosion inhibiting effect of sodium benzoate on steel requires certain oxygen content in the solution, content increases with lower pH.

Phosphates: Phosphates are often used as corrosion inhibitors in boilers and pipes. It is mainly due to the pH stabilizing effect of phosphates providing the corrosion protection of steels.

Silicates: In order to ensure successful use of silicates as corrosion inhibitors a good knowledge about these colloidal complexes is required. Corrosion inhibitor effect is highly dependent on pH, temperature and salt concentration in the solution. Silicates protect steels, lead, tin and aluminum against corrosion.

Chromates: Chromates protect all major metals against corrosion. Before nitrobenzoates became known as good corrosion inhibitors for various metals, chromates were the only inhibitors which protected a large number of metals (including iron, steel and copper). Water solutions having low salinity require approximately 0.2 to 0.3 weight-% sodium chromate, Na_2CrO_4, to obtain protection against corrosion. Calcium chloride, $CaCl_2$ solution of 20-30% are very aggressive and require approximately 2 g /l chromates. If aluminum is used in the system, it requires up to 10 g /l chromates. NOTE: Chromates are considered dangerous to health (carcinogenic), which strongly limits their use today!

Benzothiazole: Benzothiazole is a copper corrosion inhibitor which is added to certain inhibitor mixtures.

Benzotriazole: Benzotriazole can replace benzothiazole as a copper inhibitor in inhibitor mixture. A copper inhibitor is often necessary for copper and aluminums contained in the system. If small amounts of copper are dissolved, there is a risk that copper particles precipitates on aluminum surfaces and starts rapid corrosion attack on aluminum surfaces.

(BFR R11:1986, Ignatowicz 2008)

5.7. Corrosivity and various types of secondary fluid

5.7.1. Introduction

There are several ways to reduce corrosion. One way is to remove dissolved oxygen from the system, because oxygen is one of factors initiating corrosion. However, it is not possible completely to avoid the oxygen diffusion through the plastic tubing which is included in most secondary fluid systems for ground source heating purposes.

The most common types of cooling-and heating secondary fluids are glycols, ethanol and salts. From a corrosion point of view, there is a big difference between salt solutions and other organic secondary fluids. The salts solutions are generally the most prone to corrosion, especially due to the high electrical conductivity (see Chapter 4).

It is strongly recommended that each site has an "identity card" that clearly states the type of product being used, antifreezes, concentration, charging dates and maintenance. One important reason is that different mixtures of corrosion inhibitors may have undesirable effects when mixed together such as precipitation and / or gel formation, which negatively affects or completely prevents the fluid circulation.

5.7.2. Glycols

It is essential that the products contain an inhibitor package which provides good corrosion protection. Consequently not pure ethylene glycol and propylene glycol are used, but products containing one inhibitor package for the current application. In most cases, these products have "Brand names' which can complicate the chemical identification of the product. In addition, most of them are dyed in different colors.

In case of glycol solutions the corrosion potential is comparatively the same as for the water. This means that basically the same construction materials can be used as for fresh water systems. After a long period of time glycols can become slightly acidic due to oxidation or degradation of glycol which makes fluid more corrosive. Typically corrosion inhibitors are added to stabilize pH and avoid corrosion. Normally, the pH of a glycol solution should be around 8.

The most common glycol solutions used in cars are usually not suitable for indirect cooling and heat pump systems. Corrosion inhibitor varies both in quantity and quality and can therefore be directly harmful to pumps, seals, and structural components of the system if made from not compatible materials.

Glycols with appropriate corrosion inhibitors exhibit generally lower corrosion activity than e.g. organic and inorganic salt solutions. This fact is confirmed by ASTM 1834 tests reported in the product data sheets and technical specifications.

The low electrical conductivity (conductance) and well-chosen anti-corrosion packages are the key factors to extend the life time of a glycol system. In order to achieve maximum life time to the system and secondary fluid it should be noted that:

Great emphasis is placed on the proper material selection and system purity. Corrosion residues and other contaminants can act as abrasive particles (erosion) and can affect the protective properties of corrosion inhibitors. Fit always a filter in the system.

Soldering is performed with so-called silver or hard-solders. Brass parts must be of high quality so-called ‘not bleeding’ zinc. Avoid designing the system from different materials that can cause the galvanic corrosion.

Packing material must be made of the so-called oil-resistant material. Your supplier should be able to provide information about the appropriate materials. This also applies to the pump supplier. Zinc is highly influenced by glycols and salt solutions. Never use galvanized materials in direct contact with salt solution.

Although secondary galvanized sheets may be affected e.g. liquid-lubricated pumps, drains, taps, etc. Painted surfaces or painted zinc plate provides significantly better corrosion protection.

Water dilution: It is safer to use deionized water, but municipal tap water is often perfectly acceptable to use as dilution water in a glycol solution. Typically, a municipal tap water is adjusted to protect the municipality's system from corrosion. In municipalities with very hard water> 8 ° dH (German hardness) water should be selected from another municipality or deionized water.

De-mineralized water is much more sensitive to changes than a normal tap water, e.g. lack of buffering capacity, which may cause sudden pH decrease leading to the corrosion. Using so-called deaerated water is not necessary for the dilution of glycols. The de-aeration water process is expensive, sensitive and insecure. Additionally, the water usually takes up oxygen very rapidly after the process.

Limitations of a glycol solution are usually: chlorine, sulphates and water hardness. Chloride ions and sulfate ions increase the electrical conductivity (conductance), thus initiate corrosion. High calcium content can result in deposition of highly insoluble calcium compounds, which affect the fluid circulation, inhibitor functions and corrosion /erosion. Your manufacturer has to indicate maximum values.

For the so-called FDS / NSF approved propylene based glycols deionized / demineralized water is always used. Due to the certification issues, producers have been forced to replace or remove inhibitors that provide protection for certain metals, increase the buffer capacity and keeps calcium precipitates in the dispersion.

Important: Do not dilute a formulated glycol more than 33% by volume. At lower concentration glycol loses its growth-inhibiting property and enhances instead the organic growth. In addition, the corrosion resistance is reduced significantly because all inhibitors are diluted to the same degree. If there is need for the weaker solutions, consult your supplier. For some glycols the inhibitor concentrates can be added to avoid organic growth and provide the corrosion protection even for weaker solutions.

5.7.3. Alcohols

As mentioned earlier, alcohols have higher corrosion tendency than glycols. As mentioned ethanol solutions sold without corrosion inhibitor can be more corrosive than the glycol solutions containing the corrosion inhibitors. Since the water content of the diluted product is relatively high likelihood of corrosion can occur due to the incorrect material selection.

5.7.4. Salts

Salts are generally more corrosion prone than other types of secondary fluid, which must be considered during the material selection and maintenance. The advantage of the salts is the low viscosity, especially at low temperatures.

General: Both organic and inorganic salts have a substantially higher conductivity than e.g. glycols and alcohols. A generally higher corrosion activity is observed in the salt solutions. Therefore care must be taken in case of an appropriate choice of materials, construction and the operating temperature range.

Salt solutions can be divided into inorganic and organic salts. The most common inorganic salts are calcium chloride and potassium carbonate, while the organic salts are potassium acetate and potassium, and mixtures of these two types of salt.

Chloride-based solutions, for example, solutions of potassium chloride are highly corrosive, especially in case of carbon steels and requires buffering (to pH of about 7.5 to 8.5) and the addition of at least two powerful inhibitors, which are often less environmentally friendly. Chromates used as corrosion inhibitors constitute health hazard, especially during the mixing process. For chloride solutions the same design and materials should be used as for sea-water constructions.

Potassium carbonate has very high pH (11-12) and is corrosive to zinc, aluminum and even soft solder. This applies even if the organic salts have a high pH (>> 9).

As mentioned previously, both organic and inorganic salts have substantially higher conductivity than glycols and alcohols (see Figure 5.3). Products with high pH value also give the risk of damage to skin and eyes, especially if they come into contact with liquid or spray droplets from the ventilation system.

If the system contains galvanized surfaces, they can be eaten if they come into contact with salt solutions with higher pH. Corrosion of aluminum can be reduced by adding appropriate corrosion inhibitors. Various commercial products of the same basic type of salt may have different inhibitor package and behave in different ways. Therefore, care must be taken while choosing the construction materials, designing and the system’s operating temperature range.

Organic potassium salts: Products of organic potassium salts, potassium acetate and potassium formate, and mixtures thereof, have lower pH than potassium carbonate but still have relatively high pH and can attack any material that contains zinc or aluminum.

5.7.5. Ammonia

High concentrations, 25-30% NH_3 in water, is highly corrosive and can cause burns. It is not compatible with copper or elastomers from 100% soft rubbers, isoprene, natural rubbers, polyurethane and silicone rubbers (Schweitzer 1996). But the (pure) ammonia is not corrosive to steel. If anhydrous ammonia is used even aluminum can be applied.

(BFR R11:1986, Ignatowicz 2008, KBL4 2002, Hägg,Melinder 2002, Melinder 2008)

6. Environmental aspects at secondary fluid systems

(D. Kroon; M. Ignatowicz, Å. Melinder)

6.1. Environmental effects of secondary fluids

6.1.1. Introduction

The chemicals that commonly are used as secondary fluids on the cold and warm sides do not pose any serious threat to our environment. They mainly consist of inorganic salts or organic substances that rather easily are broken down to carbon dioxide and water. However, the various products can have effects that on a short-term basis can affect our ambient environment. If for instance secondary fluids unintentionally are released in a watercourse they can locally cause lack of oxygen. The products can also containa low contants of additives, for instance corrosion inhibitors that do not always break down as easily. In view of this you should minimise the risk for leakage especially to ground water as the prerequisites for break down is worse there.

It is however important to understand possible health risks and the damage on environment and ecology that can be caused by a leakage eor incorrect handling of secondary fluids. In order to evaluate possible influence of such an event it is important to aquire knowledge of poisonousness or toxicity, degradability, stability and influence of the fluid used in a system.

The biological environmental effects of secondary fluids are connected to the following properties: toxicity, degradability, movability, concentration in organisms, products at degradation effects at degradation. The greatest risk of unwanted environmental effects are likely with heat pump plants. With biological effects is meant environmental, health and ecological effects that can appear during handeling and leakage of secondary fluids. The effects due to temperature decrease in ground or water during heat withdrawal is not treated here.

When dealing with secondary fluids, there is a need to understand direct toxic effect that can cause poisoning of humans and contamination of ground water as well as degradability and the effect the fluid has on the organisms in the ground. Here follows a short description. Data of toxic effect and degradability is given in later sections.

6.1.2. Direct toxic effect

Direct toxic effect (toxicity) is usually given as an LD_{50} (Lethal Dose) value, as LD_{50} oral rat, and refers to the amount of the chemical that is added to a test animal per kilo body weight that results in the death of 50% of the test animals that are exposed for the substance. A high LD_{50} value corresponds to a low toxicity for the substance. Substances with higher LD_{50} values than 2000 mg/kg are not classified as damaging. Note that the mortal dose of concentrated ethylene glycol is as litttle as ≤ 1.5 g/kg or ≈ 100 ml for an adult person. This must be considered whenthere is risk for unintentional consumtion (Compare also 6.4 with Table 6.2).

LD_{50} values given by manufacturers have often been taken from literature for pure substances and may not include effects of inhibitors and other additives. We can inot expect that a commercial product will have bätter values than the basic product.

6.1.3. Degradability

The degradability of a fluid depends on many factors, such as temperature, type and pH-value of soil or ground, water content and oxidisation potential. Degradability in the pressence of oxygen (areobic), happens much faster than when there si lack of oxygen as the oxidisation potential is much higher. Degradability is usually given as a value of oxygen consumption (oxygen demand), or theoretical oxygen consumption, ThOD, or biochemical oxygen consumption, BOD. A higher value (of ThOD or BOD) requires more oxygen for degradation of the substance that then will stay longer in the ground. A BOD_5 value specifies the amount of oxygen that is consumed by micro organisms during 5 days at +20°C. Products that result from the degradation process can give harmfula effects, that ay be different from the effects of the original materials. Due tothis fact it can be difficult to fully appreciate the environmental effect of a fluid. Values given are based on material influence on humans and animals but may not include influence on ground and other organisms.

(Aittomäki 2003, BFR R11:1986, Hägg 2002, Liu 2008, Melinder 2008).

6.2. Environmental effects at heat pump plants

6.2.1. Introduction

Most technologies for heat extraction from ground, rock, water and air are viewed as a very good alternative with regard to its environmental effect. This does not mean that there is no risk at all when such plants are installed and used. The most serious effect of a secondary fluid leakage is that water catchments and water wells become contaminated with the secondary fluid and that humans or animals then use the water. When making an installation it is important to do what is possible in order to prevent future leakage of secondary fluid to nature and ground. If this is done in a correct way then the risks will be minimal.

Secondary fluid used in heat pump plants today usually consist of water and some type of freezing protection additive, such as ethylene glycol, propylene glycol, ethanol, methanol, calcium chloride or other salt solution. Besides freezing protection the secondary fluids may contain substances for corrosion prevention, bacterial inhibition and foam reduction etc. Since the properties of the various secondary fluid types vary it is of utmost importance that right type of fluid is chosen to fit the plant.

The Swedish Environmental Protection Agency, SNV, has published several reports that partly deals with indirect systems with secondary fluids: ”Värmeutvinning ur mark, vatten och luft” (SNV pm 1833), year 1984, ”Värmepumpar & närmiljön”, report 4444, year 1995, ”Villavärmepumpar med energi från ytjord/energibrunn”, report 4994, year 1999 (SNV 1984, SNV 1995, SNV 1999).

6.2.2. Toxicity

General

The most serious that can happen at a secondary fluid leakage is that water catchments and water wells become contaminated with the secondary fluid and that humans or animals after that use the water. Because of this it is important to clarify how poisonous the different secondary fluids are. In BFR Report R11:1986 that also refer to other reports, a few common secondary fluid types, propylene glycol, ethylene glycol, ethanol och sodium chloride, have been compared and tests have been made of how poisonous each substance is. Even a few occur additives have been included in the study.

LD_{50}-values are used to be able to compare the results (Comp. 6.1.2). The study showed that propylene glycol vas the substance that was least toxic with an LD_{50}-value of 33 g/kg. For ethanol (ethyl alcohol) the corresponding figure was 13.7 g/kg, while the value for ethylene glycol only was 5.5 – 8.5 g/kg which makes this secondary fluid the most poisonous in the study among glycols och ethanol. The LD_{50}-value of sodium chloride was only 3 g/kg (but we all know how hard it is to even gurgle with a little table salt in water).

For additives the resultat showed that 2.9 weight-% triethanolamin had an LD_{50}-value of 9.11 g/kg and corresponding value for 5 weight-% sodiumbensoat was 4.1 g/kg. Sodium nitrit (0.5 weight-%) that was most toxic had an LD_{50}-value 0.33 g/kg. Sodiumnitrit is also classified as poison according to SNFS 1978:5 PK:6. Conclusions that can be drawn are that the additive chemicals are of great importance for the toxicity.

If we assume that a normal size villa plant gets a secondary fluid leakage in a bore hole where the whole volume (about 400 l) fluid leaks out in the surrounding bore hole and the same bore hole is also used as cold water source, you can calculate how toxic the water becomes. A solution with 25 weight-% ethanol should mean that the water in the bore hole gets a concentration of about 5 volume-% ethanol. Deadly dosis for an adult is 300 ml within an hour and this should mean that a grown up need to drink 6 liters aof the polluted water within an hour. The risk for poisoning with fatal result would be neglible.

If the same assumption is done where the secondary fluid is 30 weight-% ethylene glycol the risk for poisoning is much larger. The leakage results in about 6 volume-% ethylene glycol mixed in water. The deadly dosis given is 1.4 ml/kg, which should mean that a child of 20 kg could be able to die if it should drink 0.5 liter of the water and get agonizing symptos at somewhat smaller dosis. Already half a deciliter concentrated ethylene glycol can lead to life threatening poisoning of a child, In view of the fact that ethylene glycol is sweet, this situation must be viewed as the worst possible that can occur from a poisoning aspect. Table 6.2 gives LD_{50} values also for other types of water solutions. (BFR R11:1986)

Taste

One thing that has not been treated in connection with the previous part about toxicity, is that glycol and alcohol polluted water gives taste already vat low concentrations. This makes the risk for poisoning via the cold water less likely. A taste test carried out at SNV, the Swedish Environmental Protection Agency, (Dietrichson 1983) showed that propylene glycol and ethylene glycol clearly could be felt at a concentration of 1 % but was harder to discern at 0.1 %. For calcium chloride and wood alcohol the presence was clearly noticable already at 0.1 % but was harder to detect at 0.01 %.

The two mentioned glycols are clear, almost odourless liquids but with a slightly sweet taste. Pure ethanol is a clear, colourless liquid with a typical pleasant smell and taste. Ethanol is often denatured during the manufacturing process which gives it a very unpleasant taste already at low concentrations. The freezing protecting agents used in collector systems are usually coloured to be easier distinguished. In this way a leakage may easier be detected. (BFR R11:1986)

6.2.3. Degradability

If secondary fluid through leakage should come to surrounding ground, rock or lake, this liquid will sooner or later be broken down or degraded. However, many factors influence how quick these substances degrade, which causes the half-value period for degradation can be anything from a few days up to 3- 4 months and more. Some factors that affect the degradation speed. (BFR R11:1986)

Temperature dependence

The degradation speed is temperature dependence. Degradation tests are normally performed at room temperature, i.e. at +20°C. The degradation speed at this temperature is considerably quicker than at a real case for heat extraction, where the temperature is around 3 – 6°C. The micro biological activity decrease with the temperature which causes the degradation speed to decrease much at lower temperatures.

Oxygen dependence

Even the access to oxygen has a large influence on the degradation speed of the secondary fluid. During degradation of glycols and alcohols consume large amounts of oxygen. In order to completely break down one litre ethylene glycol 1.43 kg oxygen is required. For propylene glycol and ethanol the corresponding values are 1.73 and 1.64 kg oxygen respectively (BFR Report R28:1985).

During aerobic processes (good oxygen supply) the micro biological activity is high and degradation can take place faster. From the ground surface down to 1m the access of oxygen is usually good, but even thi region can be affected if heat withdrawal takes place and the temperature in the ground is lowered. The access of earthworms is an important factor of how porous and oxygen rich the ground is and at lower ground temperature the supply of worms is less leading also to lower oxygen levels in the ground. The deeper in the ground you get the lower oxygen level, which is the case for rock heat pumps and deep ground heat plants. At these anaerobic conditions, the micro biological activity is inhibited and the degradation speed is lowered. The later case can also mean secondary effects as iron and manganese can deposit in ground water due to lack of oxygen. During micro biological activity with low oxygen hydrogen sulphide which has a bad smelling character is formed and may pollute the ground water and near by situated water catchments.

Concentration dependence

It has also bcome evident that the concentration of the secondary fluid affects the degradation speed, as high concentrations can ”knock” micro organisms that are active at the degradation.

Type of ground

The surrounding type of ground is also a factor that affects the degradation speed. If the ground is rich in water the degradation is hastened compared to a dry ground. The pH-value in the ground affects the micro organisms that in turn affect the degradation speed, causing certain soil types to break down the secondary fluid faster than other.

Oxygen consumption

A great amount of oxygen is as earlier mentioned used to break down glycols and alcohols. This can cause certain zones in the ground to lack oxygen. The resultof this is that iron and manganese can deposit in the water, as lack of oxygen causes these minerales to not be bound in oxide form. The result of this can be that surrounding water catchments and ground water is discoloured and becomes brownish and that the water quality is affected. Another effect that can affect the water quality is the hydrogen sulphide that is formed during degradation with lack of oxygen of glycols and alcohols. Bad smel and taste can make the water useless and this problem may remain for several months after the secondary fluid has actuallt degraded.

Mobility

As all antifreeze liquids use water as base, they are of course miscible with water. It has been noticed that the secondary fluids have low tendency to be absorberd to soil (Torstensson 1983) which indicates that they are att de är mobile in rock and ground where they follow the moements of the water. A leakage in a surface ground source heat pump plant should be less sensitive than a rock fray plant, as the ground collector often is above the groundwater level and hence must pass several layers of soil before the water is reached. Much of the fluid will degrade on the way down to the groundwater. If a leakage happens with a rock heat pump collector the secondary fluid is certainly rather quickly diluted leading to faster degradation but at the same time there is risk that nearby water catchments are polluted, as the dispersion takes place fast. For a collector in the laek the risks are minimal, as the dilution and degradation takes place at once. (BFR Report R11:1986)

6.2.4. Security functions

As mentioned above the installation of the collector should take place in a way that minimizes the risks for future leakage. If instructions and valid norms are followed the risks of larger leakage are minimal. The collector must as an example always be test pressured before the plant is filled with secondary fluid.

With a rock heat pump plant it is important to ensure that the collector bottom weight with u-tube is weld together according to the regulations that apply to avoid leakage in the joint splices. The same is valid for any other splices. For horisontal feed pipes to the till rock heat pump collector as well as for a common surface ground source collector the collector should be covered with fine sand before other materials are filled in. In this way the risk that sharp stones lies and frays on the collector with leakage as result.

If a leakage anyhow appears most heat pump manufacturers nowadays have security functions that minimize the leakage. As a direct function there are liquid level switches to the secondary fluid container that breaks the heat pump and warns if the liquid level is below a certain level. Often the leakage is then only a few litres.

There are also other similar solutions with pressure guards that break the heat pump and warn when the pressure becomes too low in the secondary fluid circuit. Also the low pressure gauge in the secondary fluid circuit has an indirect funktion that limits the secondary fluid leakage. Often the heat pump is situated at the highest point in the secondary fluid circuit and as a result the secondary fluid pump, when there is a leakage, in time can not pump around the liquid. The evaporation temperature will then decrease and the low pressure gauge heat pump will stop the compressor and pumps.

6.3. Why not pour used secondary fluid in the drain?

6.3.1. Cleaning of drainage water

The municipal sewage plants in Sweden accept sewage water from both households as well as public and industrial activities. The water is cleaned in several steps in order to pmove among else manures (phosphorus and nitrogen) and organic material so they can be released without damage to lakes or sea. An important part of the cleaning process is the biological stege where micro organisms, above all various types of bacteria, breakes down organic material and convert nitrogen to nitrogen gas. Micro organisms are sensitive to poison and especially sensitive are the bacterias that takes care of the nitrogen cleaning. During the cleaning process a sludge is formed that is a good fertilizer for arable land unless it is not polluted by metalls or other substances that are harmful to the environment.

Stockholm Vatten works intensively to reduce incoming amounts of substances and chemicals that can disturb the cleaning process, reduce the quality of the sludge or affect the seas and lakes negatively as they can not be cleansed away.

No secondary fluid may be let out in the drain without special permission by the community. Secondary fluids are classified as waste harmful to the environment and shall be treated in in line with appkicable statute. If leakage takes place all grocery that has been contaminated shall be destroyed, regardless of what secondary fluid that was used. The risk for accidents and harmful effects can be limited if the plants include as harmless products as possible. If a great amount of secondary fluid by mistake is released inthe drain then the local water treatment plant should be contactaced as there is risk for influence on the biological cleaning process.

6.3.2. High metal levels

Water that during several years has circulated in tube systems of metals has released metas from the tubes and can contain rather high content. Often the so called varning values that are administered by Stockholm Vatten are exceeded. If the tubes are of copper the copper levels in the water may become too high, lead can be released from brass alloys, etc. Table 6.1 shows warning values for industrial drain water that the treatment plants in Stockholm receives. The table applies in most kommunities i the Stockholm county, Gothenburg and Malmoe. Other communities may have other regulations.

Parameter	**Warning value**
Lead (Pb)	0,05 mg/l
Cadmium (Cd)	must not occur
Copper (Cu)	0,2 mg/l
Chrome total (Cr)	0,05 mg/l
Mercury (Hg)	must not occur
Nickel (Ni)	0,05 mg/l
Silver (Ag)	0,05 mg/l
Zinc (Zn)	0,2 mg/l

Table 6.1. Warning values for industrial drain water

6.3.3. Inhibition of nitrogen cleaning

Water that contains antifreeze solutions as glycol and various salts has also proved to be impeding of the bacteria that cleanses the drain water from nitrogen. Partly this can be additives of corrosion inhibitors that have this negative impact. When tests of secondary fluid are made for influence on nitrogen cleansing has very high inhibition been obtained, regardless of type of secondary fluid that is tested.

6.3.4. Waste handling

Used secondary fluids shall be treated as industrial waste and may at least not in Stockholm. The water catchment area not be poured out in the drain unless it can be shown that warning values for metals are cleared and the water is not nitrification impeded. If so this should be clear from the analysis protocol. The water shall instead be taken care of by a company that works with waste and rest product handling and that can treat such waste in a satisfactory way. From July 1, 2002 it became forbidden to pour water in landfills and in view of that waste companies had to develop new methods for water cleaning that shall not go to the drain. A cycle adjusted method that already today is applied is recycling of glycol from water with glycol content.

6.3.5. Environmental aspects and glycols (B. Bredberg.; BFR R11:1986)

Ethylen and propylene glycol are produced in so called petrochemical industries, i.e. they have their origin in crude oil. The production takes place in i so called ”cracking” processes in combination with oxidation processes and is a part of a stream of products from one process. The properties that influence the environment of the products shall be evident of the so called MSDS (Material Safety Data Sheet) that the supplier according to the law should attach a delivery. It appears that neither Ethylene glycol nor Propylene glycol is believed to have essential environmental effect properties. They both are fast biologically decomposable with moderate oxygen consumption.

Propylene glycol is sometimes environmental glycol or green glycol. The only environmental advantage with propylene glycol compared to ethylene glycol is that it is not toxic to consume, which in itself is a great advantage. Propylene glycol is apart from that neither better nor worse for our ecological or biological environment than ethylene glycol. However it has higher viscosity leading to inferior thermal properties (See 4.7.2).

Ethylene glycol and its solutions are toxic to consume and should be stored under safe forms. Propylene glycol is not toxic and can consequently be used in i Food stuff connected activity.

There are also propylene based products that are approved by NSF and FDA, which means that they can be used in food stuff. This is also valid about the inhibitors that are used, but the corrosion protection is generally poorer in these glycols as certain inhibitors have been removed. Systems where these products shall be used should be constructed in consultation with the secondary fluid supplier. Regarding products fit for human consumption it should be noted that the National Food Administration does not issue approval. It is expected that a system is tight and from that assumption the National Food Administration does not have any viewpoint about which secondary fluid that is used. The choice of secondary fluid is then a choice of the owner.

A so called. FDA or NSF approval is no requirement but it is the standard of the owner that sets limits for what secondary fluids that can be managed in ones production environment.

6.3.6. The Swedish Chemicals Agency and cases of poisoning with ethylene glycol

”Still many cases of poisoning caused by antifreezes with eyhlene glycol ((2004-02-25)

In spite of the fact that the problem with cases of poisoning, caused by eyhlene glycol in antifreezes, was paid attention to alredy 1987 the number of cases of poisoning has not been reduced. Most cases of poisoning are real accidents. Already half a deciliter ethylene glycol can lead to life threatening poisoning in a child.

The Swedish Chemicals Agency *(KemI)* together with the Swedish Poisons Information Centre now urge companies that supplies antifreezes with ethylene glycol to take measures to reduce the risk for risken for accidents: All products that are based on ethylene glycol and that are intended for consumeuse should be a provided with child protected sealing. A bitter agent should be added to the liquid tomake the taste prohibitive.

The problem with ethylene glycol in antifreezes has been raised with the industry. A solution was among else discussed that would imply that ethylene glycol should be replaced with a product less injurious to health, i.e. propylene glycol. */ A total transition is estimated to happen in the future as the problem with corrosion has not been solved yet.”

*/ It should be mentioned that such a transition (to propylene glycol) usually is not suitable for use in refrigeration and heat pump plants in view of the high viscosity for propylene glycol at low temperatures.

”KemI welcomes industry recommendation about ethylene glycol (2004-06-21)

The number of cases of poisoning due to ethylene glycol in antifreezes is unacceptably higt. Already half a deciliter eyhlene glycol can lead to life threatenng poisoning in a child.

The industry organisations ”Plast- & Kemiföretagen” together with ”Kemisk-Tekniska Leverantörförbundet” and ”BIL Sweden” now admonish suppliers of ethylene glycol based antifreezes to add bitter agents and provide the packaging with child protected sealing. This takes place through a recommendation to the member companies. KemI views the initiative very positive. This is an important step to reduce the risks of poisoning caused by ethylene glycol, acc. to Inger Cederberg at the Swedish Chemicals Agency

Earlier in 2004 the Swedish Chemicals Agency called to a meeting with med suppliers of antifreezes, automobile suppliers, the industry organisation “Plast & Kemiföretagen” and the Swedish Poisons Information Centre to discuss the problem med poisoning caused by ethylene glycol. The delegates were in agreement that measures had to be taken.”

6.4. Chemical and physical properties of water solutions

Table 6.2 gives symbols, chemical formula, molecular weight and values of a number of physical properties for each type of additive used as secondary fluid.

The tables also give the following grades:

Hazard marks: **Xn**: harmful, **Xi**: irritating, **F**: flammable, **T**: toxic, **C**: corrosive, **N**: dangerous for the environment.

Poison class CH is a classification of the chemicals according to Swiss toxic law. Descriptions of symbols: **3**: strong toxins, **4**: harmful substances and products, **5**: substances with a low hazard potential, **F**: not subject to toxicity classification.

LD_{50} oral rat is defined in 3.2.1. LD_{50} values given by manufacturers may not include the effects caused by the inhibitors and other additives.

Lethal dose for humans is also given. Note that the lethal dose of concentrated ethylene glycol is ≤ 1.5 g/kg or ≈100 ml for an adult.

(Aittomäki 2003, Hägg 2002, Melinder 2008).

Tabell 6.2. Chemical and physical properties of each type of additive in water solutions

*/ Values given are for additive concentrations giving t_F = -20ºC/-40ºC.

(…)Values in brackets are for concentrations not often used in common applications.

Type of additive	Ethylene glycol	Propylene glycol	Ethyl alcohol	Methyl Alcohol
Symbol	**EG**	**PG**	**EA**	**MA**
Synonym	Ethanediol	Propanediol	Ethanol	Methanol
Chemical formula	$C_2H_60_2$	$C_3H_80_2$	C_2H_60	CH_40
Molar mass [g/mol]	62.07	76.09	46.07	32.04
Freezing point [ºC]	-20ºC/(-40ºC)	-20ºC/(-40ºC)	-20ºC/(-40ºC)	-20ºC/-40ºC
Additive conc. [wt-%] */	36.2/(52.6)	39.4/(51.8)	29.7/(53.1)	24.9/41.0
Density [kg/m^3] 20ºC */	1047/(1068)	1031/(1039)	955/(907)	960/933
Refractive index, n_D at 20ºC */	1.369/(1.386)	1.377/(1.391)	1.353/(1.362)	1.339/1.343
Boiling point at 1 atm [ºC] */	103.9/(108)	104.0/(106.6)	84.0 / (81.2)	*83.9 / 78.6*
Surface tension at 25ºC [mN/m] */	58.2/(54.8)	47.9/(44.6)	32.3/(27.5)	*69.1/ (-)*
pH value (100g/l, H_2O; 20ºC)	6 – 7.5	6.5 – 8	≈ 7	*≈ neutral*
Toxicity, LD_{50} oral rat [mg/kg]	4700	19400	6200	
Toxicity, Human lethal dose [ml/kg]	1 - 1.5	> 15	47.5	
Hazard marks	Xn	-	F	
Poison class CH	4	F	F	
Odour, at 20ºC	No odour	No odour	Pleasant	Mild
Taste	Sweet	Tasteless	Burning	Sweet

Cont. Table 6.2. Chemical and physical properties of each type of additive in water solutions

*/ Values given are for additive concentrations corresponding to t_F = -20ºC/-40ºC.

Type of additive	Glycerol	Ammonia	Potassium carbonate	Sodium Chloride
Symbol	**GL**	**NH3**	**K2CO3**	**NaCl**
Synonym	Glycerin		Potash	Table salt
Chemical formula	$C_3H_80_3$	NH_3	K_2CO_3	NaCl
Molar mass [g/mol]	92.10	17.03	138.21	58.44
Freezing point [ºC]	-20ºC/(-40ºC)	-20ºC/-40ºC	-20ºC / -	(-20ºC) / -
Additive conc. [wt-%] */	46.3/(62.1)	13.5 / 21.1	31.0 / -	(22.6) / -
Density [kg/m^3] 20ºC */	1116/(1160)	945 / 919	1309 / -	(1169) / -
Refractive index, n_D at 20ºC */	1.393/(1.416)	1.350/1.362	1.386 / -	(1.373) / -
Boiling point at 1 atm [ºC] */	105.3 / (110)	60 / 44	104.7 / -	(105.9) / -
Surface tension at 25ºC [mN/m] */	69.1 / (68)	61.5 / 58.3	81.8 / -	79.7 / 83.7
pH value (100g/l, H_2O; 20ºC)	≈ Neutral	Starkt alkalisk	≈ 12	4.5 – 7
Toxicity, LD_{50} oral rat [mg/kg]	12600	350	2000	2800
Toxicity, Human lethal dose [ml/kg]				
Hazard marks	-	C,N	Xi	Xi
Poison class CH	-		4	F
Odour, at 20ºC	Mild	Sharp	Rather sharp	Salt

Cont. Table 6.2. Chemical and physical properties of each type of additive in water solutions

*/ Values given are for additive concentrations corresponding to t_F = -20ºC/-40ºC.

Type of additive	Calcium chloride	Potassium acetate	Potassium formate	Lithium chloride
Symbol	**CaCl2**	**KAC**	**KFO**	**LiCl**
Chemical formula	$CaCl_2$	$KC_2H_3O_2$	$KHCO_2$	LiCl
Molar mass [g/mol]	110.99	98.15	84.12	
Freezing point [ºC]	-20ºC / -40ºC	-20ºC/-40ºC	-20ºC/-40ºC	-20ºC/-40ºC
Additive conc. [wt-%] */	20.8 / 28.2	27.9 / 39.0	28.9 / 42.9	13.7 / 19.1
Density [kg/m^3] 20ºC */	1189 / 1162	1145 / 1209	1181 / 1284	1078 / 1110
Refractive index, n_D at 20ºC */	1.387 / 1.412	- / -	- / -	1.362 / 1.374
Boiling point at 1 atm [ºC] */	105.3 / 109.3	106.2 / 109.8	105.3 / 110	104.4 / 107.7
Surface tension at 25ºC [mN/m] */	79.7 / 83.7	- / -	- / -	78.2 / 79.1
Specific conduct. at 20ºC [mS/cm] */	179 / 179	132 / 132	227 - 260	150 – 170
pH value (100g/l, H_2O; 20ºC) */	8 – 10	7.5 – 8.5	9 – 11	8 – 10
Toxicity, LD_{50} oral rat [mg/kg]	1000	3250	2000	1000
Hazard marks	Xi	-		
Poison class CH	F			3

6.5. Environmental aspects with various types of secondary fluids

(B.Bredberg; BFR R11:1986)

6.5.1. General

Secondary fluids are classified as waste harmful to the environment and shall be treated in accord with valid decrees. No secondary fluid can be released in the drain without special permission by the commune. The environmental properties of the products shall be evident from the MSDS (Material Safety Data Sheet) that the supplier according to law have to attach a delivery.

6.5.2. Glycols

Ethylen and propylene glycol are produced in so called petrochemical industries, i.e. they have their origin in crude oil. The production takes place in i so called ”cracking” processes in combination with oxidation processes and is a part of a stream of products from one process.

From safety data Sheets it appears that neither ethylene glycol nor propylene glycol is believed to have properties that essentially influence the environment. They both are fast biologically decomposable with moderate oxygen consumption.

A great advantage with propylene glycol froman environmental aspect is that it is not toxic to consume. Propylene glycol is apart from that neither better nor worse for our ecological or biological environment that ethylene glycol.

Ethylene glycol (EG): EG is viewed as a great health risk for people as EG does not taste unpleasant, but is very poisonous for humans, the smallest mortal dosis is 1-1.5 ml/kg or ≈ 100 ml concentrated glycol in an adult. Short time exposure can result in irritation in öyes, skin and lungs. Repeated or long time exposure can result in kidney and brain damage (See also 6.5.5.).

Propylene glycol (PG): PG is not viewed as a health risk for people as minimal deadly dosis for an adult is 15 ml/kg, much higher than for EG. In line with this propylene glycol is at times called environmental glycol or green glycol. But for our ecological or biological environment propylenglykol is neither better nor worse than ethylene glycol.

Glycerol (GL): Swedish product from rape oil fields is viewed as environmentally friendly and not poisonous.

6.5.3. Alcohols

Alcohols can be dangerous when they are exposed to heat and flame mainly due to its low flash point. Alcohol vapour is often heavier than air and can form explosive mixtures with air.

Ethyl alcohol, ethanol (EA): EA can cause intoxication when consumed. About 10% of the denaturing addtives (in Sweden isopropanol and n-butanol) are added to prevent consumption. Smallest deadly doses for human: 47.5 ml/kg.

Methyl alcohol, methanol (MA): MA should not be used in many common applications when it is strongly poisonous. It can cause blindness if consumed and can also be deadly during inhalation. Smallest deadly doses for human: 0.3-1 ml/kg.

6.5.4. Salts

Chlorides: They will not easily cause poisoning, but can cause irritation in nose and throat during inhalation and in eyes from contact. Poisonous vapour develop chlorides are warmed up.

Potassium salts: Potassium salts on the market have more or less high pH-value.

Potassium carbonate (K_2CO_3): A high pH-value of about 12 causes risk of eye damage from contact with the liquid, which can happen during de-aeration of the system. Safety goggles shall be used during deaeration.

The organic potassium salts are viewed as environmentally friendly. They have rather high pH-value (baut lower than K_2CO_3) depending on type of inhibitor.

Potassium acetate (KAC): KAC has rather low risk potential but inhalation can lead to mild irritation.

Potassium formate (KFO): KFO is not believed to cause irritation in eyes, skin or lungs.

6.5.5. Ammonia

Ammonia (NH_3): NH_3 is very poisonous and can cause panic.

Ammonia has a very pervasive and unpleasant smell. It can cause strong irritation in eyes, skin and lungs and can lead to death if not fast vacation can take place. To swallow ammonia is corrosive for mouth, throat and stomach. Exposure to high concentration can cause temporary blindness and eye damage. Ammonia is very poisonous for water organisms. Ammonia vapohur is lighter than air and there is certain fire risk. During spill or disposal most is evaporated in the atmosphere where it quickly disintegrates. High concentrations, 25-30% NH_3, are very corrosive and can cause burns.

7. Projecting and dimensioning

7.1. Security aspects in view of the risk of breakdown

In refrigeration and freezer plants there is always the risk of a breakdown on any system or component. In unfavorable cases this can cause the plant (fully or partly) to collapse, which can mean that materials for large sums may be destroyed if the breakdown is long lasting.

For this reason it is very important that the risk of breakdowns is considered already when the system is constructed in order to reach a reasonable security. What is meant with reasonable security can vary depending on the type of plant and materials. The demand for availability should in each case be weighed against the added cost for the plant owner. The points below can serve as a check list for determining the consequences during breakdowns:

- One compressor falls away (breaks down).
- One refrigerant circuit in a unit falls away (breaks down).
- One secondary working fluid circuit falls away (breaks down).
- A whole unit falls away (breaks down).
- A good rule (usually a minimum requirement) is that one compressor will stay in operation with acceptable consequences.

Example: In the cases where freezer plant condensors are connected to the secondary fluid system for the refrigerating unit (so called cascade connection with the secondary fluid as intermediate circuit) the consequences of a possible breakdown on the refrigerating unit must be analysed in particular. The connecting alternative means that if the refrigerating unit breaks down, the freezer plant(s) will also be knocked out. Often this can not be accepted.

One way to avoid this is to equip the condensers of the freezer plant with double secondary fluid circuits. City water can fittingly be used as secondary fluid. The change-over can either take place automatically after alert indication or manually. The emergency cooling circuit is usually connected in series with ordinary condenser on the refrigerant side. It is then essential that the ordinary condenser is placed before the stand-by condenser. If not, then there is risk of freezing in the stand-by condenser if this is filled with cooling-water. (Ekman 2003)

7.2. Advice for projecting, constructing and assembling

7.2.1. Advice for projecting indirect refrigerating systems

It is important to know the prerequisites of the chosen secondary working fluid as to material compatibility and thermophysical properties so that the system will be designed in a right way. In order to avoid damage and problems of corrosion the following should be considered (see also the article: ”Rengör värmeväxlarna och avgasa rörsystemen” in AMA-nytt 2/99). Reference has been made to the BSAB-codes where corrsponding demands or advice can be found:

- Avoid galvanic corrosion by choice of material with as homogenous composition as possible. Preferably choose higher quality of steel or metal, they are normally less sensitive to corrosion and consequently give longer lifespan.
- Follow the material recommendations of the liquid supplier and other suppliers. Keep in mind that certain materials can be unsuitable at the temperatures in the system.
- Construct and mount the system so that it easily can be filled, deareated and maintained. Oxygen in the system always benefit corrosion processes. High and local pressure drop can cause air – and consequently oxygen – to penetrate the system. Avoid this by choosing suitable tube dimension.
- Correct tube dimension also gives correct flow velocity in the system and lessens the risk of erosion corrosion (inner eroding of tube wall).
- Keep the liquid clean. Metal ions and corrosion rests can cause shaft sealings in pumps to be damaged.
- The air in the system is separated wiht effective air purgers placed where the the static pressure of the system is low. Vacuum degasification is a very good method. Automatic high point air purgers shall be equipped with a valve.
- Filters shall be mounted at the pressure side of the pump. In systems with plate heat exchangers skall filters with maximum mesh size of 1.0 mm shall be mounted before the heat exchanger.
- Expansion tank with sufficient volume and prepressure shall be connected just before the suction side of the pump. The pump shall be ensured with a sufficiently high static pressure the suction side so that gas deposits do not emerge in the pump.
- Systems with operating temperature below 0^oC shall preferably be welded or soldered. Couplings shall be chosen in view of the risk of frostburst.
- Pumps shall be provided with shutting off valves and fully covering drip plate.

Brief "Ange-texter" can be found in RA 98 VVS and RA 98 Kyl (Refrigeration) that should be considered during planning. Advice about putting into operation and operation of indirect refrigeration systems is also given.

VVS AMA 98 or Kyl AMA 98 contains no prescriptions about secondary fluids.

Codes and heading for secondary fluids can however be found in RA 98 VVS and RA 98 Kyl. (AMAVVS EL 2/2000)

7.3. Sectioning, adjustment, draining of system

7.3.1 Sectioning

Secondary fluid systems both on the evaporator and conderser sides should be designed with a suitable number of sectioning valves so thatt the whole system can be divided in smaller part systems.

A suitable division make it easier for service and maintenance or other interference in the plant. Every part system should also be able to be drained of fluid to a separate catchment vessel (or drain if that is permitted). The sectioning implies that liquid volymes will be blocked in without the possibility of communication with installed expansion vessel.

A way to "internally" handle expansion of these liquid volumes is to mount non-return valves as a standard parallel over the various possible closing valves and that opens to the expansion vessel at rising fluid temperatures. The solution means that no secondary fluid will be released to the surrounding. Loss of fluid and surface corrosion (for salt solutions) can in this way be avoided. The principal arrangement is clear from Figure 7.1.

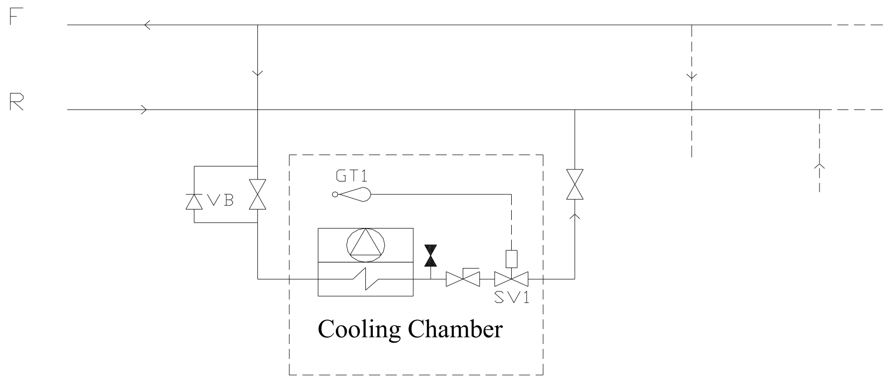


Figure 7.1. Sectioning valves och non-return valves for a part system

7.3.2 Adjustment, draining of system, pressure testing

In order to adjust and drain fluid from the system, combination valves available on the market are recommended (from i.e. TA or Armatur Johnson). These are also equipped with outlets for pressure drop or flow measurement that can be used during the adjustment phase (See further Appendix 2 Chapter 7.8 Adjustment). These valves also serve as closing valves.

Secondary fluid systems are usually pressure tested with water. During drainage some water usually remains in the system as pipes do not have a little slope toward the drainage place. Water in the system can imply frost risk of i.e. liquid coolers during cold weather. It is essential to find out how much water that remains in the system after the pressure test and then compensate the freezing protection of the secondary fluid. (Ekman 2003)

7.4. Pump dimensioning (AMA Nytt 2_2000; Bredberg 2001; KBL1 1999)

7.4.1. Introduction

The pump curves found in manufacturers' catalogues refer to the pumping of water at a temperature of +20°C. The pump curve (at the operating point) should be corrected and adjusted as to change in power demand. The density and viscosity of the fluid at the operation temperature should be considered. Dimensioning of the pump and choice of type of shaft sealings shall be performed by the pump manufacurer on the basis of given data.

Give the following values when there is inquiry about pump dimensioning:

Head pressure	kPa	Viscosity	mPas, cP or cSt
Volume flow	kg/sec, 1/sec or m^3/h	Operation temp.	°C
Sec. fluid type	% or freezing point °C	Ambient temp.	°C
Density	kg/m^3	Surface treatment	Type

Revise stated value with regard to selected fluid and system resistance.
Revise stated value with regard to selected fluid in the system.
State freezing point or concentration of selected fluid!
State density at the operation temperature!
State viscosity at the operation temperature!
State minimum operation temperature!
State ambient temperature for pump motor!
State desired surface treatment of the pump!

7.4.2. Dimensioning of hydraulic pump work

Q / H – curve: Desired volume flow (Q) and head pressure (H) within the operating range of the pump is looked for.

While dimensioning outside normal operating range the risk of cavitation and formation of steam.

The operation point should be chosen to achieve best pump efficiency near 2/3 from Q=0.

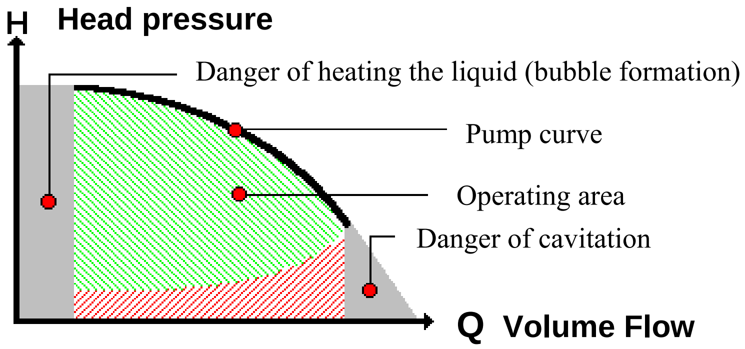


Figure 7.2. Head pressure

NPSH – curve: Net Positive Suction Head, NPSH - curve shows the lowest required inlet pressure to the suction side of the pump to prevent the liquid from evaporating.

Minimum pressure of suction side of the pump is ensured to prevent steam formation and gas precipitation.

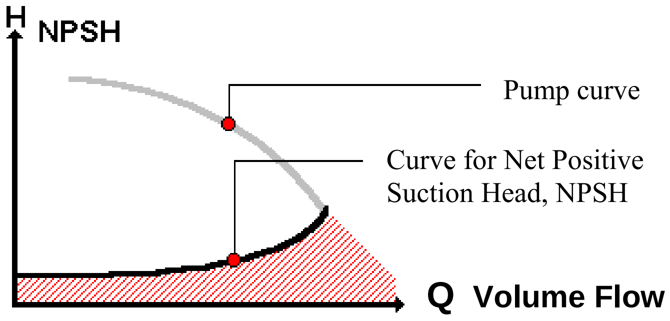


Figure 7.3. NPSH - curve

Power curve: Pump curves in catalogues from pump manufacturers are valid for water at +20°C.

The operating power of the pump is here illustrerated as function of the flow.

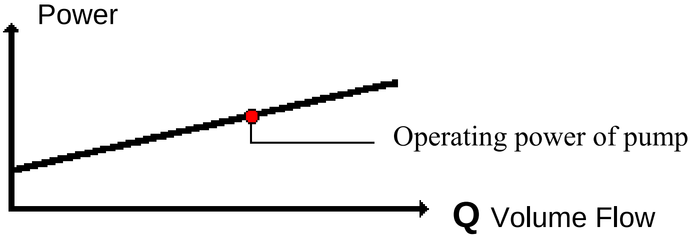


Figure 7.4. Operating power of the pump

7.4.3. Dimensioning of pressure drop in system

In a closed system it is only the friction resistance of the system that the pump shall overcome, as is illustrated by the following example.

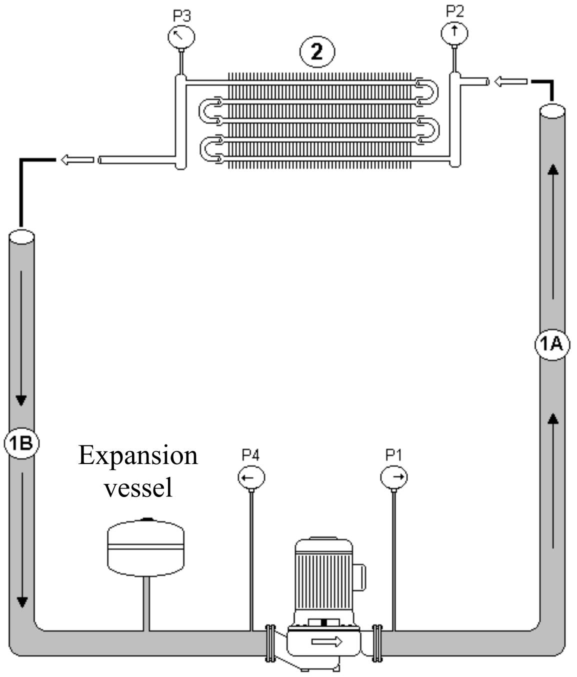


Figure 7.5. Dimensioning of system pressure drop

Calculation of pressure drop:

Let us calculate the differential pressure of a circulation pump in the given principal system with water. The pressure drop to overcome is:

1A Inlet tube: The resistance in the tube = 10 kPa

2. Battery/Coils: Pressure drop over the coils. = 30 kPa

1B Return tube: The resistance in the tube = 10 kPa.

Total pressure drop = 50 kPa

The altitude difference in the inlet tube (1A) is counteracted by the difference in (1B) and should not be calculated as a load for the pump. The liquid columns create balance.

Over pressure (static pressure) in the system, 1.5 bar, is kept constant by the expansion vessel, ensuring the inlet presure to the pump. The vessel shall always be connected to the suction side of the pump.

If we assume that the total elevation from the lowest point to the highest is 10.2 meter = 1.0 bar we get the following manometer pressure during operation:

Manometer P1 = 2.0 bar; P2 = 0.9 bar; P3 = 0.6 bar; P4 = 1.5 bar

The pump is dimensioned for a differential pressure of 0.5 bar when using water. Flow, pressure and operating power need to be corrected when using another secondary fluid.

7.4.4. Heat emission from circulation pumps to the secondary fluid

A part of the added operating power P_1 strains the secondary fluid system through addition of heat to the secondary fluid that has to be cooled off by the refrigation unit.

Incorrect choice and dimensioning of pumps can give a substantial heat addition.

Dry pump:
In a dry pump the heat addition amounts as an example to approx 30% of supplied energy P_1.

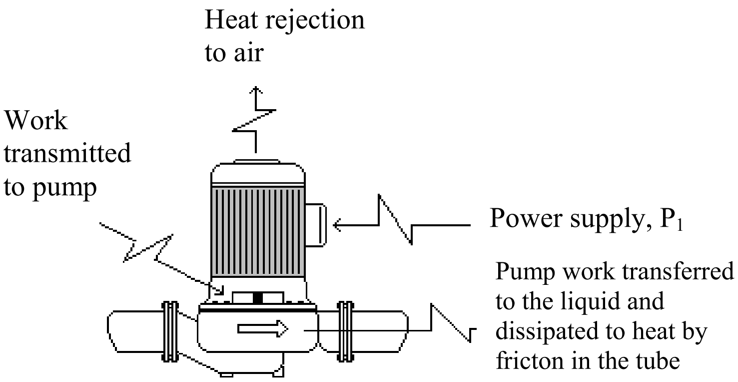


Figure 7.6. Dry pump (air cooled motor)

Wet pump:
In a wet pump heat addition amounts as an example to approx 50% of supplied energy P_1.

Example: Pump DN 50 with supplied power P_1 = 420 W; 2-pole motor.

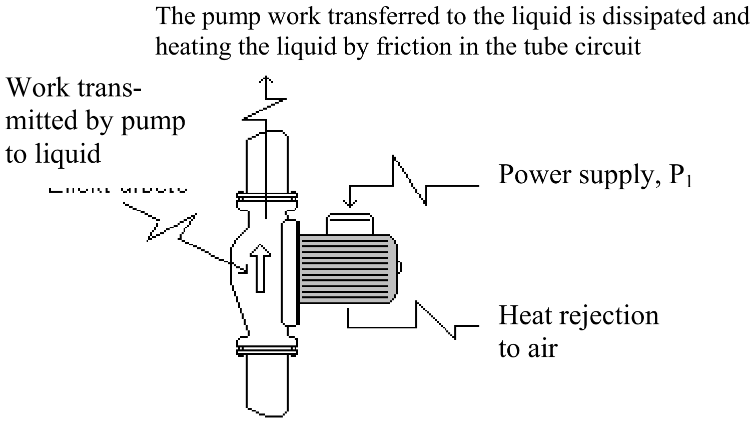


Figure 7.7. Wet pump (liquid cooled motor)

When choosing pump type in indirect refrigeration systems it is important to choose correct type. In systems ith pump power P_1 less than 1000 W, the wet pump can be as it minimizes the risk of leakage at the shaft sealing and contributes a system that does not leak.

Wet pumps have temperature limits in view of risk for condense at secondary fluid temperatures below 2°C. Most wet pumps can today work down to about –10°C. Small pumps in special design can operate down to –25°C.

7.4.5. Description of two system types

Let us look at an indirect refrigeration system with four cooling rooms. E need to consider heat transfer properties of the secondary fluid, achieve calulated temperature decrease over the coils, and get the flow and pressure drop for the secondary fluid. Only calcualting the transport of power from the object gives us a rong view of flow and pressure drop. We must focus on what we want to achieve, to take up and remove calulated power from the object.

The power is transferred from the air over the refrigerating coils to the secondary fluid that then transports the entered power. Secondary fluids with good heat transfer properties at low pressure drops give low energy usage for the pump work.

Let us consider a system with the same cooling surface and tube dimension and only look at the differences between the secondary fluids.

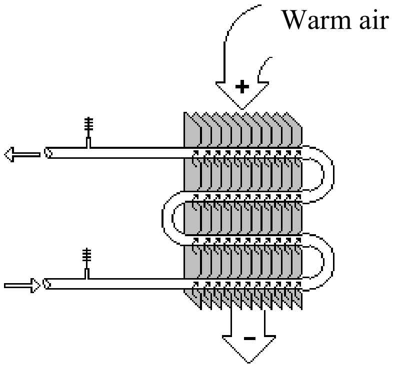


Figure 7.8. Heat transfer from air

What flow and pressure drops they get to achieve the same temperaturee decrease over the coils as well as the energy consumption to the pump ork. The dirrerence in using new pump technology with variable speed-controlled pumps rather than pumps with fixed rotational speed. With variable speed controlled pumps the steering system for temperature regulation and flow adjustment can be simplidied. This leads to substancial savings and improvement of technical, operational and economical properties. Installation, operation and maintenance are simplified by use of variable speed controlled pumps.

SYSTEM 1: System with one main pump P-1 and regulation using three way valves (constant flow in the whole circuit)

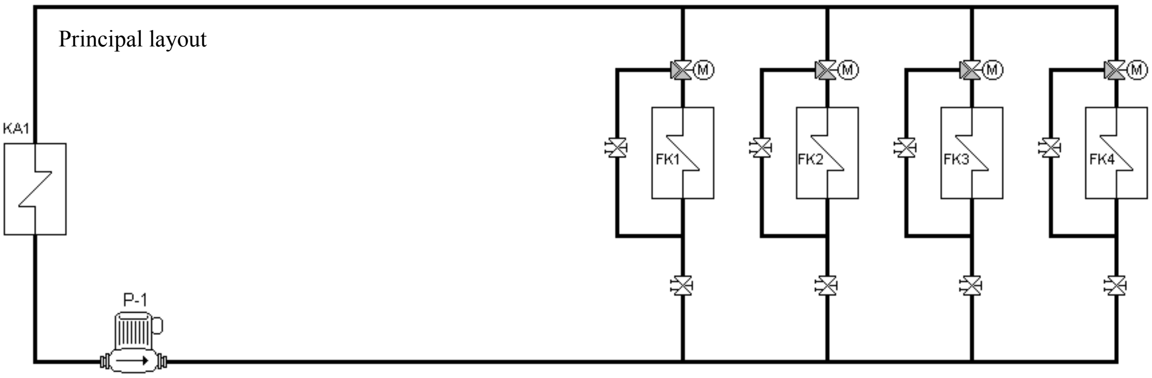


Figure 7.9. System with one main pump P-1 and regulation using three way valves

SYSTEM 2: System with two pumps, one charging pump P-2 and one main pump P-1. Regulation with two way valves and variable flow in the circuit. P-1 is chosen as variable speed-controlled pumps. (As P-2 a pump with stepless adjustable rotation speed is choosen)

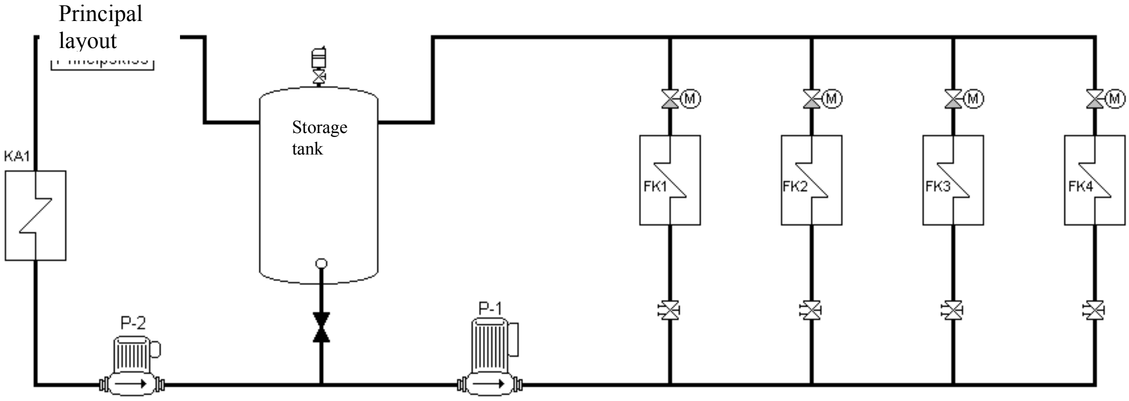


Figure 7.10. System with two pumps, one charging pump P-2 and one main pump P-1.

With the use of new pump technology we can achieve considerable energy gains in indirect refrigeration systems. With variable speed-controlled pumps (system nr: 2), lower energy usage is obtained due to adjustment to the momentary flow demand.

7.5. Checklist – Dimensioning and choice of components

Below follows a checklist for dimensioning and choice of components:

Choose suitable characteristic of the control valve. For cooling and heating coils, secondary fluid coolers etc a logaritmic characteristic should be chosen. In the case of 3-way valves a regulating port (connected to the coil) with logaritmic characteristic is chosen while a shunt port with linear characteristic is chosen.

Choose valve med suitable regulating range and an actuator with sufficient dissolution. In certain cases the speedof the actuator can be a parameter of importance.

Dimension valve for current circuit so that the regulating leverage amounts to at least $\beta \geq 0.5$.

Check and adjust the flow in the circuit.

Avoid to oversize cooling and heating coils.

If possible use shifting in suggested values for adjustment of heating and cooling demand. (Ekman 2003)

8. Design and control of system, humidity and condense

(AMAVVS 1/2005; Frygner, Wahlgren 2007, KBL1 1999, KBL2 2000)

During construction and assembling it is important to know the conditions of the chosen fluid regarding *material compatibility and thermophysical properties* so the system can be designed and assembled after these conditions.

8.1. Design of system to avoid various types of problems

8.1.1. General

Select and assemble material with as homogenous composition of metals as possible to avoid galvanic corrosion. Preferably choose higher quality of steel or other metal. Higher qualities can be less sensitive for corrosion and can give longer operation life.

Follow the material recommendations of the suppliers of secondary fluid and other material. Consider that certain materials are not suitable from a temperature aspect.

Construct and assemble the system so that it easily can be charged, air purged and serviced.

Presence of oxygen always favors corrosion processes. High and local pressure drop can locally give pressure below the atmospheric and if the systeme is not tight, air can come into the system. This can be avoided by choosing correct tube or pipe dimensions and by avoiding local changes in dimensions.

When choosing suitable flow velocity the risk of erosion in the system (invändigt slitage av rörväggar) is avoided or lessened. Contamination of fluid, with metallic ions and corrosion deposits, might damage shaft seals in pumps.

Air in the system is removed with effective air purgers placed where the static pressure of the system is low. UVacuum air purging is an excellent method. Automatic top air purgers shall be provided with a valve, so these can be shut off from the system when necessary (see passage headed Air purging).

A filter shall be installed on the delivery side of the pump. In systems with plate heat exchanger a filter with maximal mesh width of 1.0 mm shall be mounted before the heat exchanger.

An expansion vessel of sufficient volume and prepressure shall be connected just before the suction side of the pump. A sufficiently high static pressure at the suction side of the pump has to be ensured to avoid cavitation in the pump.

Systems with operating temperatures below 0°C shall preferably be welded or soldered. Connecting pieces shall be chosen considering the risk of frost burst.

Pumps should be supplied with shut-off valves and fully covering drip plates.

8.1.2. Clean systems (KBL1 1999)

Systems operating with water based anti-freeze solutions, *secondary fluids*, have to be cleaned before charging the system. Solid contaminants cause problems in the systems by clogging and damage on pumps by increased risk of corrosion.

Recommendations for cleaning of system:

- Be careful when assembling the system. Use end covers for the pipes.
- Clean the system section by section.
- When changing the scondary fluid, contact the supplier for information.
- Pressure test the system before charging!

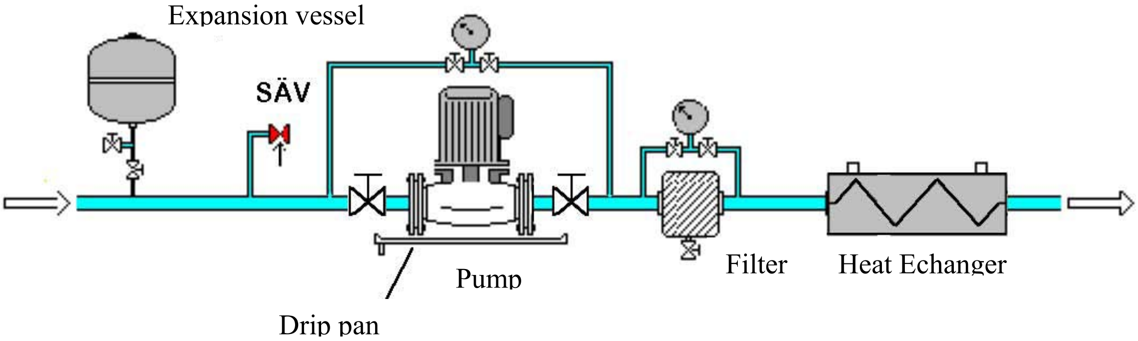


Figure 8.1 Part of system with expansion vessel, filter, closing valves and drip plate

8.2. Delivery, charging and storage (KBL1 1999, KBL2 2000)

8.2.1. Delivery

When secondary fluid is delivered it is important to check that it has not been contaminated via damaged packaging. When delivery in a tanker truck there should be a certificate showing that the tanker has been properly cleaned. Take a sample of the fluid before charging. Check that the liquid is clear and not discoloured. Mark the sample and save it as reference for 6-12 months.

8.2.2. Recommendations during charging

Recommendations for cleaning of system:

- Before charging the system must be thoroughly cleaned. Charge the entire system at one go or by sections, to avoid unnecessary exposure to oxygen, for example pumping via open vessels or similar.
- Vessels used for charging must be sealed to prevent contaminationand emptied after use.

- When charging the system it shall if possible be charged from the lowest point so the liquid can press out air at top points in the system. Charge slowly to avoid air pockets.
- Keep in mind that if the system is test pressured with water it is difficult to empty the system completely. Remaining water dilutes the solution that is pumped in which may cause the freezing protection and the corrosion protection to be impaired. The calculated charged volume will also not be correct.
- Charge the whole system at the same time or shut off charged sections so that secondary fluid and material cannot be exposed to oxygen in the air.
- Mark the system visibly with:

-Name and concentration / freezing point of secondary fluid

-Name and address of supplier

- Charging date

- The name of the person responsible for the charging.

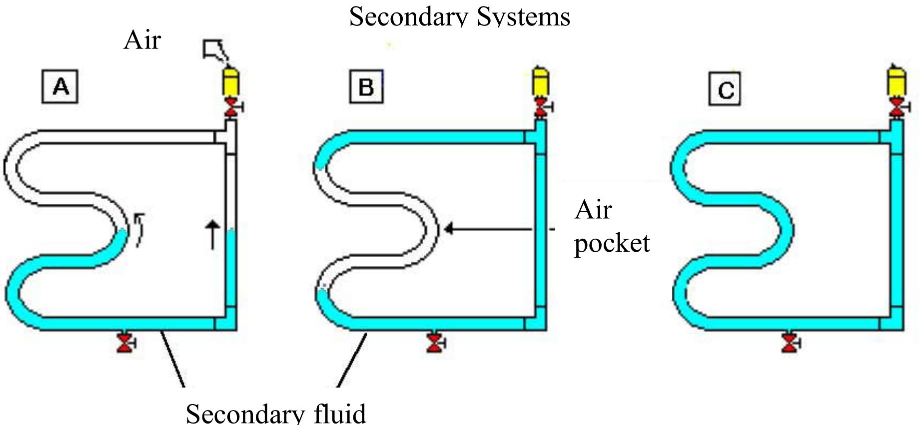


Figure 8.2 A-C. Charging of secondary fluid system

<u>Figure A</u>: It is important to charge the system slowly in order to prevent air pockets in the system.

<u>Figure B</u>: A poorly charged system with air pocket.

<u>Figure C</u>: The system should be fully air purged. The secondary fluid shall be premixed. Donot pump water and concentrated solution separately.

8.2.3. Storage

It is important that secondary fluids are stored in sealed vessels to minimise the risk of contamination. Before using a secondary fluid that has been stored, check carefully that the liquid is not discoloured (rust-coloured). Half-filled metal drums rust internally due to condensation.

8.3. Start-up and operation disturbances (KBL1 1999)

8.3.1. Start-up

The cooling machinery and other equipment can be started when the system is carefully air purged. Keep the system under observation especially with regard to air in the system and that overpressure is maintained during the entire cooling down period.

During start-up of a secondary fluid system there is risk that the system corrodes. Air dissolved in the secondary fluid and remining air or leakage in the system can give rise to conditions that cause corrosion in the system. There is then a risk that the corrosion inhibitors are consumed and that the system then lack functioning corrosion protection during continueda operation. Selection of materials that are not compatible with the secondary fluid used may mean chemical reactions in the secondary fluid that affect the quality. It is therefore fitting to make tests of the secondary fluid 1-2 months after start-up. Check-ups should then regularily be made with 0.5 - 2 years intervals, depending on the recommendations of the supplier.

It may also be suitable to include as a requirement that the supplier should hand over a protocol from performed and approved control of secondary fluid solution the latest two months after start-up and then the latest at the warranty inspection.

8.3.2. Operation

If the secondary fluid need to be replaced or in any other way changed, it is advisable to contact the supplier in order to avoid future operation disturbances. The supplier may assist with analysis, advice about mixing, pump changes etc. Dry pumps have a shaft sealing that should be chosen with regard to secondary fluid used. The shaft sealing is constructed in a way that it always shall be wet (lubricating film in sealing areas).

During operation water evaporates and addtives may be deposited around the pump, which is normal. This may give rise to outside corrosion on materials around the shaft sealing and pump (especially if the secondary fluid is a salt solution). Depending of choice of secondary fluid this can appear as deposites or liquid droplets. A mechanical shaft sealing is a detail that wears and need to be checked and replaced when needed.

8.3.3. Operation disturbances

Operation disturbances usually depend on poorly deaired system. Air in the system cause poor dålig heat transfer in heat exchangers, impaired pump capacity with lower flow as a result and increased risk of degradation of corrosion inhibitors in the secondary fluid.

Abnormal wear/leakage in shaft sealing can be caused by dry operation, contaminations; faulty inhibitor composition an/or wrong water quality. These cases are not to be classified as material or manufacturing fault.

When systems are heavily contaminated it might be necessary with chemical cleaning using i.e. weak and inhibited acids. Engage well-known companies with good references. Explain carefully selection of materials in the system. See to it that the contract includes after-rinsing and neutralisation (See further the article ”Rengör värmeväxlarna och avgasa rörsystemen” in AMA-nytt 2/99).

8.4. Control and maintenance

Materials and components that are compatible with the chosen fluid should be used. Information about this can be obtained from the supplier. If the plant is not in "balance" this can be costly in the form of high energy consumption or other operation problems in the plant. Continual control and care of the plant leads to increased life expectancy as possible faults and leakage are detected early.

8.4.1. Control of fluid

Introduction (KBL4, 2003)
Make a regular control of the fluid according to the specifications of the supplier. This can be done by self control or by sending samples to the supplier or its representative for analysis.

A sample of about 0.5 liter is sent packaged in unbreakable emballage, preferably of plastic.

The sample is marked with name of company, type of secondary fluid, test date and contact person etc. or acc. to specifications of the supplier. Follow rules of the transporting company.

In order to be able to draw correct conclusions of the results of the analysis it is also important to attach information about the plant. Clearer descriptions make it possible to draw more correct conclusions of the results. Correct values of the commercial product may be obtained from the supplier.

- Analysis of the pH – value of the fluid (acidity).
- Density/refractive index - indicate freezing protection of the fluid.
- Colour - Compare with reference sample that always should be kept during filling.
- Smell - Compare reference sample that always ought to be saved during charging.
- Corrosion products - or "rust products" as well as corrosion protection/inhibitor usually requires advanced equipment. Test sample should be sent to the supplier or its representantive.

Control of secondary fluid quality (AMA Nytt 2005/1)
The inhibitors can be consumed in the system and the secondary fluid is affected by various factors. For these reasons it important to periodically check the quality of the fluid. Without corrosion inhibitors systems with salt solutions can quickly be exposed to extensive corrosion attacks that radically shorten the life expectancy of the system. The first check-up should be done 1-2 months after start-up. Check-ups should after that be done regularily according to the specifications of the supplier, usually at 0.5-2 years interval.

Control of product properties (AMA Nytt 2005/1)
Properties that are important to check can vary somewhat between different products, but the following should usually be checked:

- pH-value (shall be near the value in the originally charged solution).
- Density/refractive index give an indication of concentration and freezing point.
- The occurence of metal iones and corrosion products.

- Colour and smell (compare with reference sample).
- The concentration of corrosion protection.

Density, pH-value as well as colour and smell can be checked with rather easy methods. A liquid with correct pH-value indicate a healthy system. Inorder to check the occurence of metal iones and the concentration of corrosion protection requires more advanced equipment. Laboratories can check the occurence of metal iones.

Suppliers of secondary fluid solutions often have equipment to perform the check-up. Samples can be sent to the supplier for analysis. If you order a complete test you get a report and advice about what measures should be taken with the secondary fluid solution. The analysis may also contain ideas about why the secondary fluid solution is in the actual condition. The cost for a complete analysis with protocol and suggestions of measures to take differ between various suppliers but costs about USD 3-400. The supplier of secondary fluid solution should be able to determine what quality requirements the liquid should meet.

Refill of glycol solution

Car radiator glycols are not produced to be used in heating and cooling systems, but for car engines with other materials and temperatures. The inhibitors differ depending on usage and it is far from sure that glycol types that are mixed will function together. It has happened that a relatively small amount of the added glycol has knocked the corrosion inhibitors of the original solution. Products that are formulated for heating and cooling systems should be used.

8.4.2. Control of components

Function

Pumps: - Check regularly that the pump with fully covering drip plate and splash protection functions well. Rinse at once away any spill.

Valves: - Check regularly that the valve functions and is tight. Rinse at once away any spill.

Air purging devices: - Check regularly that the device functions well and is leakage free which is important for satisfying operation. Air purging valves shall when need be demounted and cleaned, suitably with water. Air purging is most effective with a warm system. [See also earlier publications from the Secondary Refrigerant Team.]

Expansion vessel: - Expansion vessel pre-pressure – shall be checked to ensure sufficiently high static pressure of the pump. See to it that the device has the pre-pressure correctly set in pressure isolated state. The equipment should of course be leakage free. The pressure of the system shall regularly be checked.

Filter: - Filter is placed on the pressure side of the pump. Select suitacle mash size and clean the filter regularly.

Measuring devies, manometres etc.: - Check that the equipment functions and is leakage free.

Leakage

Welding joints: - Check that the joints have no leaks. Act on any leakage.

Soldering skarvar: - Check that the joints are leakage free. Act on any leakage. (Use hard soldering.)

Flange connections: - Check that all flanges are leakage free. Re-tightening of bolts may be needed.

Fittings: - Check that the fittings have no leaks. Re-tightening may be needed.

Cooling coils and heat exchangers: - Check that the units are clean leakage free. Act on any leakage.

Spill and leakage: Spill and leakage slould be collected and then flushed clean with water – in order to avoid i.e. external corrosion problems. Note: Used secondary fluid may not poured into drains. Pay attention to leakage through the insulation.

Notice: Maintain even service intervals and document all services that are performed.

Find out the reason to the problem

Do not only act on a fault but find out the reason to the problem, or else the risk is great that the problem appears again. If necessary, change to another type of a certain component.

8.4.3. Analysis of the corrosion condition of a plant

Introduction

During analysis of the corrosion condition of a plant with a secondary fluid containing an electro chemical inhibitor analysis of the inhibitor is relatively simple. The inhibitor is loose in the fluid and is temporarily bound to areas with initiation corrosion corrosion and then is returned to the fluid. Hence, the inhibitor concentration will be constant in a sound system when equilibrium conditions again have been obtained. The inhibitor concentration should not be below a level rekommenderad by the manufacturer. In a plant with film forming inhibitor it is harder to analyse the inhibitor if the protective film in the system is intact.

The sample of the secondary fluid solution should be collected in a place where the secondary fluid solution can be expected to be representative of the system. If the sample is taken at a place where the fluid is not circulating the pipe should be flushed before the sample is taken. The sample is then put in a clean bottle and half a liter is sufficient for the analysis.

Do not only act on the symptoms

If it is evident that the inhibitors in a secondary fluid solution largely are consumed then an investigation should be made of the reasons why the inhibitors ae gone before new inhibitors or new secondary fluid solution is charged. New inhibitors can as mentioned put added protective layer on the inside of the tube system which impairs heat transfer. If the reason that the inhibitors are consumed is not acted upon then there is risk that they will soon be consumed again after a refill. Then it is only the symptoms of a not fully healthy system that has been acted upon.

At times the secondary fluid system is refilled with ordinary water. The result is that the secondary fluid solution is diluted, the inhibitor concentration is lowered and the freezing point is increased. This can often be detected in the analysis as ordinary water usually contains calcium, which ready mixed secondary fluid solution seldom does.

In order to make a leakage test of a system it is common that it is filled with water. It can be difficult to empty all water from a secondary fluid system. Water remaining in low points will during refill with secondary fluid solution dilute this with higher freezing point as a result. When there is water remaining in a system then a somewhat higher concentration of the secondary fluid solution should be chosen than what is given by the lowest dimensioning temperature of the system.

8.5. Humidity and condense of tubes in secondary fluid systems

(Frygner 2007)

8.5.1. Condense on tubes in secondary fluid systems

Cold tubing and installations

The most common consequences of condense on tubing in secondary fluid systems are humidity and water damage, shorter life expectancy and poorer insulation ability.

Cold tubes and installations can be found in several different applications. Lacking condense insulation on cold installations can cause several types of problems where the most common and serious are condense precipitation and humid areas on cold surfaces. Wth time this can lead to connected problems in the form of moisture and water damage, shortened life expectancy and increased energy losses.

Condense precipitation may cause problems in all parts of a secondary fluid system: at the refrigerating unit (cooling machines, circulation pumps, filter, etc.) on secondary fluid tubing and at cooling units (cooling coils, fan convectors, cooling baffles, etc.). Tubing in secondary fluid systems are insulated to lessen the heat uptake and to prevent condense on the installations and humidity accumulation in the insulation. Faults in condense insulation can cause several problems.

Problem with faulty condense insulation in secondary fluid systems

Moist and water damages: The problem that first appears in a secondary fluid system with faulty condense insulation is condense dripping on surfaces below. This takes place when sufficiently great amount of water is gathered on the secondary fluid tube, in the insulation o on the installation, and it can cause humidity and water damage on the affected material.

Lessened lifespan: Installations with condense precipitation and humidity accumulation in the insulation will with time lead to increased corrosion. The weight of the insulation increases at the same time causing increased pressure stress on tubings and their suspension components. All these effects lessen the life span of installations and of insulation in the secondary fluid system.

Impaired insulation ability: Moist insulation causes impaired insulation ability which brings increased heat uptake from the surrounding to the secondary fluid transported in the tube. An impaired heat insulation ability may lead to a warmer secondary fluid when it reaches the place where it will be used. This leads in turn to increased electric power consumption as the refrigeration plant must produce more cold, among else as even colder secondary fluid is required. As a result the refrigeration plant may not be able to cool the plant. Insufficient cooling fability also cause negative consequences for health and well being.

For all these reasons it is important that a faulty condense insulation is acted on åtgärdas before it leads to serious problems.

8.5.2. Handling of moist and condense on tubings in secondary systems

Problem with moist and condense

Condense from surrounding air always occurs on a surface with lower temperature than the dew point of the air Normally humidity accumulation in insulation as well as condense are avoided at unsuitable places in the secondary fluid system condense insulation of exposed parts of the system. When a problem arises it is often caused by one of the following reasons:

- Wrong dimensioning during projecting.
- Faulty performance during installation.
- Choice of wrong method.
- Changes of condense insulation of existing installationes (damage, reconstruction etc.)

When building a secondary fluid system you should from condense point of view see to it that all cold surfaces with risk of condense precipitation should be insulated with sufficient amount of condense insulation and that the installation is perfomed in a correct way and with the right method.

Cold and unisulated surfaces

There are however parts of the system where you may permit and actively choose to let water condense on certain cold surfaces. It can be components in the system with moving parts that needs to be reachable for service, for example circulation pumps and filters, or units where wet cold is used to cool the air, as i.e. cooling coils and fan convectors.

Drip plate and collecting vessel

When water condenses on the cold side of the components this water will in time start to drip. It is then important to provide the unit with a drip plate (usually of stainless steel), a condense trough or similar where the condense water can be gathered. The drip plate shall fully cover all the length of the uninsulated component, including the fins, and be adjusted to the installation. Also the collecting vessel shall when needed be condense insulated. The collected water should be transported away from the plate or tray via a draining pipe to the drain and, when there is need, be complemented with an internal condense water pump.

Ways to handle humid air

Condense insulation serves to ensure that the temperature on the surface does not get lower than the dewpoint of the ambientair as well as protection against humid air to be transported into the insulation and remain there. Correct surface temperature is obtained by insulating with a material that has sufficiently good heat insulating properties and sufficiently insulating thickness for the temperature and humidity conditions for the current installation. Transport of humid air to the cold surface is handled according to two main principles:

1. The cold surfaces are provided with a vapor breaking layer that hinders the humid air from reaching the cold surface. This can be obtained either by insulating the cold surfaces with foam material with closed cell structure (cell rubber) or that the cold surfaces are provided with insulation that let through the vapor (i.e. polyuretan foam, mineral wool) and an outer vapor breaking layer on the insulation.
2. The humid air is allowed to reach the cold surface making it condense on the surface. A capillary absorbing wick then transports the condense water from the inside of the insulation to its surface, where the moist can evaporate. The purpose with the wick is to avoid filling the insulation with water. This does not work when the tube temperature is below 0°C as ice then is formed.

Method two can not be used if corrosion on the tubes can be expected as the surface will be moist.

In view of the negative consequences that a faulty condense insulation causes it is important that material and construktion functions well and that the work performance is correct. In order for the condense insulation to function all joints och lead-through must be fully tight to prevent the humid air to reach the cold surface.

If method number one is used then tube insulation ends should be glued against each other. It is also important to use the correct junction method (glue, double coated etc.) in order for it to be tight and getting long life length. When a method with steam brake is used it is important that it is protected against damage.

Reasons for condense precipitation on tubes in secondary fluid systems

A number of reasons for condense precipitation on tubes in secondary fluid systems are here summarized:

Joints and glued areas let go or are not fully tight.

The insulation is damaged or not fully restored from damage, reparations or other interferences in existing plants.

Uninsulated or poorly insulated tubes.

Poorly insulated drip plates or tray with condense precipitation on the bottom side of the plate.

Uninsulated or poorly insulated componentes (pumps, heat exchangers, filters, air purgers, valves etc.) without drip plate.

Other problems caused by condense:

Clogged draining pipe from drip plate.

Faulty condense water pump from condense water tank.

Condense precipitation on cooling units for dry cold (i.e. cooling panels and baffles) that are not equipped with drip plates or trays takes place when the secondary fluid temperature in the cooling unit are too low in relation to the humidity content of the room air.

(Frygner, Wahlgren 2007)

9. System construction and installation directions

(DEM/Frigotech 2001, Temper Technology 2006)

9.1. System construction - summary (Temper Technology, 2006)

The life expectancy and reliability of a refrigeration plant depends greatly on how it is installed, the purity degree of the system before the first start and on how the system components are chosen and assembled in relation to each other.

The following directions are intended for the installer/constructer and can be viewed as an aid in their work. The installation company must have needed competense to perform a complete well functioning indirect system.

Note: View points/recommendations below are based on available experience today and may be revised in the future.

- Secondary fluids shall generally only be used in closed systems in order to avoid air/oxygen. Oxygen always means an increased risk of corrosion.
- Consideration should always be paid to the operation temperature in the system when choosing materials. The higher temperature the better material quality needed.
- Suitable materials depend on secondary fluid type.
- Junction of secondary fluid systems shall be done with: 1. Welding; 2. Hard soldering (copper, silver solder); 3. Flange bonds; 4. Thread bonds (in exceptional cases); 5. Avoid clamp ring couplings of various types (See also 9.5)
- Air purging of the system is very important. Always mount high point air purgers (manual or automatic (manual can be closed at normal operation).
- Air purging is more effective at a higher temperature. The best resultat is obtained when the liquid is warm, ca +35°C. If possible use a dip heater i.e. Air purge properly before starting the cooling unit, especially if heating is not possible.
- Always mount a filter (0.6 – 0.8 mm filter mash size) that is easy to clean on the pressure sideof the pump.
- Consult your pump supplier regarding secondary fluid used for selection of pump / shaft sealing etc.
- For estimation of pressure drop and heat transfer coefficient in the tube system (See for instance calculation program from the supplier).
- Choice of correct flow velocity depends of dimension, viscosity and system design. Keep inmind that small dimensions in i.e. cooling coils can not take more than appr. 0.7 m/s but in larger dimensions the velocity canbe increased somehat (to about 1.0 – 1.2 m/s).
- Do not let the system stand empty or partly filled as surface frost then can be formed on the inside of the tubes.

9.2. Selection of materials (Temper Technology, 2006)

Most common materials can be used such as: copper, brass, steel, stainless steel, cast iron as well as plastic tubes (ABS, PE). All materials shall be adjusted to the min/max emperatures of the designated system.

Galvanic steel, zinc and soft solder are not suitable materials for potassium salts. High temperatures imply increased risk of corrosion. This is above all true for certain types of materials (i.e. cast iron). When choosing materials consideration should be taken to the operation temperature in the system. A higher temperature requires better material quality.

Sealing and packing materials shall be adjusted for min/max emperatures of the system. Fiber and teflon gaskets have poor flexibility which means they have more difficult to adjust to temperature fluctuations in the system and in that way increase the risk of leakage.

Rubber used as jointing material shall be compatible with the type of liquid used, if not the problem with leakage can develop, especially at the pump. Selection of materials in the system is affected by type of liquid, not only the temperature (Ignatowicz 2008).

The galvanic potential range

When choosing materials it is advisable to choose metals as close as possible to each other in the galvanic potential range.

It is optimal to use the same material consistently in the whole plant.

Most noble

- Graphite
- Stainless steel
- 18-8 Cr-Ni-Fe
- Silver
- Nickel, passivated
- Silver solder
- Copper-Nickel
- Bronze
- Lead

Cont.

- Copper
- Brass
- Tin
- Cast iron
- Steel, Iron
- Galvanized steel
- Zinc
- Aluminum

Least noble

9.3 Components

9.3.1. Pumps

- Always choose dry running pumps in place of wet running as the efficiency is considerably higher in a dry running pump.
- The pump shall be surface treated and a stainless steel drip plate mounted underneath.
- Stainless bolts shall be used.
- The pump shall if possible be surface treated on the inside, especially on the ”warm side”, temperatures above + 20°C.
- Consult the pump supplier and inform about the type of secondary fluid to be used.
- Experience shows that selection of shaft sealing is important and of the advisability to choose "hard" shaft sealing material (information is given by respective pump supplier).
- A small "leakage" shall exist to give a cooling effect between the stationary and the rotating sealing surface. When the secondary fluid reaches outside the sealing water will evaporate and remaining crystals will be visible. These can cause ugly corrosion damages

and should with regular intervals be flushed away with water. These problems can be avoided if a sealing free pump (wet running motor) is used.

- Follow the operation and och handling instructions of the pump supplier.

9.3.2. Valves

Valves shall be of such a type that is compatible with the list of materials, (See Chap 2). Keep in mind that a number of valve types i.e. ajustment, control and magnet valves can contain fiber gaskets, these should be replaced by rubber gaskets.
Consult each respective valve supplier for choice of valve, telling them type of secondary fluid and current operation temperature.

9.3.3. Expansion vessel

The function of the expansion vessel is to level out the volume and pressure variation that may occur when the operation temperature varies.

The dimensioning must be performed to give the plant a good function as to correct expansion volume and suitable system pressure at varying operation cases.

- In order to control the prepressure of the vessel the expansion vessel shall be supplied with closing valve and drainage valve.
- The expansion vessel is placed at the suction side of the pump where normally the lowest system pressure prevals, in order for the pump to get sufficient static pressure to avoid cavitation.
- The expansion vessel shall be compatible with the secondary fluid as to temperature and material.
- During dimensioning of expansion vessel it is suitable to use the directions and calculation programs given by the supplier.

A common fault i secondary fluid systems is that the expansion vessel is underdimensioned and has too low prepressure adjustment in relation to the total pressure of the system. This causes variation of the system pressure that may result in underpressure in the highest parts of the system. If automatic air purgers are mounted they may bring in air rather than remove air. A shut-off valve shall be mounted between the system and the air purger in order to separate the automatic function when the system is unguarded. If the expansion vessel is too small then the volume changes of the fluid can not counteracted by the expansion vessel.

9.3.4. Filter

Filter should be installed on the pressure side of the pump. With filters in the system corrosion products can be removed. These filters shall be easy to clean, for instance by flushing through. The filter mesh size should be 0.6-0.8 mm. Filter with small filter mesh size can be placed in a by-pass flow. This is recommended if the liquid contains large amounts of small particles.

9.4. Other directions for the supplier (Temper Technology, 2006)

9.4.1. General advice

- After mounting and pressure test and at correct operation temperature a re-tightening of flanges and couplings shall take place. Insulation work on these places shall be done after these measures have been taken.
- During mounting with ABS, PE or coresponding tubes it is important to use the glueing/connecting proceeding that the supplier recommends. The plastic must also be adjusted for the current operation temperature.
- Avoid using plastic flenges against the collar casing of plastic as there is great risk of wry pull and leakage. Use steel enforced flanges.
- Avoid fiber and teflon gaskets. A secondary fluid with low viscosity and surface tension makes the risk of creepings/leakage great at aulty mounting etc.
- Gasket material of EPDM or nitrile rubber is recommended. Note that even small amounts of oil can ruin EPDM gaskets.

9.4.2. Storing, cleaning

Storing: Storing of secondary fluid shall always be done in closed vessels in order to prevent evaporation with changed viscosity and density as a result.

Cleaning: Before charging the system shall be well cleaned from dirt, welding rests, possible remaining water from pressure testing and other "foreign" particles. Cover tube endings during the installation in order to hinder dirt and moisture to penetrate the tubes. Note: Cleaning is important as loose particles otherwise can cause galvanic elements and pitch corrosion in the system.

9.4.3. System charging

Preferably order ready mixed secondary fluids! If that is not done, always request instructions from the supplier about mixing ratio, mixing temperatures and how long the fluid should be mixed before the system is charged. Never use tap water during mixing or dilution, only de-ionized water is acceptable.

Charge the system with the fluid formulated for it. Remember that weight and volume percent are two different units. Check the concentration of antifreeze fluid in the system.

The system shall if possible be vacuum-sucked to facilitate the charge of the system, which results in less risk for air pockets. Vessels and tanks must be split off in order to avoid damage of material (implosion). Check with the supplier if installed components can be exposed to vacuum. Charge the system from the lowest point and fill slowly to avoid air pockets.

Charging is done with a separate pump or directly from tank. If the temperature in the system can be raised to about +35°C most ”micro bubbles” in the system can be removed.

Liquid drawn from the system shall be stored according to earlier directions and should not be stored in open container as it then will be oxygenated and dirt stacks up.

9.4.4. Conversion

A thorough review of the system is required when one type of secondary fluid is replaced with another. This is done to ensure that included components are compatible with the new secondary fluid and that the system is cleaned in a correct way.

9.4.5. Air purging

Air in secondary fluids and coolant systems are one of the greatest reasons for break down in liquid systems. In order to get a well functioning plant it is very important to remove as much air as possible from the system. Various problems can appear in poorly deaerated systems, such as:

- impaired pump capacity
- impaired heat transfer
- corrosion, erosion
- sealing problems

How much air a liquid can retain depends fully on pressure and temperature. High temperature and low pressure retains less air than low temperature and high pressure. (Warm up the system to about +35°C if possible, as this makes air purging easier.)

Air purging is mainly performed in the following ways:

- manual or automatic air purgers (high-point air purgers)
- air separators (micro-bubble separators)
- sub-pressure air purgers (vacuum degassers)

Manual or automatic air purgers shall be mounted in all high points and shall be placed at the end of the flow direction of the tube. If automatic air purgers are used then closing valves shall be mounted between the system and air purger. Automatic air purgers shall only be kept open during charging and service as in time there is risk that theses are clogged and instead serve to bring air into the system.

Air separators (micro-bubble separators) are mounted where the secondary fluid is as warmest (return pipe). Micro-bubble separators only remove the ”free air”.

A sub-pressure air purger can be used to remove the bound air from the system. This doesnot have to be permanently connected is the system. It is normally enough if it is connected during the start-up during a certain time of operation depending on the system volume. Contact the supplier of the sub-pressure air purger to get more information.

The above mentioned suggestions/recommendations are as earlier mentioned based on available experience today and may be revised in the future.

9.5. Installation guide for secondary fluid systems (DEM/Frigotech 2001)

9.5.1. Installation of tubing system

When building a secondary fluid system it is important that all low points can be drained and that air purgers (deareators) are mounted in all high points where air can be gathered in the system. Air purgers should be mounted before a descending tube (see example in Figure 9.1.). Dimension the flow velocity in the tube system so that the liquid can bring along possible air bubbles to deairing clocks. A recommendation is to think of future expansion already at the time of installation by mounting valve sets and/or T-tubes in the tube system. All T-tubes not used should be blocked.

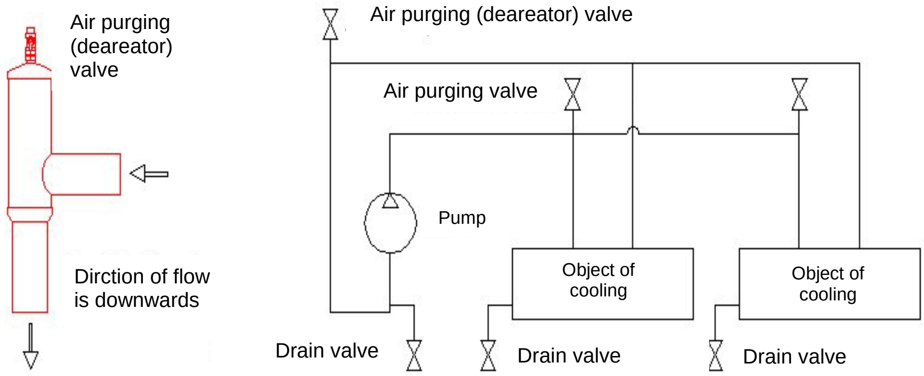


Figure 9.1. Positioning of air purging and drain valves in a system (example)

9.5.2. Installation of the tubing

Tubing is fixed with consideration of the weight as well as the stress during pressure testing and operation. The risk of corrosion should be considered when choosing brackets or fixing devices. Fixing and suspension devices of steel that are not zincked shall be painted for rust protection.

Tubing shall be installed in is horizontal location to avoid air pockets, that is in lengthwise direction of the tubing. A simple means to avoid mix-up of inlet and return tubes is to mark the tubes with flow direction arrows already during mounting (see 9.5.14. ”Marking of tubing”).

Be careful to keep the distance between the fixing points of the tube dimension (See suitable distances in Table 9.1.)

A standing tube is mounted in such a way that stress due to self-weight is placed to suitably situated pivit point, in order that horizontal tube is not burdened.

Tabell 9.1. Distance between hangers for horizontal tubing

Pipe dimension DN/mm	Maximum distance (m) between pipe hangers Steel	Copper	PVC, PEH (Plastic)	PEL (Plastic)
20/22	2.5	1.25	0.7	0.3
25/28	2.5	2.5	0.9	0.4
32/35	2.5	2.5	1.0	0.4
40/42	2.5	2.5	1.1	0.5
50/54	3.0	2.5	1.2	0.5
65/70	4.0	2.5	1.4	0.6
65/76.1	4.0	3.0	1.5	0.6
80/88.9	5.0	3.0	1.6	0.7
100/108	5.0	3.0	1.7	0.7

Tubing must be installed considering that the expansion can be taken up in view of temperature differences in the system (See Table 9.2).

Tabell 9.2. Copper pipe coefficient of expansion/contraction (mm)

Pipe length (m)	Temperature difference of media (ºC·m) 40	50	60	70	80	90	100
1	0.7	0.8	1.0	1.2	1.3	1.5	1.7
2	1.3	1.7	2.0	2.3	2.7	3.0	3.3
3	2.0	2.5	3.0	3.5	4.0	4.5	5.5
4	2.7	3.3	4.0	4.7	5.3	6.0	6.6
5	3.3	4.2	5.0	5.8	6.6	7.5	8.3
6	4.0	5.0	6.0	7.0	8.0	9.0	10.0
7	4.7	5.8	7.0	8.1	9.3	10.5	11.6
8	5.3	6.6	8.0	9.3	10.6	12.0	13.3
9	6.0	7.5	9.0	10.5	12.0	13.5	15.0
10	6.6	8.3	10.0	11.6	13.3	15.0	16.6

All brackets or fixing points are to be adjusted to the location so that the insulation is not compressed or in any other way damaged. Most insulation suppliers have fixing systems that are developd with a hard core of polyurethane foam and with contact surfaces of cell rubber for satisfying gluing with the rest of the tubing insulation. Pressure shims can also be used to distibute the weight of the tubing and the liquid content.

When tubings are located above the cooling objects, portruding tubes (droppers) shall always portrude beneath the main tubing all to avoid the forming of air-locks. If this is not possible and portruding tube has to be made above the main tubing then satisfying air purging has to be made from this purtruding tube. If the main tubing is installed below the cooling objects then tubes must portrude from the top oof the main tubing.

9.5.3. Tube lead-through in wall, beam or joist floor

Tube sleeves of plastic (where accepted) or steel shall be installed to give better protection for the insulation through fire wall. Sealing between tube and insulation shall be performed on both sides of the lead-through.

9.5.4. Pressure testing

Calculated pressure is the highest calculated pressure in the system and the pressure is secured with a safety relief valve at the corresponding pressure level. The installed tube system shall also at least be pressure tested with the corresponding test pressure level = 1,3 x calculated pressure (bar). Pressure testing shall be done before the tubings are insulated.
If water (H_2O) is used as pressure testing liquid all water must be removed before the antifreeze secondary fluid is charged. Other substances used for pressure testing are Nitrogen or compresed air (recommended). During pressure testing the pressure should be stable for at least 4 (four) hours and then a thorough ocular inspection of the system be performed.

9.5.5. Thermal insulation of soft cellular plastic

Always check with the supplier of the insulation with regard to the choice of insulation thickness that always is dependent on circumstances inside and outside the tube that is to be insulated.

The insulation shall to avoid joints along the tube preferably be thread on the tube when the tubing is mounted All joints along and across the tube as well as at ends shall be glued after thorough cleaning. The insulation should at all joints be glued to the tube.

Joints must not be taped. Insulation of discs are fully glued.

Tubings that normally are not run through are only insulated at a cold connection to prevent condense. At chillers the insulation is drawn well inside the drip shield.

9.5.6. Termal insulation of polyurethane cellular plastic

A number of manufacturers of prefabricated tubes/bends with polyurethane foam insulation that give much better insulation qualities compared with soft cellular plastic. This is often economically advantageous for larger tube dimensions than for smaller. The reduction of the mounting time and the improvement in quality can also be decisive. An alternative to prefabricated tubing is to install a light metal sleeve and then foam on site.

9.5.7. Connection methods and sealing material

Be careful to choose gaskets and sealing material suitable for the liquid that wil be used.

Recommended connection smethods and sealing material:

1. Welding
2. Hard soldering (copper - silver solder)
3. Pressure coupling with EPDM sealing (should not be used in freezer applications)
4. Flange connections (use rubber gaskets type EPDM)
5. Thread connections (only in exceptional cases, can then be sealed with joint sealant type Loctite, Omnifit or flax and paste/putty).

Compession fittings or fittingssuch as ”conex” can create great problem due to the high temperature differences at defrostings. Freezing stress due to condense water in these connections is not ususual and must be taken seriously.

If anyhow these types of couplings are used the copper tubes must be of a hard type. If soldering is done near the connection cooling is needed to prevent the material to soften. If the cooling is not sufficient support sockets should be mounted before the coupling is applied.

<u>Note</u>: <u>Never</u> use compession fitting in a freezer secondary fluid system.

Thread connections can be used together with the right type of joint filling material.

A good alternative is flax together with a soft spreader such as ”unipac”.

Before the flax is applied the threads are scraped to give the flax a good attachment. Take out an adequate amount of flax and fill up the threads in the thread direction. Finish by applying the sealing spreader on the flax. <u>Note</u>: Never draw the thread to the bottom position.

Keep in mind that certain components may need to be disassemblied during service or maintenance visits. With a mix of solder coupling and solder nipple/muff at the installation the work at a possible disassembling is simplified.

At all joints of a thread connection two tools should always be used if no work bench can be utilized (one as utility dolly and one as drawing).

9.5.8. Tube connection of two, three or more refrigerating units

Always make sure to make a well balanced system with tube lengths. If balance can not obtained with tube dimensions adjustment valves need to be applied to achieve balance. If there is imbalance then the flow will always pass the ”easiest way” and as a result the refrigerating unit that has the poorest position will get an unfavorble evaporation temperature. The lower temperature in the system the more it affects the imbalance.

9.5.9. Installation of the tubing in cooling cabinets

Always let the cabinet manufacturer at the start give direction on how to install the tubing inside the cooling cabinet.

Keep in mind that with correct tube dimensioning balance and correct temperatures will be achieved in the different units.

Avoid install the tubings in such a way that future service or maintenance, such as change of fans, is obstructed. Installation of tubing inside the cabinet should not cause any choking of the cabinet air flow or similar.

9.5.10. Installation of the tubing in freezer cabinets

Always let the cabinet manufacturer at the start give direction on how to install the tubing inside the freezer cabinet.

Tubing should be installed between the cooling coils and fans so that, during defrost, that area is warmed up above 0ºC (fans are then off but are put on just after defrost). One alternative is to install the tubing at a suitable place and then box the piping with sheet cover and insulate with polyurethane foam. Closed cell rubber (neoprane) insulation is not suitable due to climate and humidity inside the cabinet.

Keep in mind that with correct tube dimensioning balance and correct temperatures will be achieved in the different units.

Avoid install the tubings in such a way that future service or maintenance, such as change of fans, is obstructed. Installation of tubing inside the cabinet should not cause any choking of the cabinet air flow or similar.

Make sure that the installation of tubing will be correct from the beginning to avoid future problems with ice formation.
Several criteria need to be met depending on cabinet manufacturing and defrost technology.

9.5.11. Pipe connection to cooling and freezer chillers

Tubing must always be connected in such a way to achieve a cross flow heat exchanging over cooling coils takes place. The manufacturer must always indicate flow drawings of tube connections with flow direction arrows. Insulation of tubings must always over the top of the drip tray edge.

All cooling rooms operating below +2ºC need to have drip tray heaters that operate during the defrost period.

The coils should have air purging point on the highest level and a drainage point on the lowest level.

9.5.12. Probe positioning

Avoid welded probe-type pots in tubings in secondary fluid systems below +1ºC. If this type of probe pot still is used, it must be placed vertically (in horizontal tubing) and be filled with has to be done glycerol and be sealed off in order that no condense water can come in and through icing blow the tube.

A safer method is to fix the probe directly on to the surface of the tube. Make sure to get as good contact surface as possible. Contact spreader and aliminum tape give a good reinforcement of the contact surface.

9.5.13. Flow measurements

A flow measurement protocol should always be compiled after the plant has been started up.

A universal instrument can cover most manufacturers of balancing valves.

A coarse adjustment of the varios valves of the cooling objects will facilitate the measuring. Always start to measure the higest flows in the system.

Keep in mind that temperatures change density of the liquid and hence the flow. Compensation has to be made when making adjustments are made outside given operational conditions.

Try to make the measurement quick and effective as sealings in the measuring outlets are stretched and may leak a lot when measuring nipples are removed.

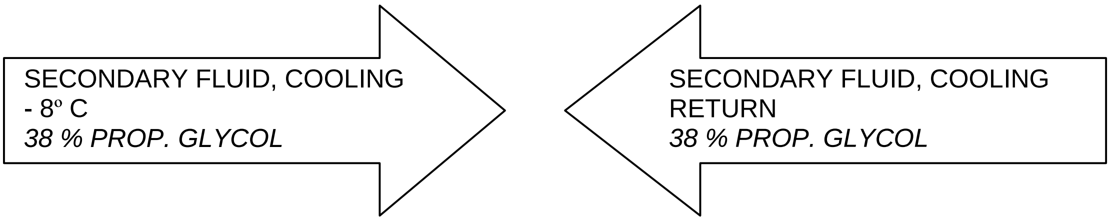
When measuring flow of a freezer system this measurement should not be done at a temperature below-8°C due to the risk of frost bite on hands from leeking freezer fluid.

9.5.14. Marking of secondary fluid tubing

Secondary fluid tubes are marked with flow directional arrows and text containing at least: System label, beteckning, outgoing temp/return and type of fluid with wt-% ro freezing protection. Given fluids are exemples of possible secondary fluids in these applications.

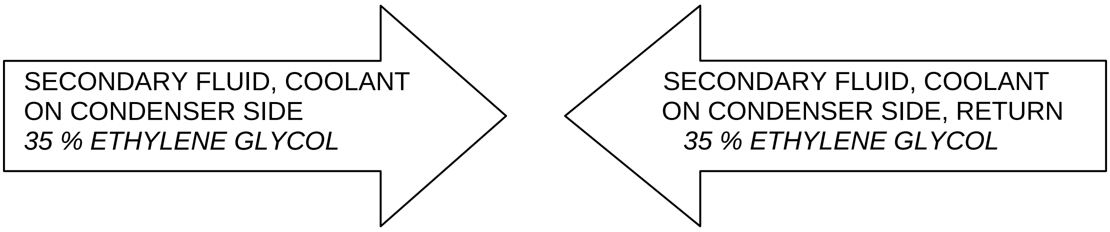
Marking of secondary fluid tubing for cooling

Secondary fluid tubing for cooling (rooms, cabinets) may be marked as indicated in arrows:



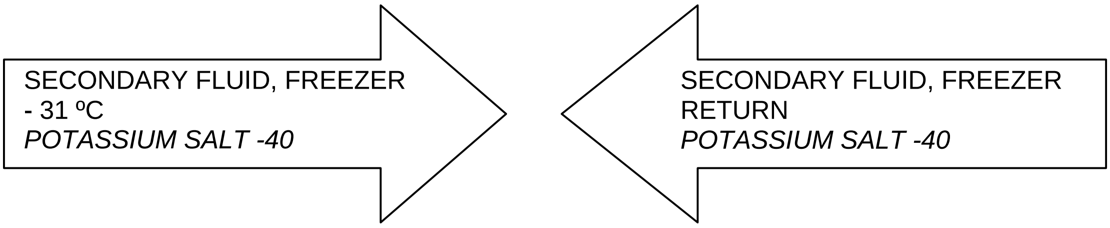
Marking of secondary fluid tubing for coolant on condenser side

Secondary fluid tubing for coolant on condenser side may be marked as indicated in arrows:



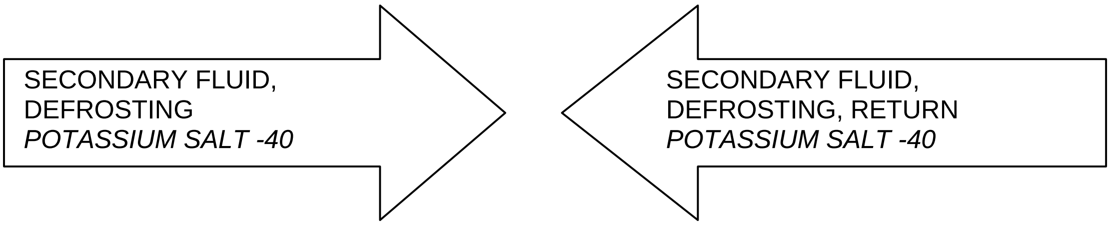
Marking of secondary fluid tubing for freezer

Secondary fluid tubing for freezers may be marked as indicated in arrows below:



Marking of secondary fluid tubing for defrosting

Secondary fluid tubing for defrosting may be marked as indicated in arrows below:



9.6. Example of documentation (DEM, Frigotech, 2001)

9.6.1. Installation protocol for fluid system

Plant: ____________________

Address: ______________________

System name: ________________

Calculation of total static pressure

Highest level in system = ____ m x 0,1 = _______ bar

Static gauge pressure = ____ m x 0,1 = _______ bar

Total calculated static pressure = ____ m x 0,1 =_______ bar

Calculation on alarm setting

Static height = ____ m x 0,1 = ______ bar

Security limit for alarm = ____ m x 0,1 = _______ bar

Lowest pressure for alarm = ____ m x 0,1 = _______ bar

Correction of operating pressure = ____ m x 0,1 = _______ bar

Alarm at operating pressure = ☐ m x 0,1 = ☐ bar

Pressure difference setting = ____ m x 0,1 = _______ bar

Pressure setting = ____ m x 0,1 = _______ bar

Correction of operating pressure = ____ m x 0,1 = _______ bar

Pressure setting at operating pressure = ☐ m x 0,1 = ☐ bar

Differential pressure, pumps

Pump A before pump = ____bar after = ______ bar

Pump B before pump = ____bar after = _______ bar

Date: _____________

Checked by: _____________

Print name:

9.6.2. Test and och tightness control

Plant: ____

Address: ____

System name: ____

Material:

Steele ☐
Stainless ☐
Copper ☐
plastic ☐

Test medium

Compressed air ☐
Water ☐
Nitrogen ☐

Test data

Manometer grading *From/to* ________-________*bar*

Read test presure ________*bar*

Test pressure time ________*tim*

Ambient air temperature ________*ºC*

Calculatin pressure ________*bar*

Remark:

____

____

____

Date: ____________________

Controlled by: ____

Print name: ____

9.6.3. Service/maintenance - Indirect refrigeration systems

1(2)

Plant: ____________________

Address: ____________________

Pump modul no.: ____________________

System name: ____________________

Control of Pumps		**Ja**	**Nej**
	Rust onset	☐	☐
	Leakage sealings	☐	☐
	Leakage couplings, flange connections	☐	☐
	Abnormal sound from pump(s)	☐	☐
	Pump changing function OK	☐	☐

Comment: ____________________

____

Ocular control of tubes and couplings of the system

No remark ☐

Remark ☐

Comment: ____________________

____

Air purging of system

Large quantity of air could be evacuated from the system ☐

Smaller quantity of air could be evacuated from the system ☐

Large quantity of air remains in the system ☐

Smaller quantity of air remains in the system ☐

No noticable quantity of air remains in the system ☐

Comment: ____________________

____

Pressure setting over pump(s)

Pressure before pump __________Bar

Pressure after pump __________Bar

Differential pressure __________Bar

Comment: ____________________

2(2)

Differential pressure over dirt filter

Pressure before dirt filter __________Bar

Pressure after dirt filter __________Bar

Differential pressure __________Bar

If the differential pressure is more than 0,2 Bar the filter should be cleaned

Differential pressure over dirt filter after cleaning

Pressure before dirt filter __________Bar

Pressure after dirt filter __________Bar

Differential pressure __________Bar

Comment: ____

____

Sample taking of antifreeze liquid (at least 1dl) to be sent for analysis ☐

Control of freezing protection Type__________________ Freezing point _________°C

Control of pH-value pH _________

Comment: ____

____

Control of total static pressure Total static pressure __________Bar

Control of expansion vessel pre-pressure Pressure in expansion vessel __________Bar

Control of low pressure alarm Alarm release pressure __________Bar

Comment: ____

____

Date : ______________________

Controlled by: ____

Print name: ____

10. Optimization of secondary systems

10.1. General viewpoints on optimization (E. Granryd)

The word *optimization* is often used uncritically. Therefore, it might be relevant to start with a short discussion on the purpose of optimization.

All optimizations involve a choice where pros and cons are weighed against each other. It is apparent that we must clearly define what considerations we are taking into account. Of fundamental importance is, hence, to define the criteria for the optimization.

For example, we might want to optimize or design in order to reach:

- Minimum *total cost for the user* including all costs for investment and operation over a certain period, more or less identical to the concept of minimum "Life Cycle Cost".

Instead we might limit our considerations to design the system for

- Minimum *first cost*

... or:

- Minimum *operational cost*

… or:

- Minimum cost of *material and manufacture*

In some cases there might be more specific consideration that are of importance:

- Shortest possible *time for the design* of the plant
- Minimum *physical volume*
- Minimum *weight*

One should also be aware of the fact that it is tempting for a contractor to chose a design which will result in:

- Largest possible income for the company constructing the plant...?

The different criteria will result in different designs. It is apparent that the word *"optimal"* can mean almost anything, unless we clearly define what is meant!

10.2. Various types of problems (E. Granryd)

There are two different ways of formulating optimization problems -- *technical* and *economical* problems.

When it comes to *economical optimization problems,* the results depend on price on material, energy and capital (interest rates) as well as operation time and expected life expectancy. In order to determine the operational costs there is also a need to know (or guess) what energy prices there will be in the future (during the years of operation for the plant). A *technically optimal construction* is independent of most of the mentioned factors and is in that way more general.

Refrigeration systems are built of various components – evaporator, compressor, condenser, expansion valve, pumps, fans, etc. There are many ways to form each one of these components. As an example, the task of a good evaporator is to take up (desired) *cooling capacity* from a heat source with *certain temperature* and to deliver evaporated refrigerant vapor to the compressor with as *high pressure* as possible, or expressed in an other way to make the temperature difference as small as possible. Here are many *technical* formulations of problems when it comes to the design of the evaporator that at times does not have to affect the price, but the function.

An *economic* dimensioning of the evaporator deals more with determining ”how large evaporator” you will afford, or formulated in another way: *What temperature difference* is economically motivated when choosing an evaporator. The economical optimization should appropriately be done for a construction as near a technically optimal design as possible.

10.3. Summary of the following optimization sections (E. Granryd)

In the following chapter there are two sections showing various aspects at optimizing secondary working fluid systems for refrigeration plants or heat pumps. The treatment in both sections is limited to the secondary working fluid system.

In **Section 10.4** Jan Erik Nowacki considers the construction of an indirect system where economic considerations are included. The difference in economical optimal tube dimensions when using different secondary working fluids and a discussion is made of fluid velocities that are economically optimal. A consideration is also made to answer the question how deep it is economically motivated to bore the hole for a normal rock heat pump well.

Section 10.5 gives an example of a simple technical optimization: Eric Granryd asks the question how to run a plant that already is built but where you, through speed control or other way, can choose the flow in a secondary fluid system. The example illustrates how the pumping power affects the cooling capacity and the coefficient of performance, COP_1 or COP_2. The example shows that there is a flow that gives the highest cooling capacity and another (lower) flow that gives the largest COP and thereby the least total energy demand. Simple correlations are given for optimal pumping power for the two cases.

In Section 10.6, Jan Erik Nowacki deals with several common (standard) problems in a simplified form. Finally in an appendix Tommy Nilsson summarize a few possible principles to connect secondary systems in heat pumps.

Section 10.4 (J.-E. Nowacki)

Economically optimal refrigeration and heat pump systems

Purpose

A simplified analysis will be presented as an attempt to answer the following questions:

- What difference in performance will there be with different secondary fluids?
- What dimensions are optimal for liquid pipes in indirect systems?
- What flow rates can be used with conventional pipes in ground source systems?
- What depth is optimal to drill for a vertical ground heat source in rock?

Method

The purpose with this simplified study is to demonstrate how one can balance cost of capital and cost of operation of indirect refrigeration systems. Using pipes of small diameters in a secondary loop system, results in a low investment, but a high pumping cost. Using large diameters of piping the investment will be high but the pumping costs will be low.

Estimations of the efficiency of ground source heat exchangers where the secondary fluid is in focus will be used as an illustration for the complex problem of simultaneous fluid transport and heat transfer. The flowing secondary fluid extracts heat though the wall of the pipe. This study is limited to PEM-materials in the tube, but the method may be applied to other types of materials, if appropriate data and relations are used.

The method that is used is just to add the operating costs and the capital cost of investment (equal to investment times the annuity) and use Excel functions to find optimum solutions. Examples of problems treated are to find optimal tube diameter, optimal flow rates, or optimal depth of the borehole. Since the optimization is based on "equations" rater than on data in tables, formulas has to be defined for, for instance how the outer an inner diameters (dy/di) are tied together in commercially available pipes (instead of discrete dy/di tables).

The cost of capital is estimated by means of an annuity (which thus includes interest rate and annual depreciation). The operating costs depend on the estimated average energy price of energy during the years that the plant is assumed to be in operation. Furthermore, the temperature drop in the pipes is taken into account as well as the required pump work.

The investment for tubes has been assumed proportional to the volume of tube material. This seem to be reasonably well based assumption at least for PEM-materials (standard deviations was found to be 11% within a wide range)

A matter of discussion is whether the pump efficiency should be taken into account in the optimization. A liquid centrifugal pump has often efficiency as low as 20% and it obviously requires much more energy for pumping than needed. The losses tied to the pump should really be optimized in a similar fashion as the pipes. A cheap pump results often in a low efficiency.

In all calculations it is assumed that there the transition between laminar and turbulent flow occurs at Reynolds number, $Re = 2300$. This has been questioned by among others Tommy Nilsson, who wants to investigate if this assumption is correct for flow in very long smooth tubes. (Compare attachment 10.7 in the end of chapter 10)

Economic dimensions of tubes and bore hole heat exchangers

Transport of one-phase fluids in tubes without heat exchange

Technical background [1]

Flow of a fluid through a pipe causes a loss of fluid pressure. This pressure drop multiplied by the volume flow rate represents a power loss for each meter of pipe.

The pressure loss per meter of tube can, according to [1] be expressed as:

$$\Delta p' = \frac{f_1 * \rho * w^2}{d_h} \qquad \textit{from } [1]\, \textit{equation } 10.18 \qquad (1)$$

where: f_1 friction factor, to be estimated by relations that will be given,

ρ density of fluid, kg/m^3

w flow velocity (= V/A), m/s

$$d_h = \frac{4 * A}{U} = D_i \quad (\textit{for a circular tube}) \qquad (2)$$

A internal cross sectional area of the tube, m^2

U internal perimeter of the tube, m

According to [1] the friction factor f_1 can be estimated by the following relations where the Reynolds number, Re, is an important parameter:

$$\mathrm{Re} = \frac{w * d_h}{\nu} \qquad (3)$$

ν the kinematic viscosity of the fluid = η/ρ (η is the dynamic viscosity, ρ is the density)

If Re is < 2300 the flow is assumed laminar and then f_1 is estimated by:

$$f_1 = \frac{32}{\mathrm{Re}} \qquad \textit{from} [1]\, \textit{equation } 10.26b \qquad (4)$$

According to Gnielinski the following relation is valid within a wide range, for smooth tubes and $2300 < Re < 5*10^6$:

$$f_1 = \frac{1}{2 \cdot [0.79 \cdot \ln(\mathrm{Re}) - 1.64]^2} \qquad \textit{from [1] equation 10.31d} \qquad (5a)$$

By numerical analysis, it can be found that the expression can be simplified to:

$$f_1 = 0{,}2767 * \mathrm{Re}^{-0{,}312} \qquad (5b)$$

…without loosing noticeably in the accuracy (the deviation is max 0.8% between $Re = 2300$ and $Re = 10000$).

The tube itself consists of a certain material with a certain tensile strength. The pipe is exposed to an internal pressure, which the pipe wall must be able to withstand. This means that the wall thickness must be a constant times the inner diameter, see Figure 10.4.1.

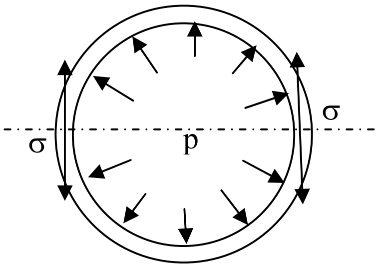


Figure 10.4.1. Section of a tube

A simple force balance for a thin walled tube with outer diameter d_y and inner diameter d_i exposed to internal pressure *p* where the tensile strength σ balances the forces caused by the internal and external pressure difference gives the following relation $(d_y - d_i) \cdot \sigma_s = p_{max} \cdot d_i$. which also can be expressed:

$$\frac{d_y}{d_i} > \frac{p_{\max}}{\sigma_s} + 1 = k_d \qquad (6)$$

where σ_s is the dimensioning tensile strength of the tube material

k_d is a quotient of diameters (which in the following treatment is regarded as a constant).

Some of the assumptions made here can be questionable. Is it really the internal pressure that sets the thickness of the wall? – Or is it for instance the safety for external mechanical exposure? A small investigation including pipes for several classes of pressure, PE-qualities and tube diameters show that a standard deviation of the estimated tensile strength. σ_s in Equation 6 varies not more than 20%. This supports the assumption that the internal excess pressure and the diameters are the main parameters for choosing wall thickness.

All equations are of course without meaning if we have not decided the purpose of the pipe. Usually we want the pipe to transfer a certain flow rate of fluid q_v.

The equations therefore are tied together by the following relation:

$$q_v = \frac{\pi * D_i^2 * w}{4} = const. \qquad (7)$$

Economic background

In general, the concept of the”Present Value” has been used. This means that an annual cost of operation is divided by the annuity in order to estimate the present value. With a continuous interest rate (applied to for instance every second) the well-known bank formula for the annuity coverts into the simpler equation:

$$a = \frac{r}{(1 - e^{-rt})} \tag{8}$$

where: a is the annuity ($1/a$ becomes the ”present value factor” for a number of periods of equal payments)

r interest rate

t time period.

There is normally an extremely small difference between this formula and the formula for capitalizing in discrete time intervals (for instance once a year).

In order to estimate the operating cost for one meter of tube we will begin with multiplying the pressure drop by the flow rate as discussed earlier. This gives us the power that is necessary for pumping. This number must then be multiplied by the annual time (in hours) that this flow rate is maintained in order to derive the annual energy demand (expressed in Wh). Dividing the result by 1000, we have the result expressed in kWh and dividing with the annuity a we have the capitalized value of the energy.

The capitalized operation cost (the Present Value) for “**dr**iving” one meter of tube in **laminar** flow during the yearly operating time period τ will thus be:

$$C_{drlam} = \underset{pressure\_drop}{\frac{32 * \mathrm{Re}^{-1} * \rho * w^2}{d}} * \underset{flow}{\frac{w * d^2 * \pi}{4}} * \underset{economy}{\frac{\tau * P_e}{a * 1000}} \tag{9}$$

Where τ is the operating hours per year

P_e is the cost of energy expressed e.g. in SEK/kWh (or €/kWh or $/kWh)

If instead the flow is turbulent (using friction factor as given in Eq. 5b) the capitalized operation cost of one meter will be:

$$C_{drturb} = \underset{pressure\_drop}{\frac{0.0408 * \mathrm{Re}^{-0.0738} * \rho * w^2}{d_i}} * \underset{flow}{\frac{w * d_i^{\,2} * \pi}{4}} * \underset{economy}{\frac{\tau * P_e}{a * 1000}} \tag{10}$$

The investment cost for the tube itself is obviously the equal to the cost of the tube because the investment is assumed to be done “now”.

The cost for the material in the tube becomes:

$$C_{inv} = P_m * \frac{\pi * (d_y^{\,2} - d_i^{\,2})}{4} = P_m * \frac{\pi * d_i^{\,2}(k_d^{\,2} - 1)}{4} \tag{11}$$

where: P_m is the cost volumetric material cost (it may also include indirect costs like cost of installation), SEK/m^3 (or €/m^3 or $/m^3).

k_d is the diameter quotient d_y/d_i as derived in eq. 6

The total present value of the costs for one meter of tube becomes:

$$C_{tot} = C_{dr} + C_{inv} \tag{12}$$

The cost of service etc. for the tube has been neglected.

Calculations

Based on specified assumptions we can for instance estimate which tube diameter is optimal. The following example is based on the following assumptions:

Diameter ratio	dy/di =	1,14	
Freezing point for secondary fluid:		-15°C	
Temperature:		0°C	
Interest rate:		5%	
Time period		15 years	resulting in an annuity of: $a = 9.5\%$
Energy price:	$P_e =$	1.00 SEK/kWh	
Annual operating time:		4500 h/year	
Material price:	$P_m =$	30000 SEK/m^3	

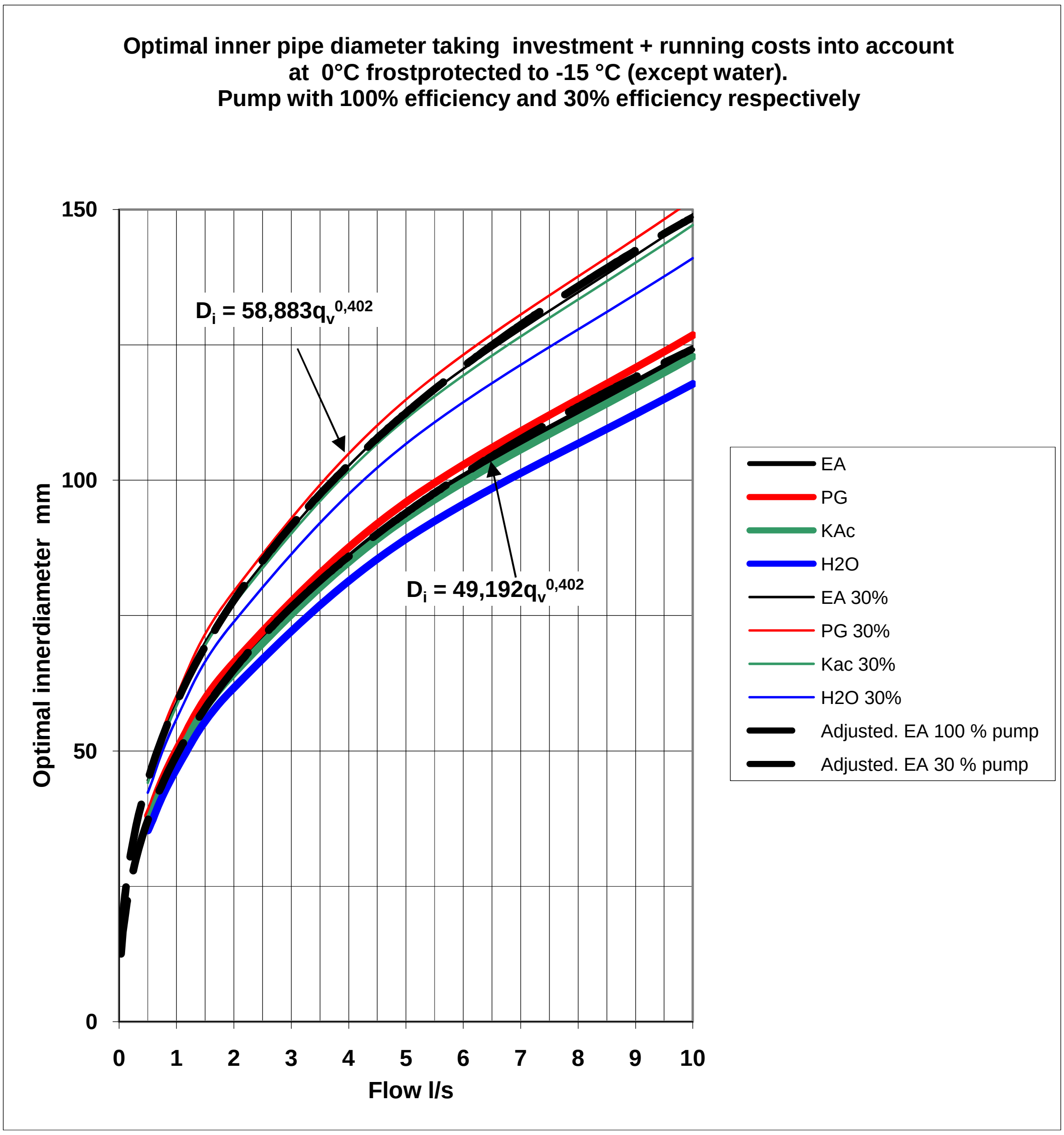
Comment: The assumption about the price of material is equivalent of 8 SEK/m (or in the magnitude a $ or € per m) for a PE-tube 40/35 mm.

Optimal tube diameters are estimated for different flow rates and different media with 100% and 30% pump (and pump motor) efficiency including cost of capital. Results are given in Figure 10.4.2. (EA=EthylAlcohol, PG= PropylenGlycol, KAc=Potassium Acetate and H_2O = pure water.)

Example: If we have a pump with 30% efficiency and use Ethyl alcohol as secondary fluid and other data as given in the data in previous table – what will then the optimal tube diameter be if the intended flow rate is $q_v = 0,3$ liters/s?

Answer: The optimal inner diameter $d_{Oopt} = 1,05 * 0,0003^{\mathbf{0,4225}} = 0,034$ m or 34 mm. The optimal fluid velocity in the tube will then be approximately 0,33 m/s.

What influence does the pump efficiency have? From the diagram, one can learn that a pump efficiency of 30% instead of 100%, increases the optimal tube diameter by approximately 19% compared to the case with a perfect pump. (This also means that the optimal velocity is 29% lower and the required ideal pumping power is reduced by 41%).



Figur 10.4.2. Optimal tube diameters for different flow rates (in the equations D_i is the optimal inner diameter in m and q_v is the flow rate in m^3/s)

The optimal tube diameter is influenced only to a minor extent of the type of fluid (standard deviation for the average ca 2%). For other concentrations of the antifreeze solutions for instance down to -30°C (instead of -15°C as on the figure) the influence of fluid properties on the optimal tube diameter is somewhat more pronounced.

Still another way to describe this is to study he optimal flow velocity as a function of the tube diameter, see Figure 10.4.3. In principle the same information is shown here as in Figure 10.4.2 but the ”accuracy” is perhaps better.
A conclusion from Figure 10.4.3 is that the optimal flow velocity, at a given tube diameter, is reduced by 35% - if a low (normal) pump efficiency is taken into account.

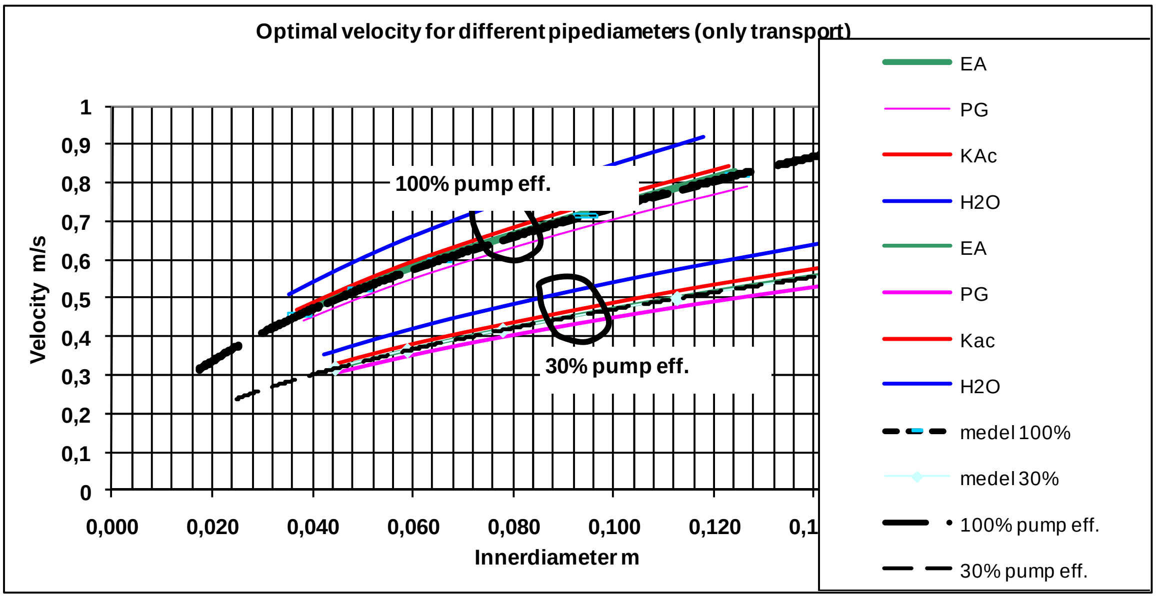


Figure 10.4.3. Optimal fluid velocity as a function of the tube diameter
(Notice: At velocities higher than about 1 m/s there is a risk for “corrosion” by erosion. This was not taken into account in the optimization process.)

Heat Exchanging pipes in vertical boreholes in rock

Technical background

The calculations here are limited to PEM-tubes with dimensions 40/35 mm – this is presently the most frequently used standard in Sweden. The situation is however somewhat more complicated compared to the case of pure fluid mass transfer in tubes due to the influence of temperature change of the fluid caused by the heat exchange resistance between the tube wall and the fluid. At low flow velocity, the heat exchange becomes poorer and thus the temperature of the fluid becomes lower for a given amount of heat energy from the ground. This also lowers the evaporating temperature in the heat pump and the energy needed to operate the compressor in increased. The loss associated with this can be handled in the calculations by estimating how many kWh of savings is lost due to a temperature drop of for instance 1°C.

This loss in energy savings is about 1.4% per degree lowered temperature of the heat source. The result has been obtained by using dimensioning programs from heat pump manufacturers, changing the incoming temperature of the heat source fluid from +2 to -2 °C, in a operating situation where the heat pump has a load coverage of 50-60% of the maximum heating power need of the house. To compensate for the lack of capacity from the heat source an auxiliary heat source must be added (otherwise, it will be cooler inside the house). The heat pump however operates less than 1000 h/year using this auxiliary heating. The rest of the hours during a year the heat pump operating time can simply be prolonged to compensate for the loss of capacity. The overall heating Seasonal Performance Factor (SPF) is however influenced much more (more than 4%). The example in Table 10.4.1 is an attempt to show these phenomenons and the relation between the 1.4% loss of energy from the ground and the 4% loss of SPF.

Table 10.4.1. A small loss of energy from the ground caused a greater loss of SPF

	Nominal ground temperature.	Nom. Ground Temperature. minus 1°C	Change %
From el. grid	10 000 kWh	10 326 kWh	+3.26 %
From heat source	23 300 kWh	22 974 kWh	-1.40 %
SPF	3.33	3.19	-4.20 % (or even more!)

Typical ground rock in Central Sweden is granite, and a ground source is often dimensioned for about 40 W heat extraction per m borehole, (or 20 W/m of U-tube). With 40 W/m borehole the annual heat extraction is about 160 kWh/m bore hole (or 80 kWh per meter of tube). These numbers are hence affected by 1.4% per m if the fluid temperature is lowered by one degree on the average.

To calculate the temperature difference between tube wall and fluid the following equation given by Dittus-Boelter for turbulent flow is often used:

$$Nu = 0.023 * \mathrm{Re}^{0.8} * \mathrm{Pr}^{0.4} \qquad \textit{Dittus – Boelter, from}\,[1]\,\textit{eq.}\,11.66a \qquad (13)$$

If the flow becomes laminar the local Nusselts number approaches a constant value of Nu = 3.66 (for constant wall temperature) or Nu = 4.36 for constant heat flux. The values given represent the limits of Nu-numbers at fully developed laminar flow in long circular tubes.

The equation given by Gnielinski is however more suited for Reynolds numbers just above Re = 2300. The only disadvantage is that it is a little more complicated. It reads:

$$Nu = \frac{(f_1/4) * (\mathrm{Re} - 1000) * \mathrm{Pr}}{1 + 12{,}7 * (f_1/4)^{0{,}5} * (\mathrm{Pr}^{2/3} - 1)} \qquad \textit{Gnielinski, from}\,[1]\,\textit{eq.}\,11.66b \qquad (14)$$

One can however simplify the Gnielinski relation so that it more suited for the intended optimization procedure. Without loosing noticeable in accuracy, it can be written:

$$Nu = 0.0148 * (\mathrm{Re} - 992)^{0{,}87} * \mathrm{Pr}^{0{,}36} \qquad \textit{Simplified Gnielinski} \qquad (15)$$

For values of Re between 2300 and 20 000 and a variation of Pr between 10 and 100 the maximal deviation between the simplified Gnielinski and the original is less than about 3%.

If the heat load from the borehole is limited to 30 W/m the temperature of the secondary fluid will increase ~2°C on the average compared to a case where the heat load is 40 W/m while for a case where 50 W/m is extracted the temperature is decreased by ~2°C. At 40 W/m, the incoming fluid from the borehole is often about 0°C on the average during the operating hours in a climate of Stockholm.

Economic assumptions

For the optimization, is it assumed that a given heat pump is already chosen but that we have a freedom to choose the depth of the borehole and which flow rate of the secondary fluid is to be used. The flow resistance in the heat pump evaporator is not included since the heat pump was considered as given. The borehole produces capitalized income and costs per meter of borehole in the average as follows:

+ Income due to "harvested" ground heat energy during the time period of operation
– Cost to drill the hole (300 SEK/m including pipes and work)
– Loss due to temperature differences on the inside of the pipes
– Cost to operate the pumps for the circulation of the fluid up and down the pipes.

The interest rate is set to 6% and the time of depreciation is (conservatively) set to 15 years. Actually, it is quite probable that the borehole can serve as heat source for a number of consecutive heat pumps - so one could argue for a much longer time of depreciation for the borehole. The capitalized gain **per meter of borehole** is illustrated in the following figure:

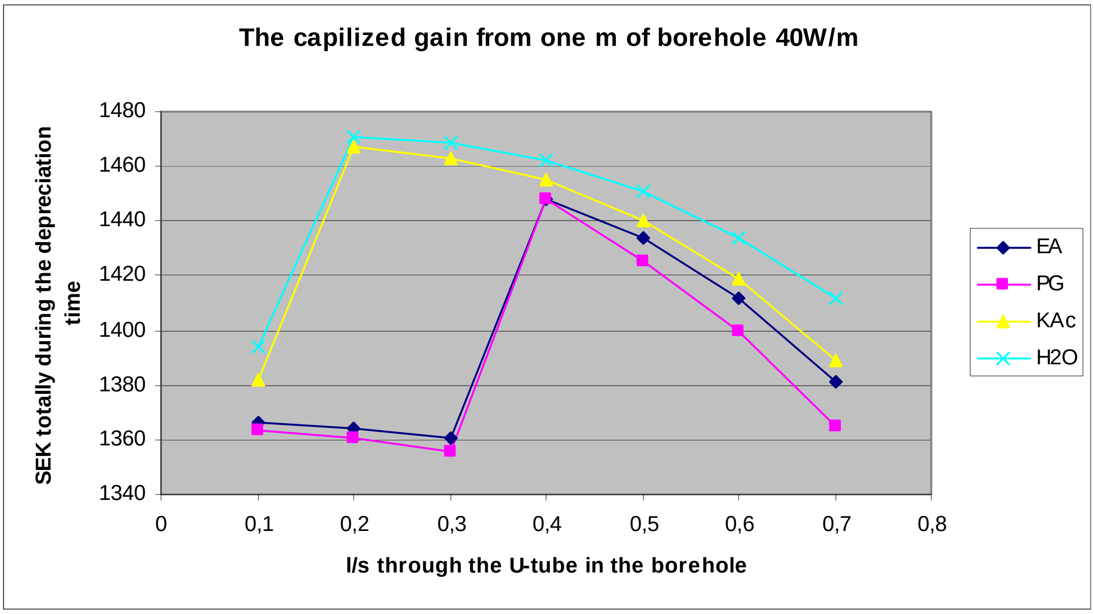


Figure 10.4.4. Capitalized gain per meter bore hole

The figure shows the capitalized gain of 15 years of heat extraction. The cost to drill the hole and the install the pipes are taken into account, as well as the loss in temperature due to heat transfer resistance in the pipes and the pumping cost to circulate the fluid through the heat source. The tube inside diameter is set to 35 mm.

From the Figure 10.4.4 we can learn that if for 40 W/m heat extraction and if EthylAlcohol (EA) or PropylenGlycol (PG) is used one should preferably use 0.4 – 0.5 l/s through the tubes. If instead PotassioumAcetate (KAc) is used the optimal flow is lower, one can use as low flow as 0,2 l/s. Even if the flow becomes "too" low, so that it becomes laminar, the losses are still relatively small. The "income" per meter borehole is then reduced by about 7 %. When operating with turbulent flow the difference between the different secondary fluids is quite small – the difference is only about 2% in the capitalized gain between the “best” and the “worst” fluid.

In an attempt to study the optimal drilling depth, the holes have been “normalized”. To decrease the heat extraction to only 30 W/m (instead of 40 W/m) the drilling depth has to be increased by 143% (when all the different losses are taken into account). If we for a case with 40 W/m had chosen to drill 150 m one would have to drill 214,5 m to limit the rate of heat extraction 30 W/m. These numbers have been estimated by means of a computer program from Viessmann. In similar fashion, one would only have to drill 75% of the depth to increase the rate of heat extraction to 50 W/m instead of 40 W/m. When drilling deeper there are of course also more meters to gain capitalized income from than for short holes.

The capitalized income, using optimal flow rates, for a case with an 8 kW heat pump, using KAc as secondary fluid, is given in the following figure. The total capitalized income is hence here achieved after by multiplying the gain per meter by the number of meters of the hole.

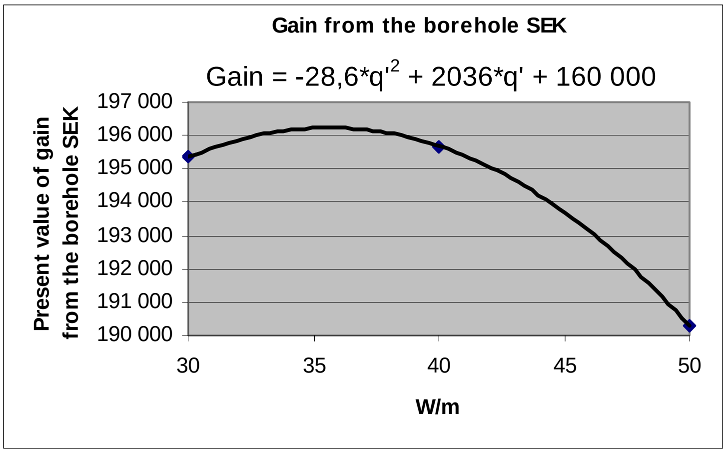


Figure 10.4.5. Gain for boreholes with three heat extraction rates, 30, 40 and 50 W/m.

As seen in Figure 10.4.5 it appears, for the conditions and with the simplifying assumptions as previously given, that the most favorable solution is to use an extraction rate of about 35 W/m. **An important conclusion of Figure 10.4.5 is however that the depth does not have a very large influence on the capitalized gain after all.**

The optimal depth corresponding to 35.6 W/m will give a capitalized gain, which is only 3% higher than at an extraction rate of 50 W/m. Of course, this optimization, based on primarily three basic assumptions, can be challenged. The science here is however not so exact that it pays off to optimize it in detail. There are in practice several questions affecting the result such as water flow in cracks in the rock, heat conduction between the pipes and the wall of the bore hole and short cuts due to heat conduction between the tubes carrying fluid flow down and up. These parameters may prove quite important. Another project within the Effsys2 program by José Acuna et al aims to investigate such factors in detail.

Reference

1) Ingvar Ekroth, Eric Granryd, Tillämpad termodynamik, KTH, Institutionen för Energiteknik, Stockholm 2005

Section 10.5 (E. Granryd)

What flow rates are technically justified in indirect refrigeration systems or heat pumps?

Introduction

In most indirect systems, a pump is required to circulate the fluid. It is easy to forget the power -- and energy – that is required for this pump. It is not uncommon that the power of the pump is in the order of 10 to 20 % of the power to operate the compressor. The purpose of this section to demonstrate how much power or energy one should spend on the circulation pump.

In the previous section of this handbook, economic relations were given for dimensioning of tubes and heat exchangers on the cold side of heat pumps or refrigerating systems. The starting point for the present section it is instead an existing plant where we can consider different strategies for operating pumps (or fans). We assume that the flow rates can be changed (by changing the speed or pumps can be replaced) with minimal cost implications. Solutions are thus not affected by economic considerations – *instead it is a pure technical optimization.*

The original background is a treatment of a closely related problem, Ref. [1]. In that paper it is shown that there are two optima for the air side fan power spent in evaporators of refrigerating systems; one resulting in maximum refrigerating capacity and another (much lower) for maximum COP2, equivalent to maximum energy saving. Simple equations are given in the reference for the two solutions.

A similar approach can be applied for the choice of circulation pump in an indirect system. A simple scheme is shown in Figure 10.5.1. We assume that we have an existing plant where the flow rates can be changed.

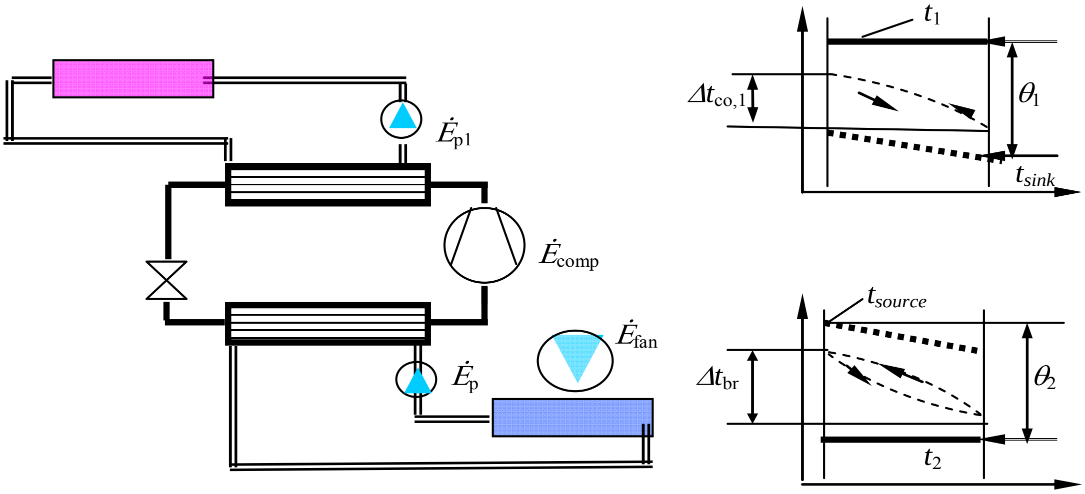


Figure 10.5.1. Scheme for an indirect refrigeration or heat pump system

Increasing the flow rate in for instance the low temperature circuit will be beneficial in terms of temperature change and heat transfer. The disadvantage is that it will require increasing pumping power.

Let us follow more in detail what happens if we increase fluid velocity on the evaporator side: It is obvious that the brine (secondary fluid) temperature change (Δt_{br} in Figure 10.5.1) will decrease when the flow rates increases. In addition, increasing flow will reduce the heat transfer resistance between the brine and the surfaces in the evaporator and the heat exchanger (the cooler). Both these effects mean that the overall temperature difference (θ_2 in *Fig.* 1) will decrease and, with given source temperature, the *evaporating temperature* t_2 *will increase*. This has the effect that the cooling capacity of the compressor (assuming given displacement) will increase and the compressor operating energy for a given cooling load will decrease. *However*, the energy spent for pumping will increase! The question is now: ***How should we balance the pump power in order to operate the system as efficient as possible***?

Example

Figure 10.5.2 shows for an example how the temperatures on the cold side will change if the flow rate is changed. As example, a plant is chosen with a fin coil air cooler where the air is cooled from +5 °C by counter flow heat exchange with the secondary fluid. In order to reduce the number of parameters it is assumed that the airflow is proportional to the flow of the secondary fluid. This means that the power demand of the fan and the pump both increase with increasing flow as shown in the Figure 10.5.3.

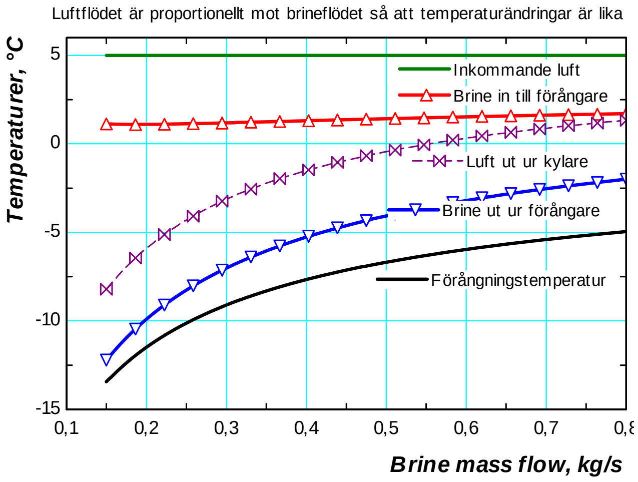


Figure 10.5.2. Temperatures for an example versus flow rate.

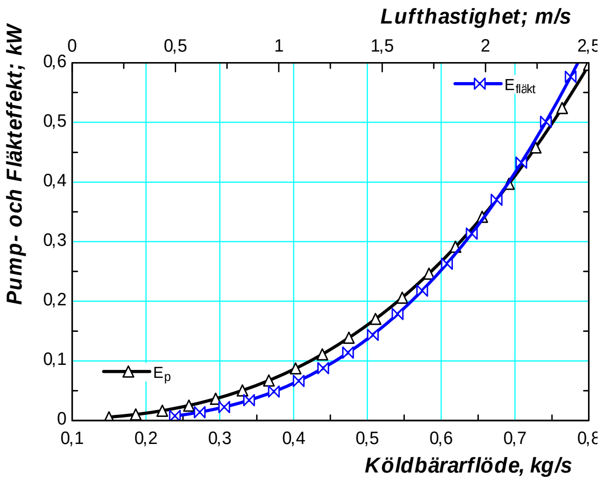


Figure 10.5.3. Power demand of pump and fan.

How will the capacity the COP of the system change with the flow rates in the secondary loop?

Results from a simulation of the system are shown in Figures 10.5.4-6 where the flow rates are varied (the airflow rate is varied proportionally to the liquid flow rate variation). For simplification the condensing temperature is set constant (t_1 = 35°C) and no pump is considered on the warm side. Data for the compressor are equivalent to a typical ‘good’ reciprocating compressor. Data for the evaporator are typical for a plate heat exchanger and a traditional finned coil is used as air cooler.

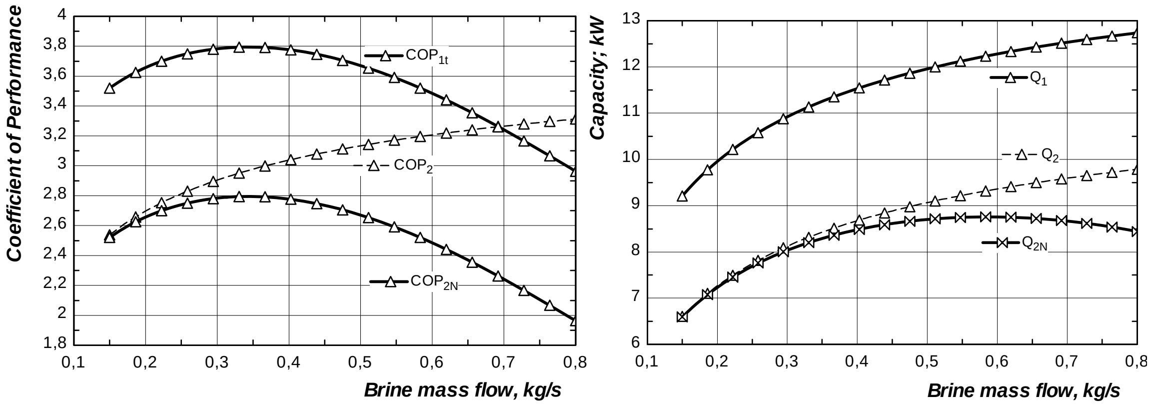


Figure 10.5.4. COP and Capacity versus brine mass flow rate for an example (t_{source} = +5°C; t_l=35°C)

Figure 10.5.4 gives an idea of the influence of the mass flow rate on the performance of the system and Figure 10.5.5 give the same information but now as a function the power for the secondary fluid pump (and fan) on the cold side ($E_p + E_{fan}$).

In the diagrams also the condenser capacity, $\dot{Q}_{1t}$, and the *heat pump COP*, (COP_{1t}) are shown. Theoretically, the maximum COP_{1t} does not occur at exactly the same Pumping Power Ratio as for maximum COP_{2N}, but in practice, the difference is insignificant and can hardly be noticed in the figures.

The diagram to the left in Figure 10.5.4 shows three curves for the cooling capacity, (Q_2), the heating capacity ($Q_1 = Q_2+E_k+E_{p1},$ if the system operates as a heat pump where all heat extraction is considered useful) and *“net cooling capacity”* (Q_{2N}). This curve shows the remaining useful capacity when the pump and fan power has been subtracted. The power used in the fan and pump does increase the cooling load. Q_{2N} is estimated by the following relation:

$$Q_{2N} = Q_2 - E_p - E_{fan} \qquad (1)$$

If the pump motor is cooled to external air (thus not by the fluid) the electric motor losses will not add to the load. For that case we can write $Q_{2N} = Q_2 - g_{elmotor} \cdot E_p - E_{fan}$. The curve in the diagram corresponds to the first case where the pump is cooled by the fluid.

As seen the cooling capacity (Q_2) and the condenser capacity (Q_1) increase monotonly by increasing fluid flow rate, but as can be expected there is a certain flow rate which will results in maximum of the ”*net cooling capacity*” (Q_{2N}).

The curves to the left in the figures show the coefficient of performance $COP_2 = Q_2/E_k$, while the ”*Net coefficient of performance*”, COP_{2N}, is determined including the also the power to the pump and fan:

$$COP_{2N} = Q_{2N}/(E_k+E_p+E_{fan}) \qquad (2)$$

For the case of a heat pump it is instead:

$$COP_{1t}= (Q_1+E_{p1})/(E_k+E_p+E_{fan}+E_{p1}) \qquad (3)$$

From the Figure 10.5.4 we can observe that the best COP_{2N} (as well as COP_{1t}) is obtained at about the half flow rate compared to the one that result in maximum net cooling capacity Q_{2N}. This conclusion is actually quite general. (Theoretical one should expect that the maximum COP_{1t} for heat pump operation would be at al slightly larger flow rate than for COP_{2N}, but the difference is so small that it can is not seen in the figures and it can be ignored in practice.)

Half the flow means that the pump power is reduced by a factor of 6 to 8 (somewhat depending on the type of flow pump efficiency etc). This is shown more explicitly in Figure 10.5.5 (even in the curve for the Q2N shows a very flat maximum). On the contrary to this, there is a more pronounced optimum for the *COP*.

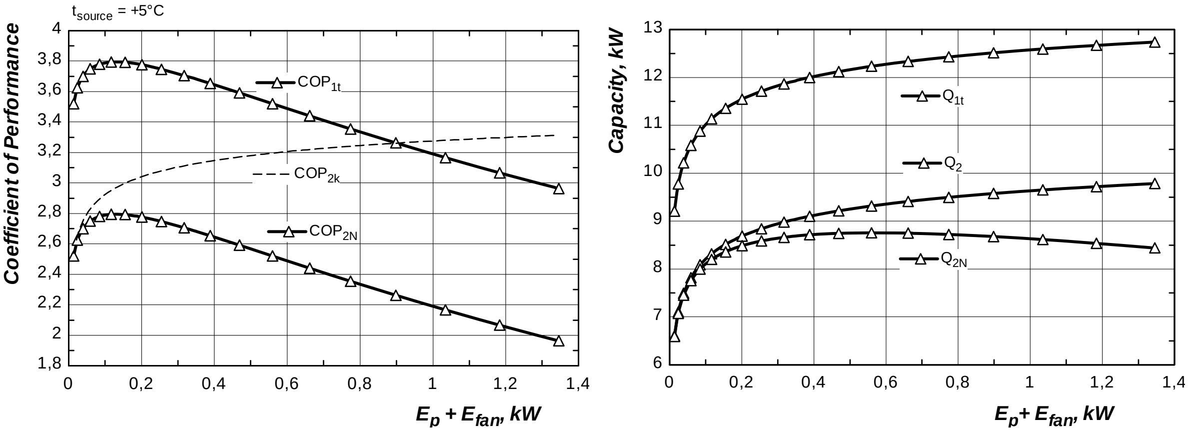


Figure 10.5.5. *Influence of Pumping Power Ratio on System COP and System capacity (same case as in Figures* 10.5.2-4)

A more general and dimensionless representation of is given in Figure 10.5.6 where the system performance is shown versus the “Pumping Power Ratio”, $(\dot{E}_p+\dot{E}_{fan})/\dot{Q}_2$. Curves are given for different source temperatures (+5, -5 and -15 °C). Also the pump and fan efficiency is varied: The efficiency is set to 20% except for one case (at t_{source} = +5°C) where 30% is used. For still another case with the same temperature (and with 20% pump and fan efficiency) the pressure drop in the brine circuit is assumed to be doubled compared to the other cases (to illustrate influence of long transfer lines). As is seen these changes have a certain influence on the capacity and the COP of the system, but the optima occurs at almost the same “Pumping Power Ratio”.

From the examples in Figures 10.5.4 – 6 it is seen that the optima are flat, especially concerning the capacity. To save energy the Pumping Power Ratio ($\Sigma\dot{E}_p/\dot{Q}_2$) should be chosen in the order of 1-2 % according to Figure 10.5.6, while 4 – 8 % would result in highest net refrigerating system capacity (for a compressor with fixed displacement). It is interesting to notice that the changes do influence the Q and COP but the optimum occurs at almost the same “*pump power ratio*” for all cases. Using a *pumping power ratio corresponding to max capacity*, *say 6% instead of 2%,* would give a *loss in net COP of almost 10%*, according to the information in Figure 10.4.6.

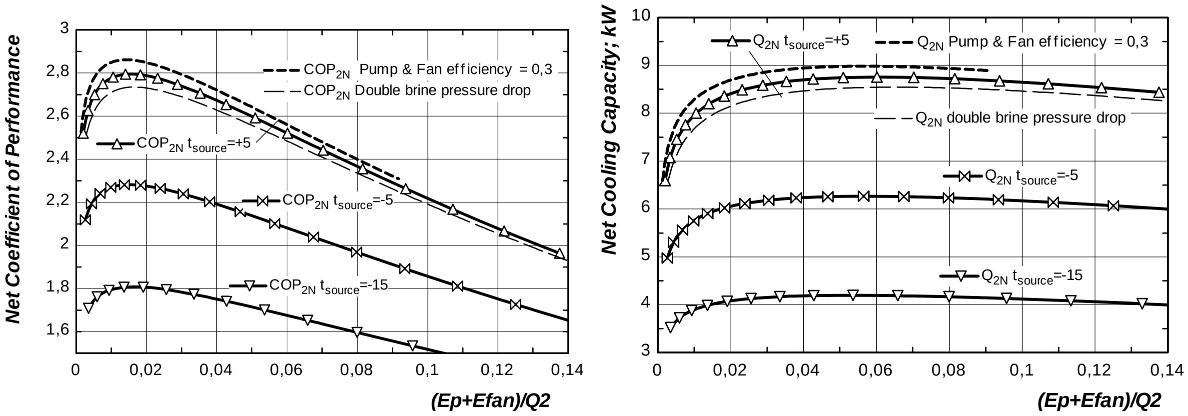


Figure 10.5.6. *Curves for three different source temperatures (+5, -5 and -15 °C). For the case of* t_{source} *= +5°C there are three curves to show the influence of different assumptions. Refrigerant R404A has been used in the model. Notice that the maximum net* COP_{2N} *occurs at roughly the same Pumping Power Ratio regardless of the source temperatures.*

In the examples shown it is assumed that there is a secondary circuit between an air cooler and the evaporator as shown in Figure 10.5.1. It is assumed that temperature change of the air and of the fluid are roughly equal which in this example gives about the same power demand for the air fan and the fluid pump (compare Figures 10.5.2 - 3).

One can also set the pump and fan power demand in relation to the c*ompressor operating power*: In Figure 10.5.6 we notice that at source temperature +5 °C the COP_{2t} is about 2,8 for $(\dot{E}_p+\dot{E}_{fan})/\dot{Q}_2$ slightly less than 2 %. This means that the ratio $(\dot{E}_p+\dot{E}_{fan})/\dot{E}_{compressor}$ for this optimum is about (2,8 · 2% =) *5 % of the compressor electric power*.

It will be shown that there are simple relations to estimate which pump power ratios are optimal for the different criteria.

Relations for optimal pump power

Data in the previous examples are relatively typical for many plants. It is possible to derive simple relations for optimal pump (or fan) power. The problem is quite analogous to the problem of choosing fan power for a direct expansion evaporator, which has been treated in Ref.s [1], and [2]. A further treatment for indirect systems is shown in Ref. [3] and Ref. [4].

Without going into detail in the treatment the following relations are given:

Maximum net cooling capacity is obtained if the pump power in relation to the cooling capacity (that is the ratio E_p/Q_2) is chosen:

$$\left(\frac{\sum \dot{E}_p}{\dot{Q}_2}\right)_{Q\max} = C_{Q\max}\cdot \theta_2 \qquad (5)$$

where

$C_{Q\max}$ has a value between 0.004 and 0.01 1/K, depending on type of heat exchanger compressor and operating conditions. (A more exact expression is shown in Ref [1])

θ_2 is the temperature difference (t_{source}– t_2) which directly influences the evaporator difference, t_2. (*To be correct it is the temperature difference that will occur at operation in the optimum conditions.)*

The best total net coefficient of performance COP_{2N} is achieved if the pumping power ratio is chosen:

$$\left(\frac{\sum \dot{E}_p}{\dot{Q}_2}\right)_{COP\,\max} = C_{COP\,\max}\cdot \theta_2 \qquad (6)$$

where C_{COPmax} is about 0.0007 á 0.0012 1/K, depending on type of heat exchanger compressor and operating conditions. (A more exact expression is shown in Ref [1])

θ_2 is the temperature difference as in eq. 5. Notice however as mentioned there, that one should use the temperature difference that occurs at the desired optimum. Since operation for the best *COP* corresponds to a lower pump and/or fan power the θ_2 to be used here is somewhat larger than for the case in eq.6. The two temperature differences are hence not exactly equal.

Similar relations can be deduced for the *best COP on the condenser side*. It can be expressed¨

$$\left(\frac{\sum \dot{E}_p}{\dot{Q}_1}\right)_{COP\,\max} = C_{COP\,\max 1}\cdot \theta_1 \qquad (7)$$

where $C_{COPmax1}$ has about the same value as C_{COPmax} in eq. 6.

θ_1 is temperature difference in the condenser, ($t_1 - t_{sink}$) at the optimum conditions.

Example:

We can use the example used for Figure 10.5.3-6 in order to test the relations. With a total temperature difference (compare Fig. 10.5.2) for maximum capacity $\theta_2 = t_o\text{-}t_2 \cong 5\text{- }(\text{-}6) = 11°C$ inserted in Equations 5 and 6 the result will be:

$(\Sigma\dot{E}_p/\dot{Q}_2)_{QNmax} \cong 11 \cdot 0{,}006 \cong 0{,}066$ or about 6 to 7 %

For maximum COP the temperature difference is somewhat larger (compare Fig. 10.5.2) and we may set $\theta_2 = t_o\text{-}t_2 \cong 5\text{-}(\text{-}9) = 14°C$. Eq. 6 gives:

$(\Sigma\dot{E}_p)/\dot{Q}_2)_{COPNmax} \cong 14 \cdot 0{,}001 \cong 0{,}014$ or about 1,4 %

These numbers are in the same order as shown in diagrams of Figure 10.5.6.

A conclusion which already has been observed is that the 'pump power ratio' for the evaporator side resulting in maximum net COP_{2N} is considerably lower than what for maximum net cooling capacity, $\dot{Q}_{2Nmax}$. The difference in power ratio is often factor 5 to 6. This means that one, in order to save energy should use roughly half the velocity compared to what would give maximum capacity. When comparing, it is however important to check that the lower velocity (equivalent to max COP) does not correspond to laminar flow. If the heat exchanger is not designed for this kind of operation, it can mean that the heat transfer coefficients are drastically lower and the temperature differences in equations (5) to (7) increase.

For a heat pump, it is natural to operate for max COP_{1t} as long as the capacity covers the demand. –*However,* for ambient temperatures below the balance point temperature (when supplemental heat would be necessary) it is suitable to use a higher speed of the pump and fans on the cold side so that the capacity is increased. This means that the balance point temperature can be lowered by a couple of degrees and that the number of hours is decreased when supplementary heating is required.

A practical solution is of course to choose power in the pumps and fan *between* the two optima, resulting in a moderate loss of COP_{2N} compared to the optimum but also a moderate loss of capacity. If we in the example that has been illustrated had chosen $(\Sigma(\dot{E}_{p2})/\dot{Q}_2) = 3\%$ the loss in net *COP* as well as in net cooling capacity would be in the order of a couple of percent only of the values for COP_{2N} and $\dot{Q}_{2N}$.

Discussion and conclusions

From the relations for optimum "Pumping Power Ratio" it is seen that the power spent for pumping in relation to the heat exchange should be proportional to the temperature difference θ in the heat exchanger. Hence, large temperature differences justifies larger pumping power ratio. This is natural if we consider that the potential benefit (increase in t_2 or decrease in t_1) by increasing the flow velocity is proportional to the temperature difference that is influenced by the flow rate.

Another perhaps amazing conclusion is that *pump (or fan) efficiency* has only a small influence on the power ratio to be spent for optimum conditions (although the temperature difference will be somewhat larger and thus have a certain influence). Spending the same power in pumps or fans, lower efficiencies means that the flow rates will be lower, and the performance of the system will of course be poorer. Additional pressure drops in transfer lines have similar influence; they do not affect the optimal pumping power ratio noticeably, but will negatively affect the system performance.

To save energy in *heat pump applications* the results indicate that roughly the same pumping power ratio for max *COP* as for refrigeration are applicable, see Fig.3. In situations where the heat pump covers the full heating demand, max COP is the mode of operation that should be chosen in order to save energy. However, for other operating conditions below the 'balance point' where supplemental heating would be required, it is be beneficial to use higher flow rates (larger pumping power ratio) as a way to increase the capacity of the system in order to reduce the demand for supplemental heating – the max Q_{2N} mode.

For operation of a system in practice, there are today efficient *speed controls* that can be used for the pumps and fans to save energy. Provided that there are no restrictions on air temperature change in the cooler it seems logical to let the speed of the fan be adjusted in the same fashion as the pump (by same frequency control unit). Relations for the Pumping Power Ratio for optimal system performance, as shown, may be incorporated in control algorithms for efficient system operation.

From the examples shown in the diagrams, it is seen that the curves are relatively flat around the optimum. For practical use, it is not necessary to be very accurate; we do not need to use all the digits. It is definitely important to keep an eye on the auxiliary power spent in pumps and fans especially on the evaporator side of refrigerating systems.

For the overall operation of systems, the concept of *Pumping Power Ratio* is convenient to check. The impression is that many systems in practice use pumping power ratios more corresponding to system maximum capacity, rather than saving energy even in situations where there is an excess capacity of the system. In practice, it not uncommon that pumps and fans consume 20% of the compressor (equivalent to 5 – 10% of the cooling capacity) and it may be advisable to try a pump of fan with lower velocity in order to save energy!

Nomenclature

$COP_2 = \dot{Q}_2/\dot{E}_k$	the COP of the compressor (COP_2)	--
COP_{2N}	Net cooling COP_2 for the system, eq. (2)	--
COP_{1t}	Total COP_1 of system, equation (3)	--
E_k	Compressor power demand,	kW
$E_{fläkt}$	Power demand of fan (on cold side)	kW
E_p	Power demand of fluid pump (on cold side)	kW
E_{p1}	Power demand of pump (on warm side)	kW
Q_2	Cooling capacity of the system	kW
Q_{2N}	Net cooling capacity of the system, eq. (1)	kW
Q_1	Condenser capacity ($=Q_2+E_k$)	kW
t_{luft}	Temperature of the heat source (on the cold side)	°C
θ_1	Temperature difference on the warm side	K
θ_2	Temperature difference on the cold side	K
C_{Qmax}	factor, defined in eq. 5	K^{-1}
C_{COPmax}	factor, defined in eqs. 6 - 7	K^{-1}

References

[1] Granryd E., 1973: *”Inverkan av fläkteffekt vid förångare och kondensor på en kylanläggnings kyleffekt och totala energibehov” (Influence of fan power in evaporators and condensers on the total capacity and energy requirements of a refrigerating plant)*, Scandinavian Refrigeration, No 4, p 124-132, 1973.

[2] Granryd E., 1998: *“Power for fans and pumps in heat exchangers of refrigerating plants”* Proceedings of the 1998 IIR International Refrigeration Conference, Purdue University, West Lafayette, Indiana, USA.

[3] Granryd E., 2002: *“Optimal circulation rates in indirect systems”*, Proceedings of the IIR International Conference on Minimum Charge Refrigerating Systems, Stockholm, Sweden,

[4] Granryd E., 2007: *“Optimal flow rates in indirect systems”*, Proceedings of the 22nd IIR International Congress of Refrigeration, Beijing.

Section 10.6 (J-E Nowacki)

A few special problems

Background

There are a few common optimization problems which can be treated in a simple way and which can supply us 'rule of thumb' solutions. Some are simplified by assuming continuous functions – an example is optimal thickness of insulation – while others are of a more 'discrete' nature for instance how to arrange different components in an efficient way. A few examples will be treated here.

Relations for energy optimal thickness of insulation

Equation 11.40 in the book Kylteknikern by Bäckström [1] treats how to estimate economic thickness of insulation.

If we simplify the relations somewhat, use SI system units and treat a plane wall the equation by Bäckström can be written:

$$s_{opt} = \sqrt{\frac{\Delta T * \lambda * \tau * c_{energi}}{c_{vol} * a}} \qquad (1)$$

In this equation, the following denotations are used:

s_{opt}	optimal thickness of insulation, m
ΔT	temperature difference over the wall, K
λ	thermal conductivity of the wall material, W/mK
τ	annual time period during which the temperature difference is maintained over the wall, s
C_{energi}	price of the energy transferred across the wall, SEK/J (low temperature thermal energy, produced by for instance a heat pump, might be cheaper)
C_{vol}	cost per volume of the insulation material, including installation and also including the cost of volume occupied by the insulation, kr/m³
a	annuity of capital investment, see Section 1 equation 8, 1/year

Example: Wall of a house:

ΔT = 20K, λ=0.04 W/mK, τ=9 000 000 s (2500 h/year), $C_{energi}=0.333*10^{-6}$ SEK/J (1,2 SEK/kWh), C_{vol}=10000 SEK/m³ (building which costs 25000 SEK/m² with 2,5 m roof height), a = 9,5% as in Section 1.

The optimal thickness of insulation becomes:

$$s_{opt} = \sqrt{\frac{20*0{,}04*9*10^{6}*0.333*10^{-6}}{10000*0{,}095}} = 0{,}05m$$

The reason for this amazingly thin optimal wall insulation is that the cost of the building volume itself has been included and that it is so high (C_{vol}=10000 SEK/m³). If instead only the cost of the insulation material is considered then a more realistic value would be C_{vol}=600 SEK/m³ which would increase the optimal thickness of insulation to more than 20 cm – which is more familiar to us in the houses we know.

Still another example of interest could be to check the insulation thickness of the pipe for a cold secondary fluid in a refrigerating system. We assume that the insulation can be considered 'thin' in relation to the pipe diameter. Let us use the following data: ΔT=20°C, λ=0,04 W/mK, t=9 000 000 s (2500 h/year), C_{energi}=0,1 $*10^{-6}$ SEK/J (equivalent to 0,36 SEK/kWh cooling – as a consequence of electricity price 1,2 SEK/kWh$_{el}$ COP_2 = 3.3), C_{vol}=30000 SEK/m³ (tube material with low diffusivity), a=9,5% as in section 1. The result is:

$$s_{opt} = \sqrt{\frac{20*0{,}04*9*10^{6}*0.1*10^{-6}}{30000*0{,}095}} = 0{,}016m$$

This result is probably double as thick as what is commonly used today. (But in this example no 'price of volume' is included for the volume that the insulation occupies.)

Relations for optimal heat exchanger area - exergy

No energy is lost when heat is transferred through an area. It is quite obviously the same number of J that goes into the wall as comes out of the wall on the other side. However there is a temperature difference and to get a measure of the loss associated we can consider the idea to use this temperature difference to operate an ideal little steam engine. The power that ideally could be extracted is a measure of the loss due to the heat transfer resistance of the wall (the *exergy* loss). This thermal resistance (on both sides of the wall as well as the resistance through the wall) results in practice in an increase of the compressor energy to operate the refrigeration plant or the heat pump and ideally this is the same as the power we could have generated by the ideal steam engine as described. By making the wall bigger, the heat flux becomes smaller and the temperature difference is reduced. Thus by investing in a larger wall area, we can reduce the energy loss…

If the temperature difference over the wall is small compared to the absolute temperature and significantly above the ambient- the following relation describes the economically optimal temperature difference over the wall:

$$\Delta T_{opt} = \sqrt{\frac{a*c_{kA}*T_1^2}{\tau*c_{exergi}*T_0}} \tag{2}$$

The following symbols are used in this equation:

ΔT economic temperature difference across the wall, K
a annuity, 1/year

c_{kA}	the price to increase the area of heat exchange, marginal price, SEK/(W/K)
T_1	the temperature of the warm side of the wall – or the warm side fluid, K
τ	annual time of use of the heat exchanger, s/year
c_{exergi}	cost of energy, SEK/J (equivalent to the price of electricity)
T_0	ambient temperature in K

Example: Radiator for room heating:

τ=9 000 000 s (2500h/år), C_{exergi}=0.333*10^{-6} SEK/J (1,2 SEK/kWh), a=9,5% according to part 1, c_{kA}=40 SEK/(W/K), T_1=313 K (40°C) an iteration may sometime be required, T_0 = 273 K (0°C). Notice that the flow rates on both sides of a radiator wall varies. If this is to be taken into account the formulas becomes more complicated. We have here used average temperatures. The result becomes:

$$\Delta T_{opt} = \sqrt{\frac{0.095 * 40 * 313^2}{9 * 10^6 * 0{,}333 * 10^{-6} * 273}} = 21{,}3\ K$$

This value is not far from what is normal for radiators at ambient temperatures around 0°C. Room temperature will be 40-21,3 = 18,7 °C….

When very close to the ambient temperature or below the ambient temperature the optimal temperature difference can be roughly calculated as:

$$\Delta T_{opt} = \sqrt{\frac{c_{kA} \cdot a \cdot T_0}{c_{Ex} \cdot \tau}} \qquad (3)$$

There are also optimization formulas specifically aiming for optimization of evaporators in heat pumps, air conditioning and refrigeration Granryd, ref [2] gives e.g. the formula:

$$\Delta T_{opt} = \sqrt{\frac{c_{kA} \cdot a}{c_{Ex} \cdot \tau} \cdot \eta_{Ct} \cdot \frac{T_e^2}{T_c}} \qquad (4)$$

where in this formula above:

η_{Ct}	denotes the Carnot to total efficiency,
T_e	is the evaporation temperature and
T_c	is the condensation temperature

Losses when fluids of different temperatures are mixed

When fluids of different temperatures are mixed, there is always a loss in exergy – or in available work. This lost work will have to be spent at some place of the system. If an air stream is heated to 40°C and it is mixed by an equal amount of ambient air of 20°C the result is air at 30°C (but the mass is dubbled). A heat pump would require more work to heat the first air stream to 40°C from 20°C than to heat both streams to from 20 to 30°C. Whenever it is possible, one should avoid mixing fluids of different temperatures.

… still sometimes we *have to* mix…

Sometimes it is difficult or impossible to adjust components so that all functions optimally without mixing. For instance, a ground source loop in a heat pump system may require a larger flow rate than what can be handled by the evaporator of the chosen heat pump without excessive pressure losses. Schemes as shown in the appendix from Tommy Nilsson can sometimes be applicable and suitable.

Do not throttle away pumping power

In large plants, it is especially important to avoid capacity control by throttling valves or shunt valves. The capital cost for a pump solution is often small compared to the additional operational costs when throttling. Speed controlled pumps is a very good alternative to adjust flow rates.

Don’t use more antifreeze substance than needed

Since water has very good heat transfer properties one should use as little antifreeze additives as possible. Too often, the freezing point is chosen to avoid problems in the worst extreme case. Instead it would be more efficient to aim for a mixture which just (barely) will be enough for the worst case. This will make the operation more effective in normal situations – but it is a difficult issue of judgment.

The choice of pumps

Some circulation pumps have their highest efficiency at high flow rate and a low-pressure head, others at low flow and high head. If a solution with several parallel circuits with a low-pressure drop have been chosen, instead of one single long circuit, a pump suited for high flow rates with small pressure head is better. It can sometimes be motivated to change an existing pump after analyzing pressure drop and flow. In addition, in this case the use of speed controlled pumps result in easier adjustment.

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Attachment:

Comments about Ground source heat pumps (Tommy Nilsson)

Ground source heat pumps are used in an increasing portion of the buildings in the country.

It is important that the heat extraction in the bore holes in the ground is as effective as possible and that a minimum of pump energy is spent in relation to the ‘harvested’ energy

Energy can be saved if the pipes are designed so that the pressure drop is small, and the pump is dimensioned permitting turbulent flow even at conditions with the lowest temperature of the secondary fluid of the year.

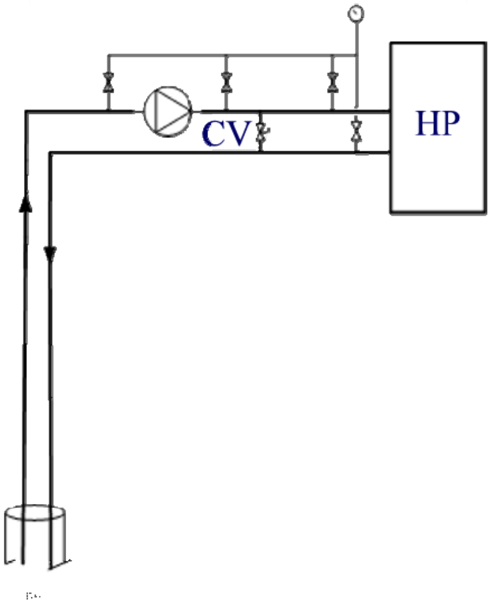
It is advisable to do a careful estimation of the pressure drop in the pipes also including the evaporator. This should be done before e.g. drilling a borehole heat exchanger. Dimension the depth so that it does not consume excessive electric power when operating over the years. Sometimes it is preferable to use two shorter holes instead of one deep. This will increase the flow rate but reduce the pressure drop; less power will be consumed though.

Choosing 20% ethyl alcohol (by volume) instead of 30% will result in a lower flow rate and pressure drop while maintaining turbulent flow in the pipes. Thus, the required pump work is reduced and the energy efficiency increases.

Scheme of a plant with one bore hole and a heat pump.

First check flow rate and corresponding pressure drop in the evaporator required by the manufacturer of the heat pump. Then check if this flow rate is enough to maintain turbulent flow in the ground source pipes and if the intended pump will be suitable to handle this flow.

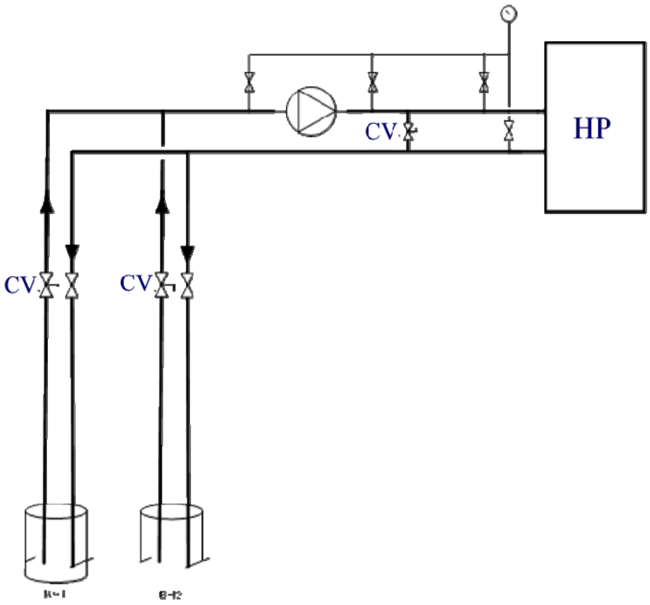
For the case when the borehole pipes requires larger flow rate than what the pump will deliver then chose a different pump and make sure *not to forget the pressure drop in the evaporator*. A solution may possibly be to install a short cut piping as shown in Circuit A to ensure that the bore hole get a flow rate high enough to avoid luminary flow. This however is not a very efficient solution, compare the discussion in thee previous section.



Circuit A - HP is Heat Pump and CV is Control Valve

<u>Plant with several bore holes and a single heat pump.</u>

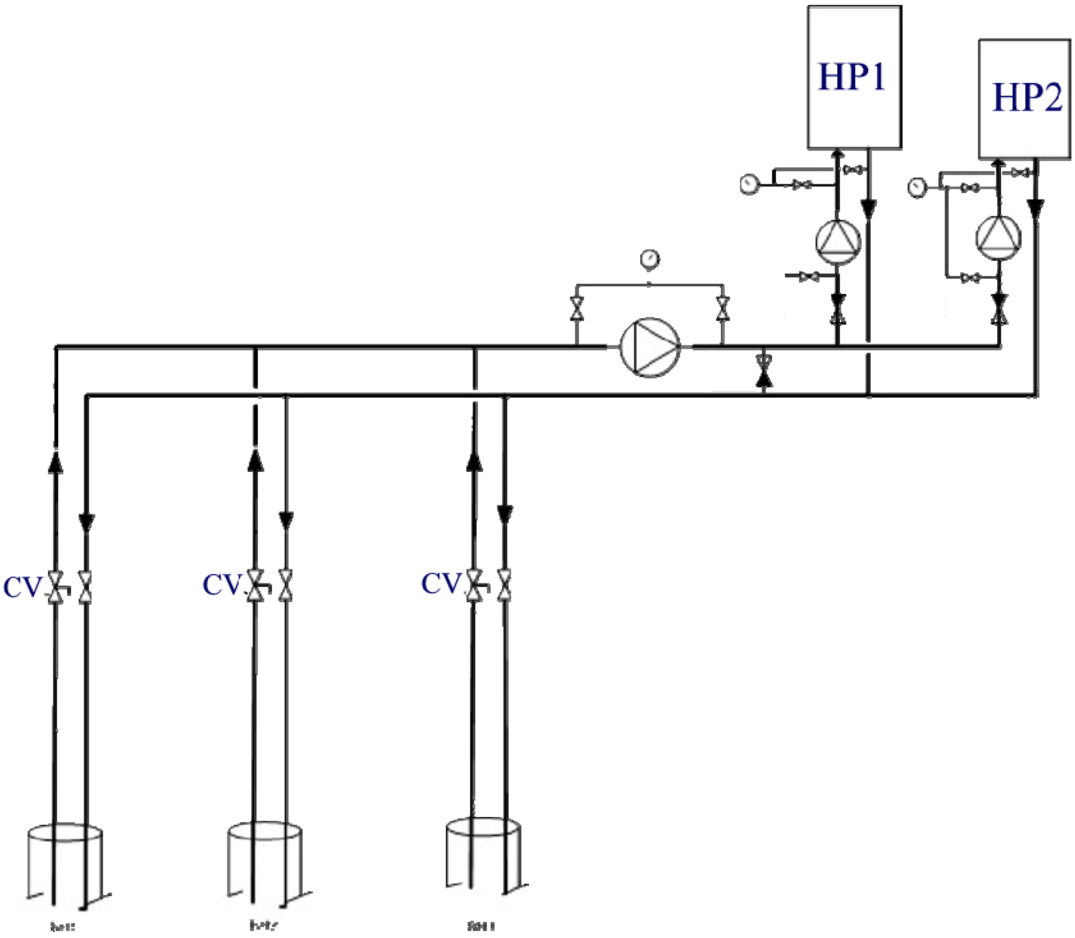
Estimate the minimum flow rate for one hole to maintain turbulent flow for the case with the lowest expected temperature of secondary fluid. The flow rate for the pump and evaporator is that flow rate multiplied by the number of parallel holes. In many cases, the estimation of the pressure drop in the evaporator then becomes excessive. A separate circuit for the boreholes may be required. (Same discussion as for the previous case). See <u>Circuit B</u>.



Plant with multiple boreholes and several heat pumps units (or one large unit)

Plant with several heat pumps and boreholes

In this case, each heat pump is often already equipped with a separate pump and there could also be a common (larger) pump, see Circuit C, for the secondary fluid to the multiple boreholes. This common pump however is (mostly) in operation only when all heat pumps are in operation. The standard pumps in the heat pump units can for this case often be substituted by smaller pumps since the pressure drop in the circuit with multiple boreholes will be smaller than for a case with only one hole. The low heat transfer coefficient if laminar flow occurs when only few heat pumps are working is compensated by the large area available.



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