Ammonia a trendy refrigerant

T.M.C. Bartholomeus
Grasso Products B.V., P.O. Box 343
's-Hertogenbosch, the Netherlands

Abstract

Reciprocating compressors are limited in their field of application due to the compression-heat. At the end of the compression stroke, the gas-temperature may not exceed 200'C. (+/- 150'C after the discharge-valves) High thermal stress of valves and carbon-scaling, will reduce valve-life-time drastically.
For ammonia, as a rule of thumb, the maximum temperature difference between the condenser and evaporator is about 50 K at full-load operation. At part load this must be reduced by 10 K, due to the heat dissipated by the non-active cylinders. The welded construction of the Grasso compressors with external positioned cylinders, suction- and discharge lines allows a 5 K higher temperature difference, due to better heat dissipation.
To cool down the discharge-gas by means of cylinder head (water- or air-cooled) is a conception, as it does not influence the thermal condition of the gas passing the discharge-valves.

When higher temperature differences are required, 2-stage compression is necessary.
The compression-heat put in the gas of the first-, must be removed before entering the second stage. Due to the low level of the second stage suction gas temperature, air or water can not be used, so the plant refrigerant is used, in different ways, to cool down the first stage discharge gas.

With this paper we want to illustrate a number of 2-stage intercooling systems, engineered to meet the latest demands on refrigerant charge and highlight new developments.

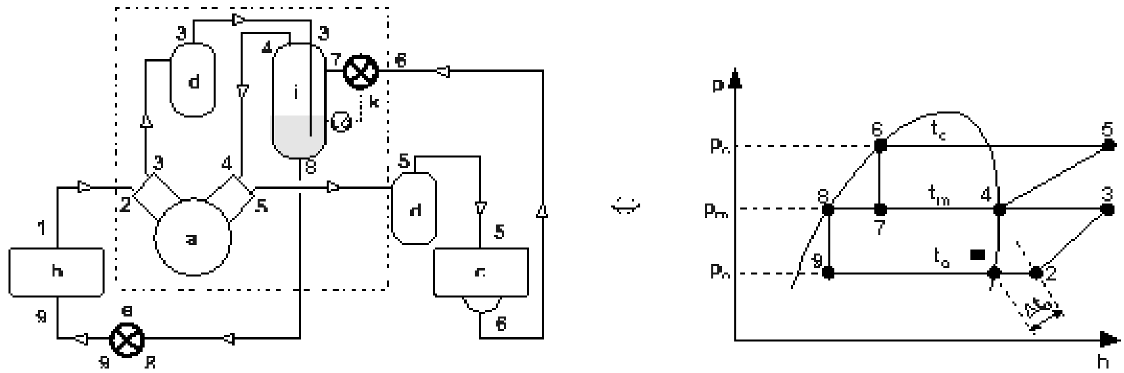
Introduction

The classical way to cool down superheated gas is to bubble it into a large pool of liquid refrigerant, the so called open flash cooler. (Grasso system C). Sizing is done related to the swept volume of the second stage cilinders. To protect the compressor against wet running, these intercoolers are sized generously.

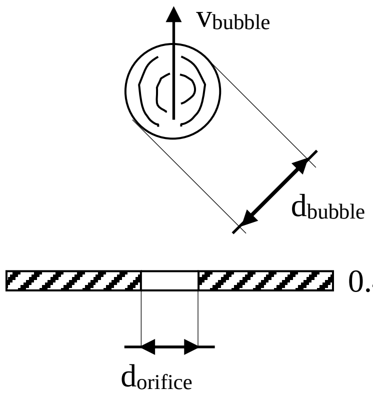
$d_{internal}(mm) := C \cdot (speed(rpm) \cdot cilnumber)^{0.5}$ Where C depends of the type of compressor.
The formula assumes highest allowable intermediate pressure, so highest possible gas flow and lowest possible droplet separation velocity. The first due to the smallest pressure ratio, thus best volumetric efficiency, the latter due to the highest gas density. The liquid level above the bubble sparger was nearly 1 meter, to obtain 5 K superheat. Regarding refrigerant pricing, the necessary refrigerant charge makes the simple and reliable open flash gascooler for halocarbons of less interest.

Liquid injected, by means of a thermostatic expansion valve, into the connecting line between first- and second stage is a simple solution, however often proven unreliable. At that time, the mechanism to explain this kind of gascooling was droplet evaporation. To fully evaporate a halocarbon droplet it takes less than a split-second. For ammonia to evaporate the same droplet would take 5 times more. Depending on the average droplett size and gas velocity, not all refrigerant should be evaporated, when entering the second stage. To increase the droplet transfer time a special pipe in pipe unit was fitted. (Grasso system A)

1. Open flash intercooler (Grasso system C)



To understand the function of the open flash intercooler, we have to understand the bubble mechanism.



$0.4 < t_{plate} / d_{orifice} < 0.7$

Depending of the Reynolds number of the gas passing through the orifice, the diameter of the orifice and the refrigerant-data, the bubble diameter can be estimated. The old type of intercoolers worked in the luminary region of Reynolds < 2300, creating bubbles triple the size of the orifices in the perforated (35%) sparger plate. Big bubbles > 20 mm can not maintain their size and break up into smaller ones. The bubbles terminal velocity is reached almost directly after creation. Increasing the orifice speed to turbulent level, the gas stream gets the appearance of a jet, breaks up 7.6 to 10.2 cm above the orifice. Actually the stream consist of large, closely spaced, irregular bubbles with a rapid swirling motion. These bubbles disintegrate into a cloud of smaller ones of random size distribution between 2.5 mm and 12.7 mm, with a mean size of about 4-mm.

The bubble terminal velocity can be computed with the friction-law of Newton, for motive balls. In case of Reynolds numbers 100<Re<100000 the Cw value remains constant at 0.44, so the terminal velocity will be reached when the buoyancy- and flow-resistance forces equal each other;

$$v_{bubble} := \left[\frac{\frac{\pi}{6} \cdot d_{bubble}^{3} \cdot \left(\rho_{liquid} - \rho_{gas}\right) \cdot 9.81}{\left(C_{w} \cdot \frac{\pi}{4} \cdot d_{bubble}^{2} \cdot \frac{\rho_{liquid}}{2} \right)} \right]^{0.5}$$

However, due to the mushroom kind of shape, of the large bubbles the projected area as well as the Cw-value increase, thus reducing the bubble velocity.

Out of many observations, it seems that the bubble velocity never exceeds 0.3 m/s.

The overpressure necessary to create bubbles must be higher than; $\dfrac{4\cdot\sigma}{d_{orifice}}$

To prevent liquid weeping through the sparger-(sieve) plate, the overpressure underneath it should always be higher than the liquid head above it, so;

$$\zeta_{orifice}\cdot\frac{\rho_{gas}}{2}\cdot v_{gas}^{2}+\frac{4\cdot\sigma}{d_{orifice}}>h_{liquid}\cdot\rho_{liquid}\cdot 9.81$$

Cooling of the gasbubble can be best described by the Nusselt number; $Nu := 2 + 0.6\cdot Re^{\frac{1}{2}}\cdot Pr^{\frac{1}{3}}$

where as; $Re := \dfrac{d_{bubble}\cdot v_{bubble}\cdot\rho_{liquid}}{\eta_{liquid}}$ $\quad Pr := \dfrac{\eta_{liquid}}{\lambda_{liquid}}\cdot cp_{liquid}$

So ; $\alpha_{bubble} := Nu\cdot\dfrac{\lambda_{liquid}}{d_{bubble}}$

Using the "old" Planck cooldown formula transformed to a gas bubble, the time (seconds) necessary to cool down the gas to the required superheat is;

$$\tau := 0.311\cdot cp_{gas}\cdot\rho_{gas}\cdot\left(\log\left(\frac{t_{discharge}-t_{intermediate}}{\delta t_{superheat}}\right)-0.0913\right)\cdot\left(\frac{d_{bubble}}{\alpha_{bubble}}+\frac{d_{bubble}^{2}}{4\cdot\lambda_{gas}}\right)$$

Taking care of turbulent flow in the orifice, we can use the mean bubble diameter of 4 mm to calculate. At the usual intermediate saturation temperatures and an accepted superheat less than 5 K, the required cooling down time will be less than one second! With a bubble velocity of < 0.3 m/s, the required liquid height above the bubble sparger is < 0.3 m, so the refrigerant charge is reduced by 70% !!

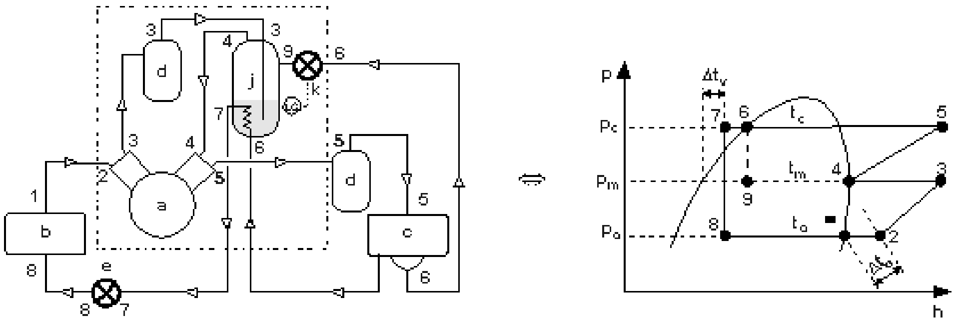
The gas hold-up increases with increasing superficial gas velocity in the liquid. As it makes no sense to go above the maximum rise velocity of the mean bubble, the maximum gas hold-up is limited to 40% (Halocarbons >50%!). If the level control is fitted to the mentioned 300 mm, at start, the liquid level could foam up to 0.3/(1-0.4) equals 0.5 m ! Just imagine what happens in the old versions, or halocarbon versions...

A height reduction, related to the old version, of 1 meter is possible!!

As bubbles collapse when they reach the surface, small droplets will jump up for at max. 200 mm, so above this level, the suction connection of the second stage should be fitted.

The liquid injection should be done in the gas sparger, to prevent splashing in the separation area above the liquid.

2. Open flash intercooler with subcooler (Grasso system D)



To obtain maximum pressure difference over the injection valves on pumpstations or evaporators, a subcooling coil is submerged in the boiling liquid of the open flash intercooler. The heat transfer calculation is the same as used to select flooded liquid coolers.

The internal heat transfer of the flowing condensate can be done with the well-known Nusselt formula for flow in tubes;

$$\alpha_{condensate} := \frac{\lambda_{condensate}}{d_{internal}} \cdot 0.032 \cdot Re_{condensate}^{\ 0.8} \cdot Pr_{condensate}^{\ 0.3} \cdot \left(\frac{d_{internal}}{pipelength}\right)^{0.054}$$

Due to the high-pressure difference available, condensation- minus evaporation pressure, it is of negligible influence to accept a subcooling coil pressure loss of about 1bar.
These enables to run the subcooling coil with high velocity thus max. internal heat transfer coefficients. However it should not exceed 3 m/s, to prevent cavitation.

The external heat transfer can be computed with the boiling formula out of the V.D.I. Waermeatlas chapter Ha ;

$$\alpha_{boil} := \alpha_{nil} \cdot f \cdot \left[\frac{heatloadcoil \cdot \frac{kW}{m^2}}{q_{nil}}\right]^n \cdot \left(\frac{r \cdot \mu m}{r_{nil}}\right)^{0.133} \cdot \left[1 + \left(2 + heatloadcoil \cdot \frac{kW}{m^2}\right)^{-1}\right]$$

Where as; $q_{nil} := 20 \cdot \frac{kW}{m^2}$ $\alpha_{nil} := 4000 \cdot \frac{W}{m^2 \cdot K}$ $r_{nil} := 1 \cdot \mu m$ $p_{krit} := 112.6 \cdot bar$

$$p := \frac{p_{intermediate}}{p_{krit}} \qquad n := 0.9 - 0.3 \cdot p^{0.3} \qquad f := 2.1 \cdot p^{0.27} + \left[4.4 + \left(\frac{1.8}{1 - p}\right)\right] \cdot p$$

The overall heat transfer must be computed by means of iteration.
The oil fouling must also taken into account;

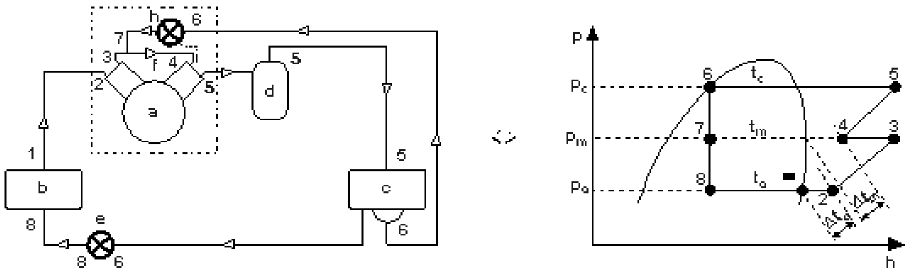
$$k_{subcoolingcoil} := \frac{1}{\left[\left(\frac{1}{\alpha_{internal}} + \frac{t_{oil}}{0.14}\right) \cdot \frac{d_{outside}}{d_{internal}} + \frac{1}{\alpha_{boil}}\right]}$$

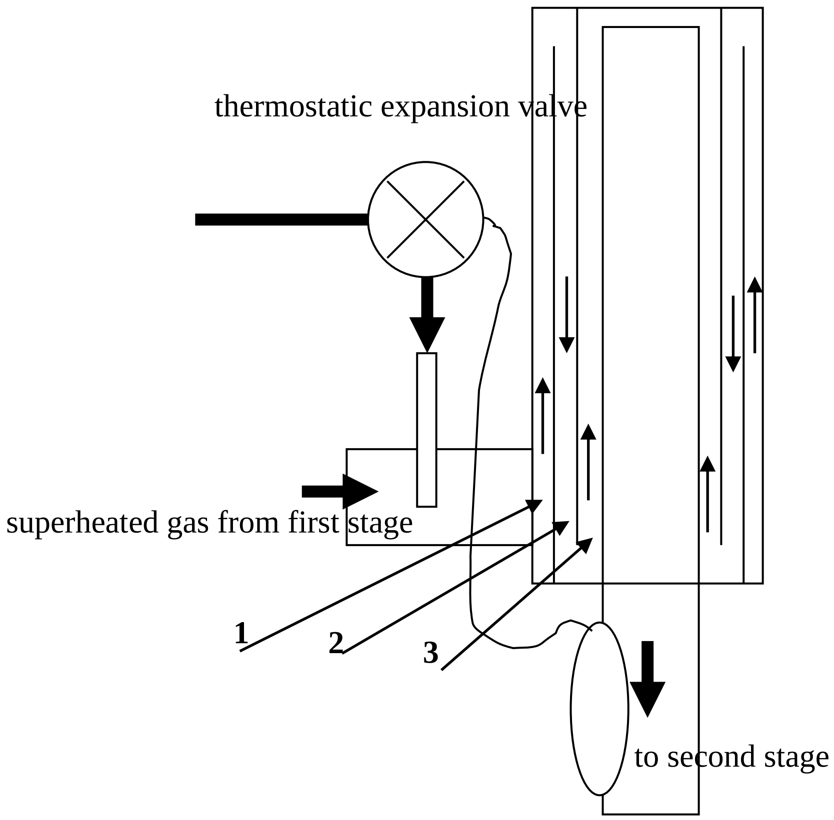
The fouling due to the oil has been described by E.Berends & J.G. Romijn during the Oslo meeting in 1998;

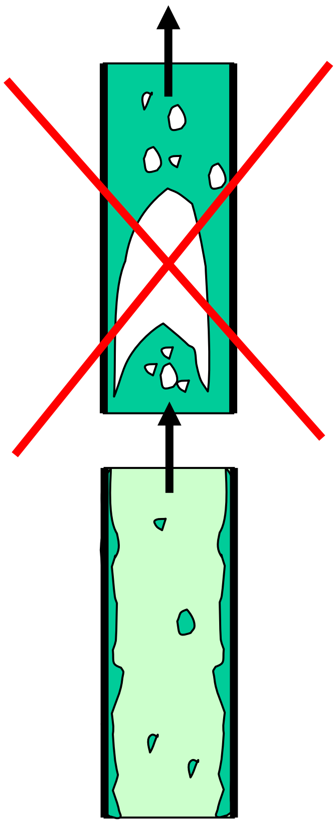
$$t_{oil} := \left(\frac{4 \cdot \eta_{oil} \cdot c_{oil} \cdot d_{internal}}{v_{condensate} \cdot \rho_{oil} \cdot \lambda_{pipe}} \right)^{0.5}$$

The concentration of oil depends on how the refrigeration-system after the condenser(s) is executed.
If the condensate line to the subcooling coil is connected above the intercooler injection, on a quit spot, the oil concentration is nearly nil.
“K”-values > 4000 W/m2K are measured in those kind of systems!
In all other situations, the value of 10 ppm in the discharge gas, as given by E.Berends & J.G. Romijn, corrected with the oil separator efficiency should be used.
Overall “K”-values will be found of about 1500 W/m2K.

3. Gascooler with liquid injection (Grasso system A)







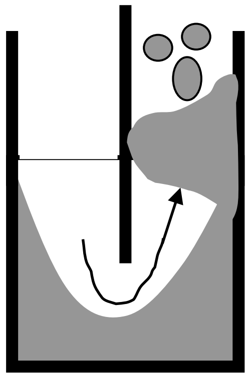
To cool down the first stage superheated discharge gas and to evaporate the injected refrigerant it is a must to create a two-phase ringflow in the annular spacing of the pipe in pipe gascooler.

Due to the ringflow, the flow guiding pipe-inserts (section 1 and 2) are wetted completely and act as heattransfer area. The gas passing with high velocity, transfers it superheat to the wetted guiding pipes by convection and evaporates the layer of boiling refrigerant. Section 3 is the dry out zone, the outer wall still has intermediate temperature as at the section 2 side, a refrigerant layer is on. Section 4 is the mixing pipe, here the evaporated refrigerant mixes up with the cooled down gas, to pass the bulb and enter the second stage with the pre-set superheat. With the process-evaluation above, the heat- and mass transfer can be computed.

Ammonia requires large heat transfer area's. As the gascoolers are meant to be built on compound-compressors, a horizontal version would be more appropriate.

Both versions require their own specific engineering-approach.

What happens, if the gasvelocity in the annular spacing is to low?
In the vertical gascooler, liquid will start to accumulate at the bottom, until a level is reached that liquid is choked upwards. Once passing the bulb of the thermostatic expansion valve, latter doesn't now how to handle and starts to hunt. Now liquid-droplet's will enter the second stage, undermine the lubrication seal on the cylinder-liner, which immediately will result in excessive wear.
In the vertical gascooler, at to low gasvelocities, the liquid will flow at the bottom. To evaporate the liquid of this wavy creek, an enormous length is necessary. The choke flow as in the vertical one will not occur.



3.1 What is the minimum gas velocity to obtain ring flow?

The ASHREA 1994 refrigeration handbook 2.14 table 15, advises certain minimum pressure losses in vertical halocarbon piping to ensure lubrication oil to be entrained with the gas flow;
Internal diameter d < 50 mm, at suction conditions –18'C, 80 Pa/m
Internal diameter d > 50 mm, at suction conditions –46'C, 100 Pa/m
When we convert above mentioned values;

$$\delta p := \lambda \cdot \frac{L}{d_i} \cdot \frac{\rho_{gas}}{2} \cdot v_{gas}^{2} \qquad v_{gas} := \left(\frac{\frac{\delta p}{L} \cdot d_i \cdot 2}{\lambda \cdot \rho_{gas}} \right)^{0.5} \qquad \lambda := 0.025 \quad \text{than :} \quad v_{gas} > \frac{20}{\rho_{gas}^{0.5}}$$

Using the flow-chart of the V.D.I. Waermeatlas, section Lgb2, we can convert that ringflow in a vertical pipe is possible when;

$\frac{m^2 \cdot x^2}{\rho_{gas}} > 100 \qquad \frac{m_{gas}}{\rho_{gas}} := v_{gas}$ so with help of $\frac{m^2 \cdot x^2}{\rho_{gas}^{\ 2}} > \frac{100}{\rho_{gas}}$ and $m \cdot x := m_{gas}$

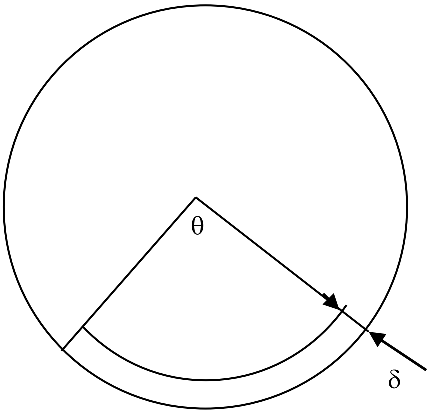
$$v_{gas} > \frac{10}{\rho_{gas}^{\ 0.5}}$$

In the first situation it concerned oil with solved refrigerant in latter pure refrigerant. As the first stage discharge gas as well as the injected liquid contain oil, we use in part load condition the lowest value. Only the oil will accumulate at the bottom, waiting to be removed at full load. Similar to the, well-known, double riser principle.
In case of the horizontal gascoolers gravity requires extra attention. The calculation-methods used in the VDI-Waermeatlas are for this use unpractical. An article read in i^2-Procestechnologie no.12-1987 was of better help. It reports about 2-fase flows with very low liquid fractions $\varepsilon_{liquid} < 0{,}04$ Liquid-fractions in the gascoolers are in this region. The usual flow-behaviour methods have shown to be very unreliable in this region. (Deviation of 100%) The authors made a correlation with help of dimension-analysis and use empirical constants, which correspond with their measured values;

$$\varepsilon_l = (1+10{,}4x(v_{gas}/v_{liquid})^{2/3}x(\rho_{gas}/\rho_{liquid})^{1/3}x(\eta_{gas}/\eta_{liquid})^{1/4})^{-1}$$

(v_{gas}/v_{liquid}) = relation superficial velocities gas/liquid

To estimate the wetted wall fraction, the model of Hamersma & Hart is used;



$\theta = (1/\pi)$ x $\arccos(1- 0{,}66$ x $Froude_l)$

$Froude_l = (\rho_{liquid} /(\rho_{liquid}-\rho_{gas}))$ x $(v_{liquid}^2/(\varepsilon_{liquid}^{\ 2} x\ g\ x\ d_{hydraulic})$

To ensure full wetting, $Froude_l > 3$

v_{gas}= gasvolume/ ring area with other words; x x $(m_{HD} /\rho_{gas})/D$

v_{liquid} = liquidvolume/ring area with other words; $(1\text{-}x)x(m_{HD} /\rho_{vloeistof})/D$,

$v_{gas}/v_{liquid} = (x/(1-x))$ x $(\rho_{liquid} /\rho_{gas})$ and $v_{liquid} = ((1-x)/x)$ x $(\rho_{gas} /\rho_{liquid})$ x $v_{gas})$

By substitution in the $Froude_l > 3$ formula, this will be;

$(\rho_{liquid} /(\rho_{liquid}-\rho_{gas}))$ x $(((1-x)/x)$ x $(\rho_{gas} /\rho_{liquid})$ x $v_{gas})^2/(\varepsilon_{liquid}^{\ 2} xgxd_{hydr}) > 3$ and

$\varepsilon_l = (1+10{,}4x((x/(1-x))^{2/3}x(\rho_{liquid}/\rho_{gas})^{1/3}x(\eta_{gas}/\eta_{liquid})^{1/4})^{-1}$

After some converting, the formula for minimum gasvelocity to obtain ringflow is as follows:

$$v_{min} := \frac{\left(\frac{x}{1-x}\right)\cdot\left(\frac{\rho_{liquid}}{\rho_{gas}}\right)\cdot\left(3\cdot g\cdot d_{hydr}\cdot\frac{\rho_{liquid}-\rho_{gas}}{\rho_{liquid}}\right)^{0.5}}{\left[1+10.4\cdot\left(\frac{x}{1-x}\right)^{\frac{2}{3}}\cdot\left(\frac{\rho_{liquid}}{\rho_{gas}}\right)^{0.333}\cdot\left(\frac{\eta_{gas}}{\eta_{liquid}}\right)^{0.25}\right]}$$

In partload situations the real wetted- so working surface area, can be computed with the θ formula, which after some work looks like this;

$$\Theta := \frac{1}{\pi}\cdot\mathrm{acos}\left[1-\left[\frac{\rho_{liquid}}{\rho_{liquid}-\rho_{gas}}\cdot\frac{\left[\left(\frac{1}{\rho_{l.i}}\cdot\frac{m_{secstage}\cdot x}{ringarea}\cdot\frac{1-x}{x}\right)\cdot\left[1+10.4\cdot\left(\frac{x}{1-x}\right)^{\frac{2}{3}}\cdot\left(\frac{\rho_{liquid}}{\rho_{gas}}\right)^{0.333}\cdot\left(\frac{\eta_{gas}}{\eta_{liquid}}\right)^{0.25}\right]\right]^2}{d_{hydr}\cdot g}\right]\cdot 0.66\right]$$

3.2 The heat transfer of turbulent gasflow in annular piping systems

$Nu = 0{,}032 \times Re^{0,8} \times Pr^{0,33} \times (d_{hydr}/L)^{0,054}$ $Nu = \alpha \times d_{hydr}/\lambda$

$m_{LD} \times cp_{gas} \times (t_{discharge} - t_{endsect}) = \alpha_{gas} \times F_{section} \times dt_{log}$

After some converting, the end-temperature can be calculated with;

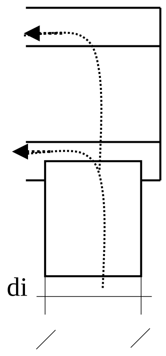
$$t_{end} := \left[\frac{t_{discharge} - \left(t_{intermediate} + \delta tpressloss\right)}{e^{\left(\frac{\alpha_{gas}\cdot areasection}{mass_{secstage}\cdot cp_{gas}}\right)}}\right] + \left(t_{intermediate} + \delta tpressloss\right)$$

The section area is the addition of internal surface of the outer pipe and the outer surface of the inner pipe.

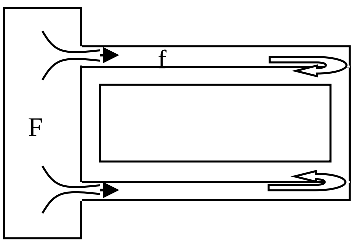
Remark: the oil will form a layer on the surface area, however the boiling refrigerant will flow over it and as such we do not have to calculate fouling.

3.3 What about pressure loss?

FS-gasinlet $\xi_{in} = 3$



$\xi_{annular\text{-}inlet\ f/F=0,7} = 0{,}54$ $\xi_{annular\text{-}outlet\ h/dh>1} + \xi_{180'turn} + \xi_{annular\text{-}inlet\ f/F=0,7} = 1{,}9$



Pressure loss along the annular length = λ x L/d_h , λ is dependent of refrigerant, flashgas-amount, mass-flow and annular-geometry.

The vertical gascoolers can be calculated with the methods out of chapter Lg 1 VDI Waermeatlas 1974.

In the same article of I^2 –Process-technologie, following formula can be extracted;

$$\lambda_{section} := \left[-0.8685\cdot\ln\left[\frac{1.964\cdot\ln\left(Re_{gas}\right)-3.8215}{Re_{gas}}+\frac{\left[\left[1+10.4\cdot\left(\frac{x}{1-x}\right)^{\frac{2}{3}}\cdot\left(\frac{\rho_{liquid}}{\rho_{gas}}\right)^{0.333}\cdot\left(\frac{\mu_{gas}}{\mu_{liquid}}\right)^{0.25}\right]\right]^{-1}\cdot 0.575}{3.715}\right]\right]^{-2}$$

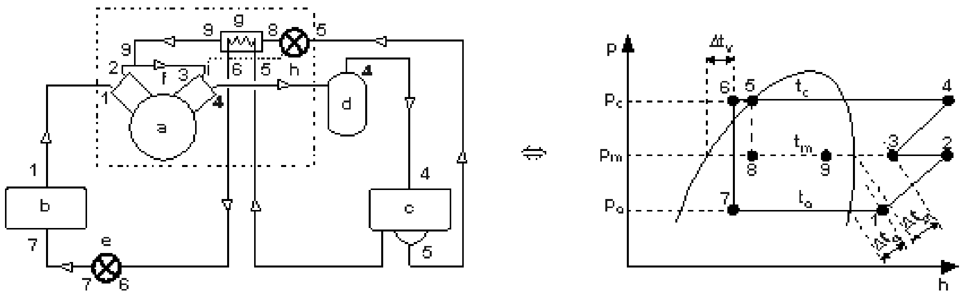
For section 1 the pressure loss will be 3 + 0.54 + λ x L/d + 1.9, section 2 & 3, each λ x L/d + 1.9

3.4 The expansion valve

A thermostatic expansion valve is a mechanic control device. As such it responds slow on rapid changes. A gascooler is one of the quickest respond heat exchangers, as the heat transfer is direct and the refrigerant charge is less than 4% by volume (direct expansion coolers > 18%) so immediately dried out or filled up. A compressor capacity step will immediately result in strong fluctuation of the superheat. Setting the superheat at 15K and sizing it 20% smaller than the list capacity, will keep the fluctuation less than 5K. Expansion valve suppliers oversize their valve to compensate flash gas in the condensate supply line. Take care of subcooling, expansion valve capacity increases about 2%/K. The most severe conditions to work under for the mechanical expansion valves is strong fluctuating working pressure differences as in blast freezer-systems and

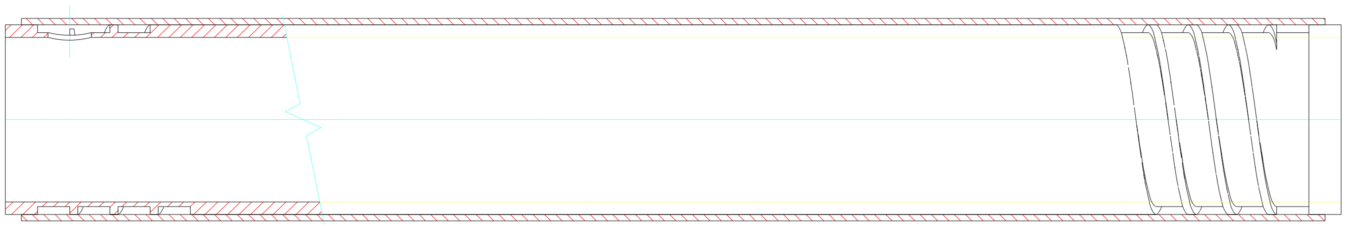
such alike, where the suction-, intermediate and discharge pressure can change rapidly. In those cases an adaptive controlled electronic expansion valve solves the problems(Grasso Ecotron). Before the capacity step takes place, the opening degree of the valve is preadjusted to the new situation. Gascoolers with such control run steady at a superheats < 10 K +/_ 1K.
Still, even the best expansion valve can't control an unproper sized gascooler!

4. Gascooler with integrated subcooler (Grasso system B)



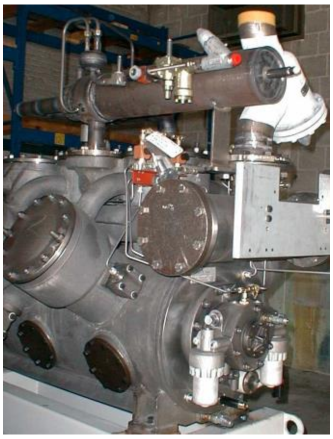
As it is very hard to distribute expanded ammonia with a liquid fraction less than 5% over multiple channels or tubes, a separate subcooler of the plate- or shell&tube heat exchanger type is no solution. The wet refrigerant out of the subcooler enters the gascooler, the expansion valve measures the superheat after de gascooler. Non uniform distribution of liquid-ammonia will immediately cause hunting. Usually the expansion valve is blamed for the disfunctioning, however the heat exchanger is not able to produce a stable M.S.S.

Realising that the flow guiding pipes are optimal wetted in the gascooler, why not use one and make it double walled. In between both walls, a spiral shaped channel is machined as in cylinderliners of water-cooled combustion engines. However instead of cooling-water, condensate flows through, which will be subcooled by the thin layer of boiling ammonia on both outside-walls.



Heat transfer calculations on the condensate side are done with the already used Nusselt formula for flow in tubes. The heat transfer coefficient of the thin layer of boiling ammonia is done with the Chawla formula;

$$\alpha_{boil} := \frac{0.0115 \cdot \lambda_{liquid} \cdot cp_{liquid}^{\ 0.06}}{g^{0.3} \cdot \rho_{gas}^{\ 0.66} \cdot \left(\eta_{liquid}\right)^{0.575} \cdot \left(\eta_{gas}\right)^{0.225}} \cdot \frac{\left(\frac{m_{secstage}}{ringarea}\right)^{1.4}}{\left(d_{hydr}\right)^{0.5}}$$



The heat resistance due to the oil in the first stage can be calculated with the same formula as used in the subcooling coil calculation. The oil concentration is about 70 ppm.
At partload, the wetted surface could reduce, as previous explained. For those conditions the dried out zones are in contact with the hot discharge gas. The passing gas will partly dissipate its heat to the subcooled condensate.

$$dt_{logwarmup} := \frac{\left[\left(\frac{t_{discharge} + t_{outsection}}{2}\right) - t_{condensation}\right] - \left(t_{condout} - t_{intermediate}\right)}{\ln\left[\frac{\left[\left(\frac{t_{discharge} + t_{outsection}}{2}\right) - t_{condensation}\right]}{\left(t_{condout} - t_{intermediate}\right)}\right]}$$

The temperature difference reduction due to the heating up is;

$$\delta t_{heatup} := \frac{\left[\left[\alpha_{gas} \cdot outsidearea \cdot (1 - \Theta)\right]\right] \cdot dt_{logheatup}}{m_{secstage} \cdot cp_{liquid}}$$

5. Conclusion

This paper shows that by proper use of 2-fase physics, well-known conventional designs, can have a face-lift" to meet today's requirements without losing reliability. The refrigerant charge of open flash coolers can be reduced with 70% or more!

A new type of gascoolers with direct expansion is developed where refrigerant charge could be reduced with 99% and the contracting costs with 75%, both related to the conventional open flash intercooler.