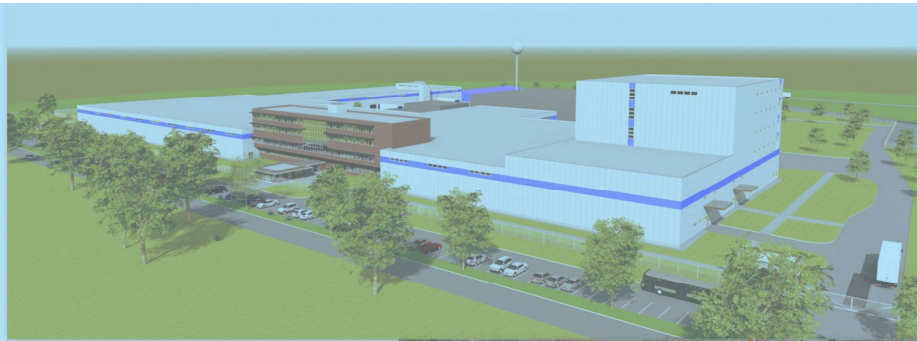


Kemény György: Worksheets for refrigeration equipment designing

Foreword:



About me in a few short words: I'm in my 72nd year now and have been solving various refrigeration engineering problems for the past 53 years. I began at 18 as a technician in a large design bureau, more precisely in its refrigerating technology department. I first became assistant designer here, before being promoted to head of design. After spending 20 years here, I moved on to a company manufacturing refrigerating equipment as head of design, then continued there as chief engineer and finally I became its director. This lasted nine years. Then I became technical director for a general contractor company specialised on building and installing cold

storage facilities. In 2001, I established my own design company, specialised on industrial refrigeration and HVAC equipment.

What led to the publication under the present title is my personal experience, namely that the large design bureaux disappeared from the market within a few short years after 1989, (in Hungary), and young graduates exiting the universities found it very difficult to acquire professional designing experience (on refrigeration). Small companies can hardly afford financing a trainee's learning period, or risk damages that may result from possible design flaws, and students at universities also do not receive enough theoretical education for planning as a process.

Nevertheless, it is my solid view that one of the most important and most beautiful activities in building a nation is engineering design. Naturally, this must be carried out professionally at the highest possible level of quality, continuously studying and applying new information, while keeping deadlines relatively short. The worksheets summarising my professional experience may provide assistance in this, as they speed up the manual part of the design work, allowing more time to thoroughly reconsider the task and to engage in advanced professional training. I hope these will help the next generation of designers and that there will be young engineers to continue the development of this work and the Hungarian refrigeration system engineering culture will not be rendered a thing of the past.

I must therefore stress that this is by no means a textbook, but a design assistance tool for engineers for their design work!

Dated: 22.11.2015

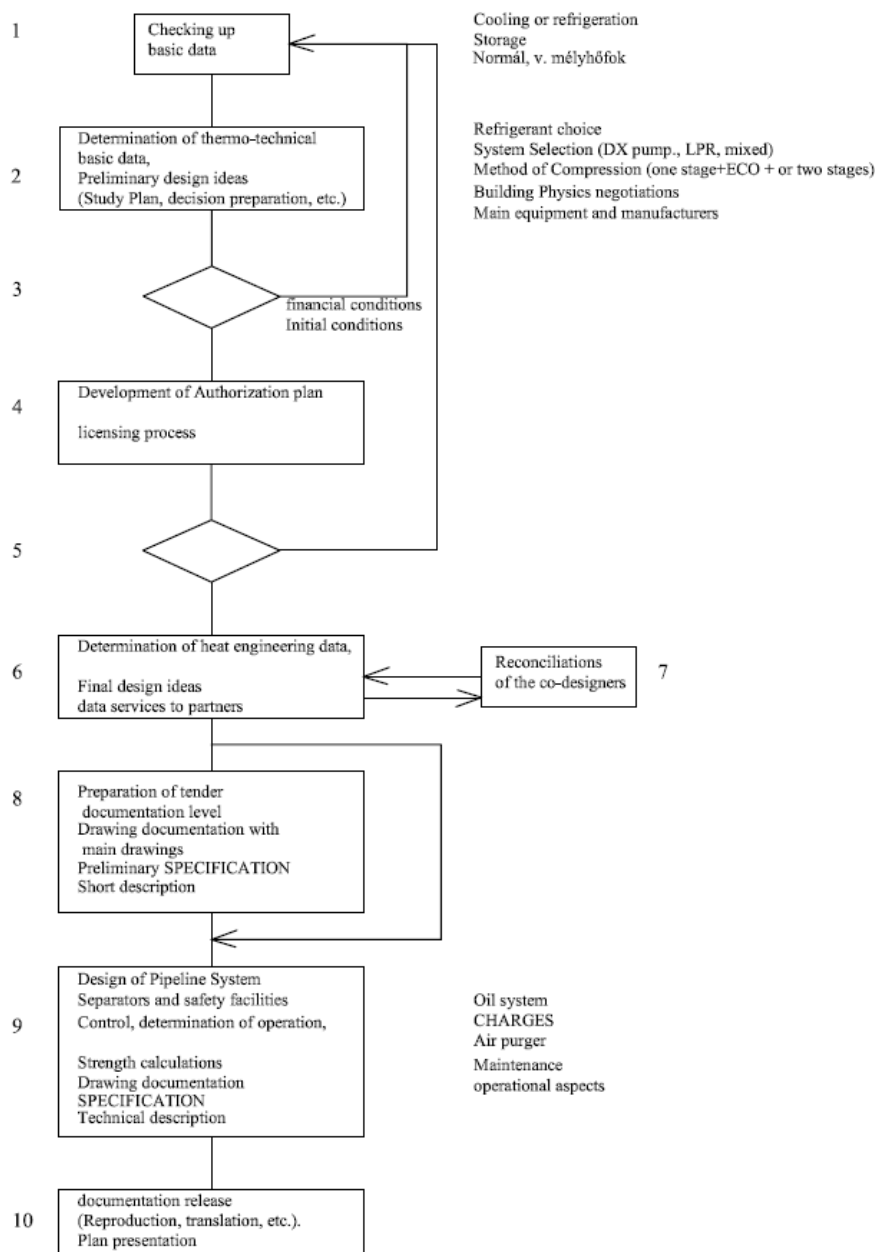


György Kemény

((In the front page some facility can be seen what designed by me, and can be seen some famous people who are role models of all engineers and somewhere its predecessors)).

The planning process:

I summarised the design process in 10 steps: It naturally corresponds to designing refrigeration systems, but it is a generally analogous process for any designing activity, be it some technology, architecture, etc. Large developments, where the entire process is implemented in this manner, are currently rare in Hungary. Generally, the task stops in the so-called “tendering” phase (point 8), after which the contractor is selected to implement the project. The preparation of the further planning documents (technical design) is usually the task of the general contractor. Minor facilities often disregard this, which, as a practice may prove disadvantageous later on for a planned maintenance test or for solving any other problem



I will try to order the following chapters so as to follow the flowchart.

Point 1 is the clarification of the initial task, the most important question to be reconciled with the customer. Inadequate preparation of this can be the source of countless disputes and damage. The reason I state this with such certitude is because I have also worked as an expert technical witness and have had to support the verdict in many such disputes. It is neither a coincidence that I have indicated a feedback to this point at points 3 and 5, as the customer's requirements may change based on the cost estimate prepared according to the preliminary plan, or the provisions of the licensing authority. The planning documentation must **contain these reconciled data by all means** in order to corroborate compliance of the designed equipment!

Point 2

- a) Selection of the system and the type of refrigerant greatly depends on the magnitude of the project. After the building physics reconciliations (materials and dimensions of boundary surfaces, building structures, screening, etc.) and based on the initial task, a cooling requirement calculation needs to be performed. **Worksheet 01, titled “cooling requirement”** provides assistance in this. This is a Microsoft Excel based spreadsheet with the necessary Visual Basic and macro supplements, basically tailored to suit the needs of the refrigeration requirements of the alimentary industry.

Naturally, it is also applicable to other materials to be refrigerated, but in this case the refrigerating requirement of the material must be determined separately, and the refrigeration of the goods and the breathing heat data must be amended according to worksheet 1. Options are also available to temporarily overwrite the specific heat capacity and the latent heat of solidification data in the “Product” column after selecting the product, allowing the software to perform the calculation. The first page of the worksheet has been formatted for data entry. Here, one can select the product to be refrigerated, the dimensions of the refrigerated space, the data of the boundary surface, etc.

By pressing the buttons indicating the various boundary surfaces, different wall structures can be selected. Selecting the “Other” button here, the thermal conductivity of a specific material can be given. Make sure to use a decimal point for separating the data, otherwise you will receive an error signal.

The programme is not able to calculate a composite order of layers. In such cases, the equivalent thermal conductivity of the composite order of layers and the corresponding layer thickness must be determined by separate calculation, using the correlation below:

$(\sum \partial/\lambda) \cdot 0.023 = D$, where D is the thickness of the polyurethane insulation equivalent to the composite order of layers (m), and $(\sum \partial/\lambda)$ is the sum of the quotients of the thicknesses and thermal conductivities of the individual layers ($\text{m}^2 \cdot \text{K}/\text{W}$).

If “freezing” is selected as the refrigeration task, an auxiliary window appears for determining the expected cooling time. The worksheet uses “Newton’s” and “Plank’s” formulae here, primarily used for refrigerating meat products. The formulae can be found in “Húsipari Kézikönyv” (*The Meat Industry Handbook*) (Mezőgazdasági Kiadó /Agricultural Press/, 1973). The calculation of the cooling time must be handled with caution, as cooling is influenced beside air velocity and meat thickness by a good many other factors, from feeding to the method of slaughtering an cutting. The data provided must therefore be used as a guideline and be supplemented with data gained from experience.

As for the product flow, a precise volume may be given in kg/day, or a percentage for the daily movement. For the percentage value, the software calculates from the volume of the chamber and the stocking density of the product. Stocking density here must be verified to comply with the actual method of storage, or the actual volume must be used for calculation.

After data entry, by clicking on the „Calculate/recalculate”, we obtain the cooling requirement in detailed and consolidated form, converted to duration of operation. As the selection of the air coolers here has not yet taken place, the value calculated in advance for ventilation must be checked back after selection.

- b) Knowing the refrigeration capacity requirement(s), we then determine the refrigerating system and refrigerant to be applied. It is not possible to attribute a precise algorithm to these decisions. As guidance, it can be said that freon refrigerants offer an economical solution for small plants, below a magnitude of about 500 kW.

A small plant in this context may be a meat processing plant with about 5~12 refrigerated spaces or a fruit and vegetable storage/processing facility. For transit warehouses and supermarkets, carbon dioxide may be a more advantageous refrigerant of choice.

Above 500 kW, the use of ammonia is generally the advantageous choice. The energy requirement may help decision as a point of consideration, if the electricity requirement is calculated for the different media. Here, the electrical power consumption of the refrigerating compressors is a definitive factor, but for a full calculation, the other loads (air cooler fans, condenser fans, ammonia and other pumps, etc.) must also be taken into account. **Worksheet no 12** found under the filename “**worksheets_1_18**” assists in this. The cells with yellow background must be filled in the worksheet, and the electrical energy requirement is then calculated by the software for the various systems. A few sample rows are filled; these should naturally be deleted or can be overwritten. Otherwise the cells for data entry being yellow applies to all worksheets!

A further consideration is the licensing procedure depending on the refrigerant used. For freon media, the involvement of the environmental authorities is by all means an obligatory point here, as are regular leakage checks and the reporting of charge replacements that must by all means be undertaken by the operator.

In case of using ammonia as a refrigerant, commissioning of the equipment must be licensed by the “Regional Technical Safety Authority”. The licensing also requires the standpoint of the disaster relief forces. For a system using more than 1000 kg of refrigerant, a disaster relief plan may be required.

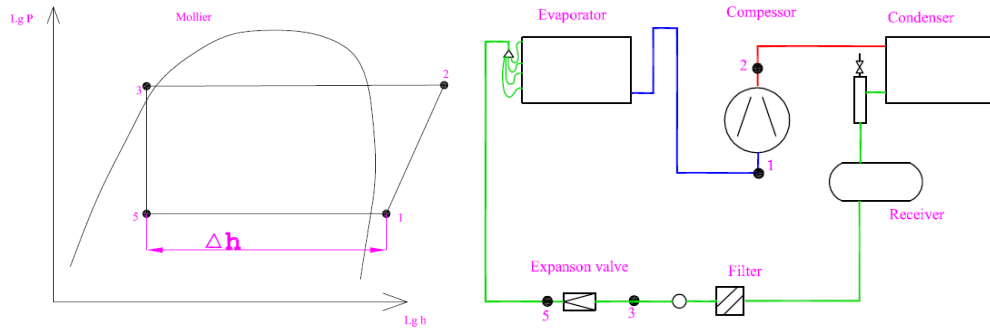
Worksheet no 11 assists in preparing the licensing documentation found under the filename “worksheets_1_18”.

The cells with yellow background must be filled in the worksheet, and the classification is then carried out by the software according to the referenced standard. A few sample rows are filled; these should naturally be deleted or can be overwritten. The completed table may be exported and used in the licensing procedure of the authorities.

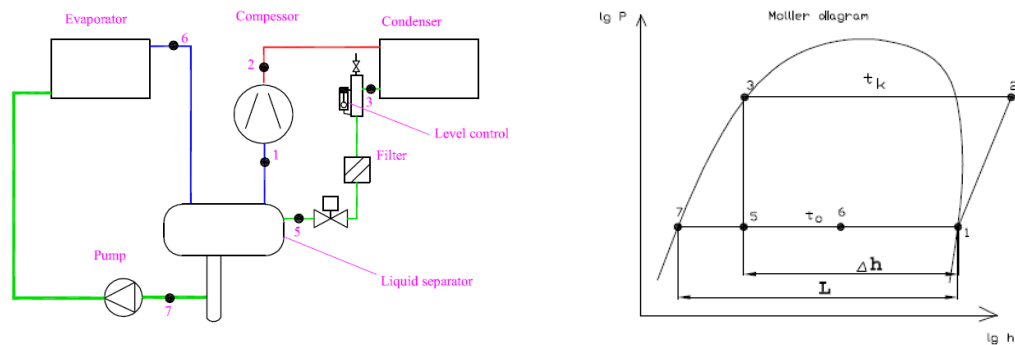
- c) In terms of its structure, it is possible to create a system with direct expansion (DX) regardless of the refrigerant, or a pump or the LPR system itself. (The definition concerns the method of fluid supply of the air coolers.) Generally, the DX system is used for small size freon plants. Pump systems are used for larger, mostly ammonia-

charged and carbon dioxide systems. The LPR system is somewhere between these two and is rarely applied in spite of its many advantages, therefore I will include a more detailed description about it.

Theoretical scheme and flowchart at a DX system:



Theoretical scheme and flow chart of the heat pump system:



An important point of pump systems is determining the dimensions of the separator. The cross section of the vapour chamber can be dimensioned based on the volumetric flow rate of the vapour. The safe value of the permissible speed is ~ 0.3 m/s. For a horizontal arrangement, this increases according to the L/H ratio, where L is the path of the vapour from inlet to discharge and H is the height of the vapour space up to the highest fluid surface.

The cross section and the volume of the fluid chamber depends on the fluid charge of the evaporator and return suction pipes. **Worksheet no 1** found under the filename “**worksheets_1_18**” assists in determining this and the calculation of the volume of the horizontal tank. The cells with yellow background must be filled in the worksheet, and the calculation is then performed by the software. A few sample rows are filled; these should naturally be deleted or can be overwritten.

The evaporator fluid content may be determined using the Slipcevic-formula:

$\Delta V = V_{\text{elpár}} * (1 - 1/(1.2 * n^{0.2}))$, where “n” is the so-called evaporation number.

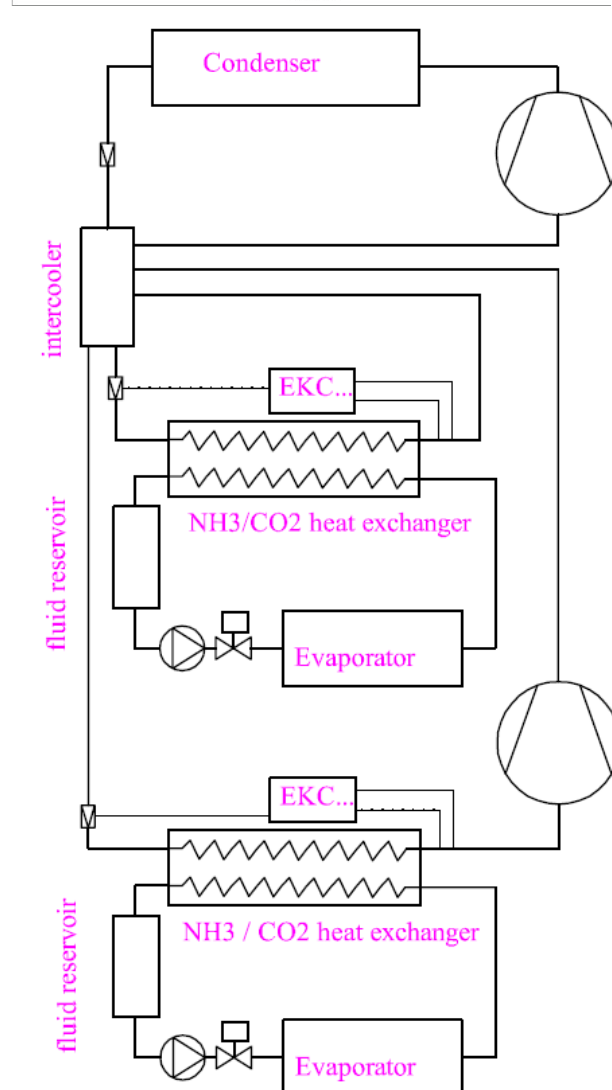
Its usual value for ammonia systems is between 3~5, and for CO₂ systems it is between 1.5~2.5.

The fluid content of the return line may be determined using the Slipcevic-formula:

$\Delta V = V_{\text{cső}} * (1 - 1/n^{0.2})$.

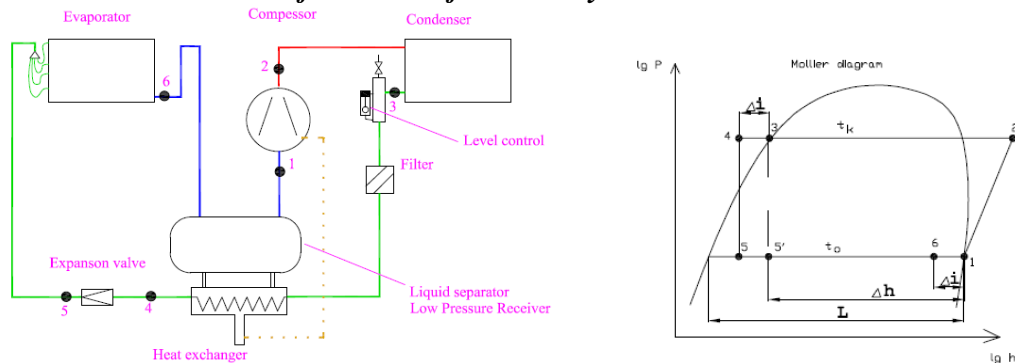
The fluid content of the inlet line is equal to its volume, no charge change takes place here, but the separator tank’s size must allow for possible drainage.

Change of state of CO₂:



Although, based on their derivation, the Slipcevic-formulae mentioned above are valid regardless of the refrigerant used, there have been experiences reported, concerning carbon dioxide as the refrigerant, of the fluid charge being 15~20 % greater. This is caused by the greater density of carbon dioxide and the ratio of the fluid and vapour phase densities being different, resulting in a different flow profile. The flow profile may be viewed with the help of **Worksheet 02 titled “FlowMap”**. Based on the medium data, the worksheet calculates the Froude numbers characteristic of the flow profile and indicates the resulting flow profile in horizontal and vertical pipes. In a horizontal pipe, for example, ammonia largely tends to flow with a layered undulated flow profile, while carbon dioxide flows with a ring profile. **Worksheet no 02.** is the work of Shell DEP 31.22.05.11, available on the internet: (http://excelcalculations.blogspot.hu/2012_02_01_archive.html)

Theoretical scheme and flow chart of the LPR system:



As it can be seen, this system is fitted with an appropriately dimensioned separator before the compressor and the high pressure fluid is powerfully aftercooled with the help of a heat exchanger. The heat exchanger is connected to the separator on the low pressure side. An important feature of the connection is that no fluid is stored on the condenser side, all fluid is stored in the separator. On the condenser side, only a single fluid lock, i.e. a single fluid level regulation is applied. As a consequence of the above, feeding takes place by means of a simple (not thermostatic) regulator valve. In order to follow the operation of the connection, the difference of enthalpy indicated in the graph as Δh must be determined. This can be determined based on the operational conditions of the heat exchanger, using the correlation below:

$$\Delta h = c_p \cdot (t_k - t_o) \cdot (1 - e^{-k \cdot F / c_p \cdot M})$$

where:

c_p	is the specific heat capacity of the refrigerant in liquid state	$\text{kJ/kg } ^\circ\text{C}$
t_k	is the condensation temperature	$^\circ\text{C}$
t_o	is the evaporation temperature	$^\circ\text{C}$
k	is the convective heat transfer coefficient	$\text{kW/m}^2 \text{ } ^\circ\text{C}$
F	is the surface of the heat exchanger	m^2
M	is the mass flow rate	kg/sec

The mass flow rate is $M = Q_o / \Delta h$

Where:

Q_o	is the refrigeration capacity	kW
Δh	is the difference of enthalpy at the evaporator	kJ/kg

Is the capacity of the heat exchanger $Q_h = \Delta i \cdot M$

The quantity of refrigerant evaporating on the low pressure side of the heat exchanger = Q_h/L , where L is the latent heat of evaporation of the refrigerant (kJ/kg).

The specific vapor content at the exit of the air cooler, i.e. point 6 of the diagram = $(1 - \Delta i/L)$

The heat exchanger seemingly implements a so-called internal heat exchange, meaning the difference of enthalpy between points 3 and 4, and 1 and 6 is identical. The overfill of the evaporator is equal to the multiplicative inverse of the vapour content measured at the exit point.

(The notion of overfilling is analogous to the circulation number used in pump systems)

For selecting the heat exchanger, care must be taken not to have a too low vapour content after the feed valve, as this may cause fluid distribution problems. This problem, more accentuated (than generally) in deep-freezing systems implemented using screw compressors when the so-called economiser is used, and the fluid temperature after the eco is too close to the evaporation temperature.

The evaporator fluid content may be determined by using the Slipcevic-formula again: $\Delta V = V_{\text{el pár}} \cdot (1 - 1/(1.2 \cdot n^{0.2}))$, where “n” is the aforementioned overfill.

Achievable overfill for ammonia is around 10~15%, i.e. $n = 1.1 \sim 1.15$, while for freon media it can be significantly higher, with values up to $n = 1.5 \sim 2$.

The role of overfilling is:

1. increasing the efficiency of air coolers,
2. “tracking” the load changes and
3. regulating the complex systems.

For the proper operation of an air cooler, it is sufficient if the exit takes place at this saturation point (i.e. $n = 1$).

The part above $n = 1$ of the mass flow rate actually created in the cycle serves as redundancy to “buffer” regulation inaccuracies, if there are multiple evaporators in the system connected in parallel. For freon media, the greatest achievable overfill therefore allows easier regulation.

From the above, it is obvious that the LPR also requires regulation of the condenser pressure, but there is no need for the so-called pump down procedure - this would even be disadvantageous upon restart.

With reference to the advantages mentioned in the introduction, I interpret them as follows:

1. The lower risks of lower refrigerant loads and the loads stored at low pressure primarily emerge in case of ammonia pump systems, sustaining the advantage of pump systems that the separator in this case may be installed without restrictions of location.

2. The safety of the compressor against fluid hammer is an advantage for all refrigerants.
3. Operating the evaporator with smaller temperature differences is advantageous in situations where the weight loss or desiccation of the product is to be avoided.
4. A simple defrosting option by reversing the cycle is primarily advantageous for small size systems with few evaporators, especially in air condenser applications.
5. The more efficient possibility of oil collection and return is an advantage with ammonia systems.
6. The low implementation cost of the system also represents an economic benefit, naturally depending on its magnitude and the refrigerant used.

The LPR was first introduced by S. F. Pearson in England at the Institute of Refrigeration Technology and at the Institute of Marine Engineering in 1996.

- d) Based on the required room temperature, it can be easily decided whether to employ single stage or double stage compression. Here, with the refrigerant known, it must also be decided whether to serve deep-frozen areas with two stage compression or a single stage economiser system. For smaller freon plants, it is easier to use a two stage compressor with intermediate cooling. For larger plant sizes, single stage economiser systems may be more advantageous, especially with a closed eco heat exchanger, as there are less pressure vessels involved and the regulation of the fluid supply is easier.
- e) Parallel to the steps listed - reconciling with the customer, if required - the manufacturers and types of the main equipment can be decided upon.

Point 3

No worksheet corresponds to this; this point provides, based on the design so far, the opportunity to the customer - if required - to monitor the design process and to amend data if necessary.

Point 4

As it had already emerged in relation to selecting the refrigerant, **Worksheet no 11** assists in preparing the licensing documentation found under the filename “worksheets_1_18”.

The cells with yellow background must be filled in the worksheet, and the classification is then carried out by the software according to the referenced standard. A few sample rows are filled; these should naturally be deleted.

This worksheet assists with the documentation to be submitted to the “Regional Technical Safety Authority”.

In addition, the design for approval must contain as a minimum a brief technical description of the main features of the refrigerating equipment, indicating the refrigerant to be used, the functions of the pressure vessels in the system and a general arrangement plan showing the layout of the key components of the refrigerating equipment. Apart from this, the noise levels of the units placed outdoors as well as their electricity and water requirement must be given.

Point 5

No worksheet corresponds to this; in this point, the occasional data amendments are effected, based on the requirements of the building permit, i.e. the observations of the authorities.

Points 6 and 7

In these points, the plans are finalised together with the co-designers. The electrical designer is provided the power requirement of the components of the refrigerating system and the expected synchrony of these; the structural designer is provided the masses of the components of the refrigerating system and their support/suspension points; the designer architect receives the data required for the appropriate locations of the components, their space requirements and the sanitary engineer is provided with the water and ventilation requirements. To determine the emergency ventilation requirement of the machine room, **Worksheet 1** can provide assistance, if the refrigerant charge is known. For the static data derived from the weights of the pipes and pipe bundles, data are provided by **Worksheets 2 and 5**. **Worksheet 1** helps determine the support spacing for the pipes and pipe bundles and to reconcile this with the design of the building structure. For copper pipes, **Worksheet 15** can be used for this.

Points 8 and 9

The final vision of the designer is documented in these points. Plans must be prepared to a depth of detail so that any relevant professional (constructors, inspectors, etc.) should precisely understand what the designer had in mind and be able to apply the information contained therein. Naturally, the initial parameters of the design reconciled with the customer as well as those provided to the co-designers must be documented. I have put points 8 and 9 together into a single point, because in some cases, either the tendering or the implementation phase may be omitted.

- a) For the strength/geometric dimensioning of the pipeline, **Worksheets 2, 3, 4 and 5** provide assistance. In order to determine the pipe dimensions and to select the appropriate pipes for the capacity, I recommend the CoolSelector software by Danfoss, which can be deemed the best in this particular field, and may be downloaded from the website Danfoss.com (<http://refrigerationandairconditioning.danfoss.com/support-center/apps-and-software/coolselector/>). The data of the refrigerants, cycles, etc. are best made available using the software downloadable from the link <http://refrigeration-utilities.software.informer.com/> published by the Department of Energy Engineering, Technical University of Denmark.
Earlier, in the 1970s, when PCs and laptops were nowhere in sight, I prepared various nomograms to make calculations easier, among others for the selection of pipe dimensions. I attach these here too, should a PC be wanting or the laptop's battery have just run down.
- b) **Worksheet 6** can be helpful if a detailed pipeline strength dimensioning must be submitted, e.g. for TÜV licensing.
- c) **Worksheet 7** serves to calculate the convective heat transfer coefficient ($\text{W/m}^2\text{K}$) achievable on the inner wall of the pipe for any medium flowing in various pipes. The

tables for commonly used transfer media are filled out, but further data may also be added (up to 12 lines), or those already entered may be modified.

- d) **Worksheet 8** serves to calculate the convective heat transfer coefficient ($\text{W/m}^2\text{K}$) for ammonia flowing and evaporating on the inner side. This may be useful for directly cooled ice rinks.
- e) **Worksheet 9** is likewise useful for ice rinks. It assists in working out the appropriate layer thicknesses, selecting the pipe material, the pipe arrangement and the required temperature of the refrigerant. The table also provides guidance concerning the thermal load of the rink. For indoor rinks to be used during the summer, the special effects must be precisely clarified, e.g. whether matches would be aired on TV (requiring heavy duty lighting), or the air drying provided by the ventilation system, if boards would be installed, etc.
- f) **Worksheet 10** can be used for dimensioning freely supported beams. This may be useful for supporting certain equipment or designing other structures, as the majority of cases can be reduced to this. For designing more complicated trusses, a structural engineer should be consulted.
- g) **Worksheet 13** helps determining the flow resistance of indirectly refrigerated systems, with the selection of the circulator pump. The physical parameters of the transfer medium are taken from the same table as for **Worksheet 7**.
- h) **Worksheet 14** can be used for dimensioning supporting columns. The dimensioning simplifies the cases of buckling as a result of a compressive force (with articulated top connection) and the load combined with torque as well. The worksheet also contains 5 standard solutions, but it is also possible to dimension different columns as well.
- i) **Worksheet 24**, found under the filename “**worksheets_21_24**” contains the dimensions for flat face flanges used in transporting ammonia and other substances, assisting in the design of various construction details.
- j) **Worksheet 15** provides the weights and support/suspension point data of the copper pipes, similarly to Worksheet 5 for steel pipes.
- k) **Worksheet 16** can be used for selecting the blow-off valves for pressure vessels. Such an option is also available in Danfoss’ “CoolSelector” programme.
- l) **Worksheet 17** can be used to study the falling speed of ammonia droplets that may be useful for dimensioning separators.
- m) **Worksheet 18** assists in selecting pressure equalising valves for refrigerated spaces. Pressure equalisers successfully prevent the pressure decrease resulting from the cooling of a closed refrigeration chamber causing any damage to the building (most commonly the ceiling caving in).
- n) **Worksheet 22**, found under the filename “**worksheets_21_24**” contains the dimensions of various frequently used support profiles and box sections and their second moments of area, assisting in designing simple supporting structures.

- o) **Worksheet 23**, found under the filename “**worksheets_21_24**” may be the source of arcs, vessel and pipe cap dimensions, valve sizes and support bracket dimensions, assisting in the preparation of detail drawings, double line piping layout plans, etc.
- p) **Worksheet 24**, found under the filename “**worksheets_21_24**” contains solutions for fixed and sliding clamping insulated and bare pipes, typically for pipes insulated with polyurethane pipe coats. Under 10 mm nominal diameter, HILTI’s insulated clamps are also used.
- q) **Worksheet 25** helps with thermal insulation of pipelines, taking into account the ambient temperature and humidity, with polyurethane, Armaflex or Kaiflex insulating materials.

The worksheets are also introduced in short videos; these can be found in the attachment.

22.11.2015



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Notes for the attachments:

The nomograms mentioned in point 9/a and below, used for determining pipe dimensions, are based on the following correlations:

1. $G = Q_o / q_{ovszm} \cdot v''$ (kg/s)
2. or $G = Q_o / \Delta h$, and $V = G \cdot v$ (m³/s)
3. $F = V / w$ (m²)
4. $F = d^2 \times \Pi / 4$ (m²)

Where:

q_{ovszm} is the volumetric heat capacity (kJ/m³)
 v'' is the specific volume of the vapour, (m³/kg)
 Q_o is the refrigeration capacity (kW)
 Δh is the change in heat capacity in the evaporators concerned (kJ/kg)
 V is the volumetric flow rate (m³/s)
 v is the specific volume of the flowing medium (m³/kg)
 w is the flow rate of the medium (m/s)
 F is the cross section of the flow (m²)
 d is the inner diameter (m)

Combining equations 1-3 yields:

$$F = Q_o \times v / q_{ovszm} \cdot v'' / w$$

Combining the parts containing the physical properties of the refrigerant into a variable A dependent on the medium and the operational parameters (operational temperature), the equation will be as below:

$F = Q_o \times A/w$, and after rearranging the formula: $F \times w = Q_o \times A$

Taking the logarithms of both sides, we obtain: $\lg(F) + \lg(w) = \lg(Q_o) + \lg(A)$

As it can be seen, a nomogram with a set of straight lines can be created in logarithmic form, which is important because this has an advantageous influence on the accuracy of construction and reading using the graphs.

The usual speeds:

Description	Flow rate (m/s)	Remarks
Suction pipe (dry)	10~20	Pressure loss ~1 K
Suction pipe (with pump)	10~12	Pressure loss ~1 K, max 2 K at -40
Pressure pipe (from compressor)	12~25	
Fluid line (from condenser)	<0.6	
Fluid line (from tank to feeder)	0.5~1.25	No evaporation is allowed due to pressure loss or any other cause
Fluid line (pumped pressure pipe)	1~1.5	
Steam speed in upright fluid separators	0.2~0.6	Other considerations (e.g. movement of charges) must also be taken into account for the selection!

The nomograms for freon media are in essence the same, but:

- The part for selecting the separator is omitted here,
- The dimensioning of the pressure pipe for the lower stage is omitted.

The usual speeds:

Description	Flow rate (m/s)	Remarks
Suction pipe (dry)	10~20	Pressure loss ~1 K
Pressure pipe (from compressor)	12~25	
Fluid line (from condenser)	<0.6	
Fluid line (from tank to feeder)	0.5~1.5	No evaporation is allowed due to pressure loss or any other cause

For variable loads, consider the volumetric flow of the oil when selecting the speed!

