DESIGN & SELECTION OF PRESSURE VESSELS & HEAT EXCHANGERS FOR AMMONIA REFRIGERATION

By

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PRESSURE VESSELS USED IN AMMONIA REFRIGERATION SYSTEMS

1. Oil Separator after compressor & before condenser
2. High pressure Liquid Receiver for ammonia storage
3. Inter-stage cooler-open flash type- for two stage system
4. Inter-stage cooler –with closed coil –for two stage system
5. Low pressure Ammonia storage vessel- for forced feed pump circulation system (L.P. Vessel)
6. Surge drum mounted on shell & tube flooded chiller
7. Accumulator for gravity flooded systems-cold storage air coolers, PHE, ice bank etc.
8. Knock out drum/liquid separator to protect compressor from liquid
9. Oil Pot for oil draining

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HEAT EXCHANGERS USED IN AMMONIA PLANTS

1. Shell & Tube Condensers
2. Heat recovery exchanger in discharge line
3. Shell & Tube evaporators for water & brine
4. Plate Heat Exchangers
5. (PHE) condensers/Chillers
6. Falling Film water chillers.
7. Evaporative condensers
8. & many other types of heat exchangers

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REFERENCE STANDARDS

1. ANSI/**IIAR-2 2014-**Safety Standard for Equipment, Design, and Installation of Closed-Circuit Ammonia Mechanical Refrigeration Systems
2. **AAR-1-** standard for safe design of closed circuit ammonia refrigeration systems
3. **ANSI/ASHRAE-15** 2013-Safety standard for refrigeration systems
4. **ASME B31.5** 2019-Refrigerant piping and heat transfer components
5. **ASME Sect V III-Div.** 1-Rules for construction of pressure vessels

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REFERENCE STANDARDS

6. **TEMA-2019**-Standard of Tubular Exchangers Manufacturer's Association

7. **IS/ISO-16528-1-2007-**Boilers and pressure vessels requirement

8. **EN 378-2-2016-**Design, construction, testing requirements

9. **IS 660- 1963-**Safety code for mechanical Refrigeration

10. **OSHA-** Occupational Safety & Health Association**-**Safety standard for refrigeration systems(ANSI/ASHRAE 15-2001, (2001)

11. **IS-2825/14430-July2019**-Closed circuit Ammonia refrigeration system-Code of practice for unfired pressure vessels

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FPS OR SI?

We shall make this presentation in FPS units for following reasons

1. In FPS unit all calculations are on per hour basis where as in SI units it is W=J/s, hence every time we either need to multiply or divide by 3600 to convert in to hourly values
2. The temperature scale is much wider-32^{0}F to 212^{0}F as against 0^{0}C to 100^{0}C
3. The atmospheric pressure scale in FPS is much wider -14.7(14.696) PSIG whereas in SI it is 1.033kg/cm^2
4. Specific heat of water is 1Btu/lb^{0}F-in SI units it is 4.186J/kg.K
5. It is easier to remember whole number rather than fraction-for example we can remember 10 psig where as in SI it is 0.7031kg/cm^2

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DESIGN BASIS PARAMETERS

1. 21 kg/cm^2 g for water cooled Applications-AAR standard

2. 27kg/cm^2 g for air cooled Applications-AAR standard

OR

3. Design Pressure-+17^{0}C above wet Bulb minimum for Water Cooled application –EN/ISO standard

4. +17^{0}C above Dry Bulb minimum for Air Cooled applications-EN/ISO standard

5. *Prefer Designing of both High and low stage Vessels for same pressures to Reduce Ammonia Charge. Ammonia can then be stored in L.P. vessel & there is no need to pump down refrigerant if one wants to stop the plant.*

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DESIGN PRECAUTIONS

1.Use Boiler Quality plates for Fabrication- IS2825/14430

IS2002Gr 2A or SA516/Gr.70 –SA517 Gr-60/70 plates, refer

AAR- 1 standard. Do not use Structural steel IS-2062

2. Use proper Thickness of plate as per TEMA standards

Consider joint efficiency as 0.7 if no radiography is done

Add corrosion allowance 0f 1.6mm in the calculated

thickness.

3. Fabrication as per IS/ISO-16528-1 or ASME sec. VIII-Div. 1

or EC/97/23

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PRESSURE VESSEL THICKNESS- CALCULATIONS

- MOC: SA 516 Gr. 70-Allowable stress 17500 psi(ASME CODE/BIS2825)
- Vessel Size say: 30" diameter x 12 ft long
- Calculations Based on ASME Sec VIII Div. 1
- $t = \frac{PxR}{(SxE - 0.6P)}$
- Where,
- t = minimum required thickness (inch.)
- P = Internal design pressure in (psi)
- R = inside Radius
- S= Allowable Stress for material
- E=Joint Efficiency
- E= 0.70 DP Test only
- E= 0.85 10% Radiography
- E= 1.0 100% Radiography
- Corrosion Allowance-1.6mm

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THICKNESS CALCULATIONS

ITEM	VALUE	UNIT	VALUE	UNIT
P	300	Psi	21.1	Kg/cm2
R	15	Inch	381	mm
S	17500	Psi	1230.4	Kg/cm2
E	0.7	%	0.7	%

t= 300-pressure x 15-I.R÷(17500-A.S.xE0.7-0.6x300)
t= 4500/(12250-180)12070= 0.37 inch or 9.47 mm
Add corrosion allowance of 1.6 mm
Thickness required is 11.07 mm
Use next Available size as 12mm thick plate

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RECOMMENDED THICKNESS FOR VARIOUS SIZES

Vessel Diameter-inch	Thickness-mm 516 Gr.70	Dish end Thickness-spun-mm
24	10	10
30	12	12
36	14	14
42	16	16
48	18	18

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WELDING PRACTICES

1. Root run with TIG(Tungsten inert Gas, Argon) or MIG (Metal inert Gas)for vessels
2. Welding by qualified welder certification level 3
3. Welding rods- Argon- High side-AWS-A5.18 or ER70S-G, Low side –AWS –A5.28 ER70S-G
4. Electric Welding – AWS –A5.1 E-7018. Do not use structural welding rods 6013
5. Remove valve internals & controls, strainers Solenoid coils etc. before welding

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OIL SEPARATORS

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FUNCTION OF OIL SEPARATOR

1. Ammonia & oil are immiscible
2. Oil is mainly required for compressor lubrication.
   The oil is used as a lubricant for the bearings,
   oil-pump, labyrinth seals, shaft seals, etc.,
3. Oil has no refrigeration cooling ability like
   refrigerant & hence oil anywhere else in the system
   is a contaminant
4. Maximum oil should therefore be removed after compressor
   discharge with oil separator
5. Installing efficient oil separator, even at higher cost, is recommended so that maximum oil is contained in the close loop of compressor and oil separator only & system remains free from excess oil

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REFRIGERANT FILM FOULING – REFRIGERANT SIDE FOULING

- Oil in the refrigerant causes resistance to heat transfer
- . Failure to control excessive oil build-up in a system occurs when the heat exchanger tubes become coated with oil, creating a thermal barrier.
- The heat transfer efficiency is reduced and drastically reduces the cooling effect & adversely affecting capacity and efficiency system .

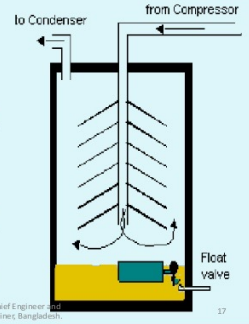
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OIL SEPARATOR

Oil Separator

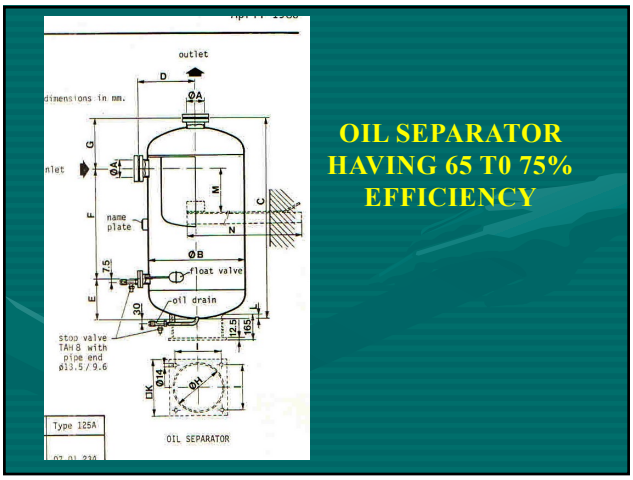
It is situated on the compressor discharge line.

The purpose of the oil seperator:
- To return oil entrained in the gas, back to the compressor sump.
The oil return may be float controlled as shown, electric solenoid controlled on a timer, or uncontrolled with a small bore capillary tube allowing continuous return.
With all of these methods a shut off valve is fitted between separator and compressor to allow for maintenance.
The oil gas mix enters the separator where it is made to change direction, the heavier oil droplets tend to fall to the bottom.

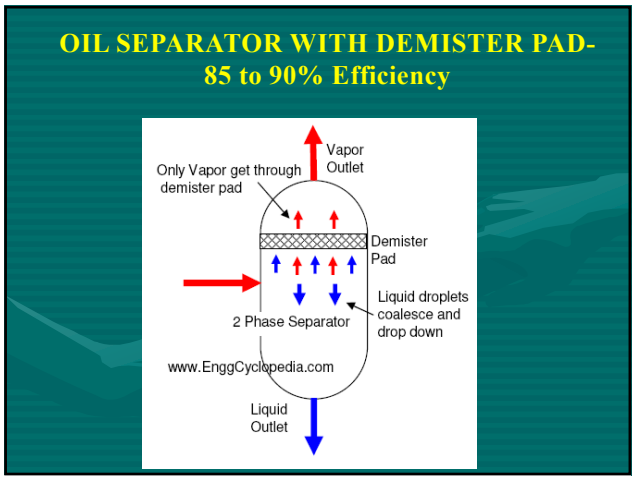


7/17/2014 Mohd. Hanif Dewan, Chief Engineer and Maritime Lecturer & Trainer, Bangladesh. 17

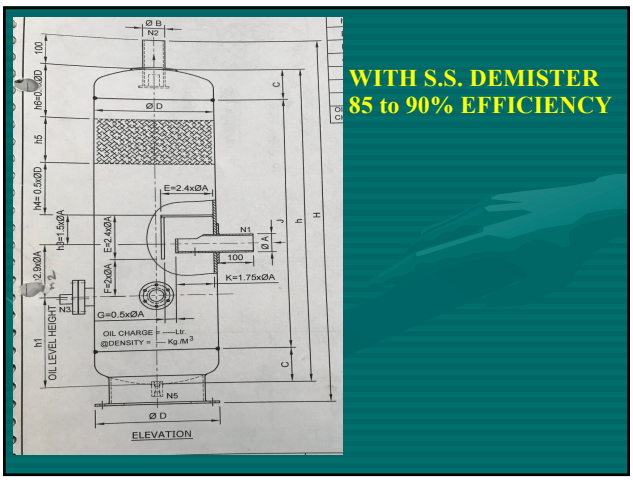
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OIL SEPARATOR
HAVING 65 T0 75%
EFFICIENCY

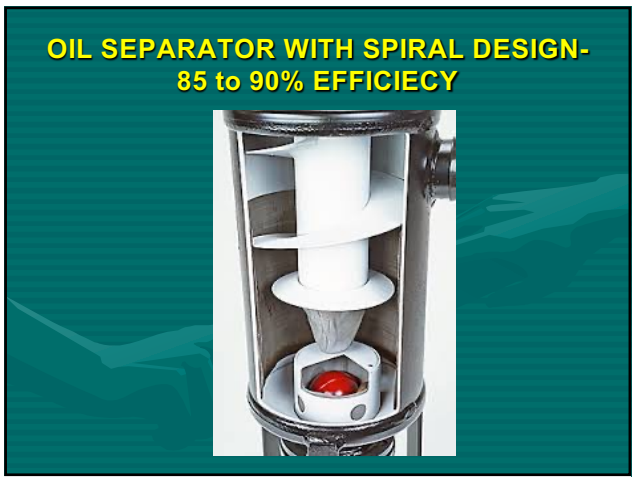
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OIL SEPARATOR WITH DEMISTER PAD-
85 to 90% Efficiency
Vapor
Outlet
Only Vapor get through
demister pad
Demister
Pad
Liquid droplets
coalesce and
drop down
2 Phase Separator
www.EnggCyclopedia.com
Liquid
Outlet

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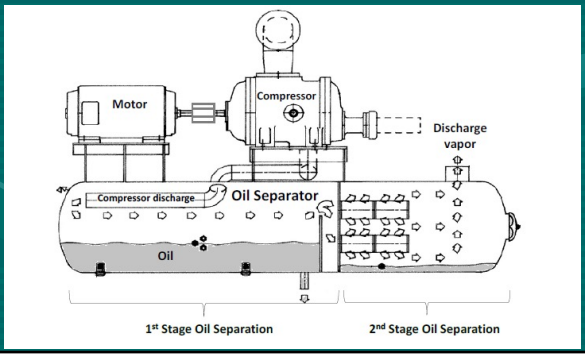

WITH S.S. DEMISTER
85 to 90% EFFICIENCY
ELEVATION

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OIL SEPARATOR WITH SPIRAL DESIGN-
85 to 90% EFFICIECY

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SCREW COMPRESSOR TWO STAGE OIL SEPARATOR
90 to 95% EFFICIENCY



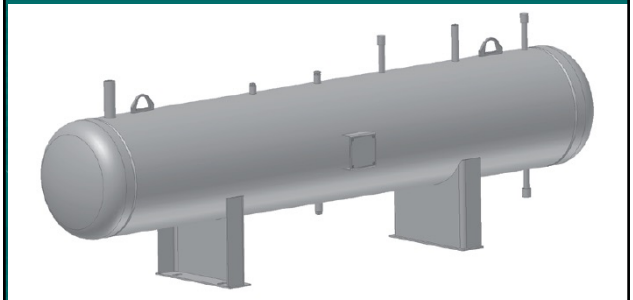
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OIL SEPARATOR SELECTION

Di min. =C x $\sqrt{\emptyset}$p
Di min.= minimum inside shell diameter in mm
C=constant-26.6 for single stage and H.P. of two stage
C-21.03 for Booster and two stage L.P.
Øp= compressor discharge volume flow rate in m3/h
Let us take 3-cylinder compressor with volume as 398CMH at 1000RPM & single stage selection
Di min. =26.6 x $\sqrt{398}$ =26.6x19.93=530mm
Select Type 100 with 600x1397x8mm shell Thickness
For Booster or two stage 3L.P. & 1H.P.=21.03x $\sqrt{398}$=419mm dia.
Select Type 80=419x1039x8mm shell Thickness

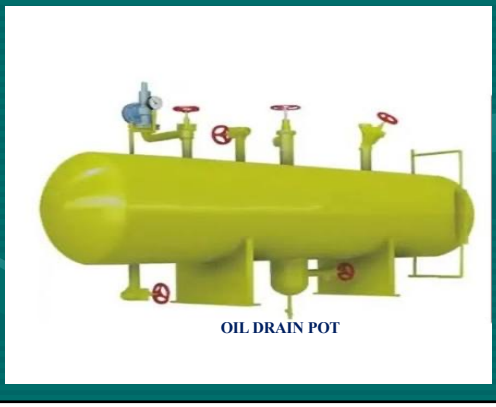
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HIGH PRESSURE AMMONIA LIQUID STORAGE RECEIVERS



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HIGH PRESSURE AMMONIA RECEIVER WITH OIL DRAIN POT



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HIGH PRESSURE AMMONIA LIQUID RECEIVER

1. Selection of receiver is done based on the assumption that it should be able to accommodate entire charge of refrigeration system.
2. This is essential as if, one wants to do some repairs, attend leakages, do some piping welding etc. one should be able to pump out ammonia from the entire refrigeration system in receiver, then close the king valve and then carry the modifications/repairs.
3. The receiver should be selected so that when ammonia is pumped down, the level of liquid should not be more than 80- 85%. This provides space for liquid to expand as per weather changes
4. Many times a standby receiver is provided as per standard requirement

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RECEIVER SELECTION

5. Receiver should maintain minimum level of 15 to 20% liquid when plant is operating at full load.

6. The refrigerant charge to be determined in the system

7. Charge comprises of 100% quantity of air coolers+30% in the gravity flooded accumulator,+ 30 to 35% in L.P. vessel if pump circulation system+ Interstage cooler +Liquid line quantity+ Wet return line quantity

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RECEIVER SELECTION FOR PUMP CIRCULATION SYSTEM- TYPICAL CASE –FTO1000, 3 BLAST FREEZERS,LP VESSEL 42"diax12ft LONG

Quantity in Intermediate vessel FTO 1000 model of liquid volume - 0.685cum x density at minus 5.7^0C is 647.6kg/cum= 443.5 kg of ammonia
Quantity in LP vessel 42" diax12 ft. long having internal volume as 105.4 and 50% liquid in the LP vessel therefore 105.4/2=52.7 ft^3 (1.488 cum) liquid volume at minus 40^0C having density of 690.2kg/m^3 =1.448x690.21=1028kg of ammonia
In blast freezers of 0.168cum eachx3=0.0.504x690.2=347.86 kg
Total charge for above equipment = 443.5+1028+347.86=1819.36 kg +20% in pipe lines=2183 kg or 4803 lbs.

TWO RECEIVERS EACH of 36" diameterx16ft long to ACCOMMODATING 2700 LBS. OF AMMONIA WHEN 80% FULL MAY BE SELECTED

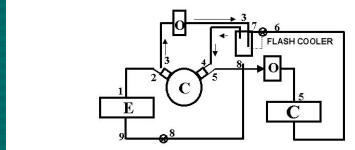
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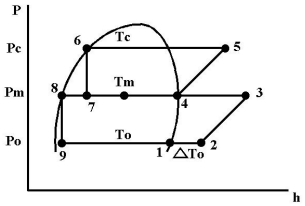
INTER STAGE COOLERS

- 1. These are used in two-stage refrigeration plants installed between low stage and high stage
- 2. The inter-stage cooler can be OPEN TYPE in which discharge refrigerant gas from low stage and liquid from high stage is mixed and the liquid mixture reaches intermediate pressure
- 3. Or it can be with CLOSED COIL in which liquid that is coming out of condenser is sub cooled by expanding small quantity liquid from the condenser and the superheat of gas from discharge of low stage is removed

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OPEN FLASH INTERSTAGE COOLING





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SELECTION OF OPEN INTER-STAGE COOLER

For Velocity=0.3m/s, the following equation exists:

D_i, min. in mm= **D_i, min. in mm= $11.2\sqrt{n\ x\ Zh}$**

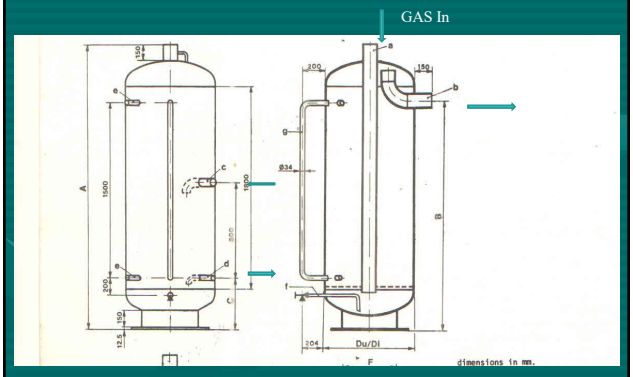
For example, if we take two No. KC42 compressors running at 965RPM then

Zh No. of High stage cylinders-would be 2+2=4

Di, min would be = $11.2\ \sqrt{965x4}$ = 695.84 mm
If we select FTO-800 with inner shell diameter of 786mm then actual velocity in the cooler at full load would be 0.3x (695.84/786)²= 0.235 m/s which is acceptable,

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OPEN INTER-STAGE COOLER



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OPEN FLASH INTERSTAGE COOLER

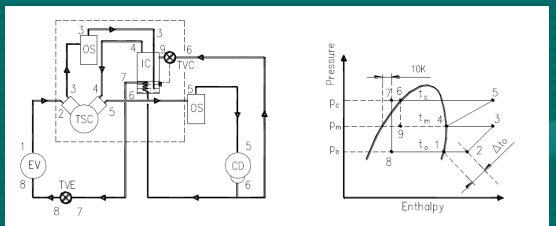
ADVANTAGES
1. MAXIMUM CAPACITY FOR GIVEN COMPRESSOR
2. MINIMUM POWER CONSUMPTION
3. MAXIMUM C.O.P
4. MINIMUM OPERATING COST PER YEAR

DRAWBACKS
1. INTERSTAGE COOLING SECTION EXPENSIVE
   REQUIRE- SHUT OFF VALVES/L.P. OIL SEPERATOR, LP DISCHARGE
   HP SUCTION STRAINER
2. QUALIFIED OPERATOR NECESSARY/AUTO CONTROL REQUIRED
3. LARGER SYSTEM REFRIGERANT QUANTITY
4. OIL TRAPING FROM LP DISCHARGE-LESS SUITABLE FOR R-22
5. LOW PRESSURE DIFFERENCE ACROSS EXPANSION VALVE
6. RISK OF FLASH GAS IN LIQUID LINE TO EVAPORATOR

APPLICATION
1. OPTIMIZE SYSTEM-MOST EFFICIENT
2. MEDIUM/LARGE PLANTS WHERE MINIMUM RUNNING COST
   IMPORTANT COMPARD TO INITIAL COST
3. COOLER IN ENGINE ROOM PREFERABLY ABOVE L.P. VESSEL
   WITH MINIMUM DISTANCE FOR LIQUID TO FLOW BY GRAVITY
4. AUTO CONTROL TO LOAD/UNLOAD CYLINDERS MUST & NOT MANUAL

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CLOSED INTERSTAGE FLASH COOLER-SYSTEM 'D'



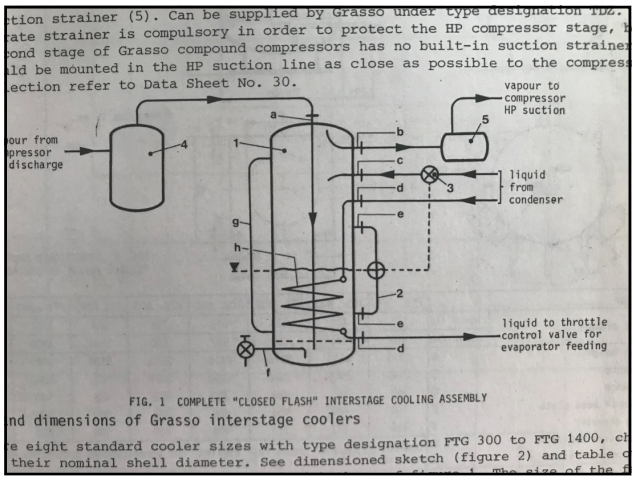
Refrigeration circuit.

LEGENDS :

TSC = Two Stage Compressor
IC = Interstage Cooler (closed flash type)
TVC = Throttle Valve for feeding interstage cooler
OS = Oil Separator
CD = Condenser
TVE = Throttle Valve for evaporator feeding
EV = Evaporator

Q_o = Compressor capacity
P_e = Compressor shaft power
n = Maximum rotational speed
t_c = Saturated condensing temperature
t_m = Saturated intermediate temperature
t_o = Saturated suction temperature
Δt_c = Liquid subcooling
Δt_o = LP Suction superheat
p_c = Saturated condensing pressure
p_m = Saturated intermediate pressure
p_o = Saturated suction pressure
h = Enthalpy.

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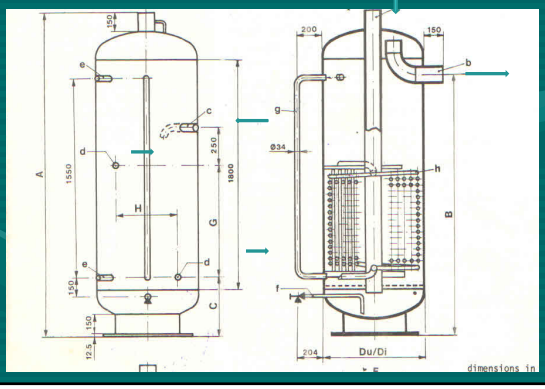
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CLOSED INTER-STAGE COOLER WITH COIL

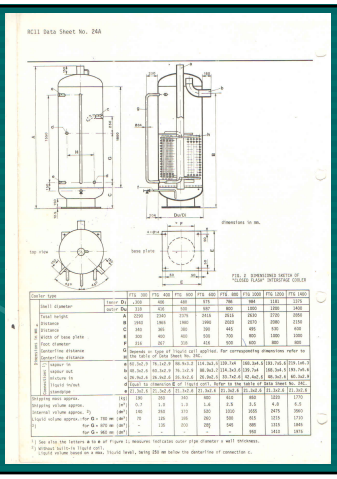
1. Closed inter-stage cooler for two stage systems is many times preferred alternative due to availability of high pressure subcooled liquid
2. The cooler can be mounted in the machine room and not required to be mounted at height since the pressure of liquid in the coil is high
3. Operation and loading unloading of compressor is much easier and chances of compressor H.P. cylinder failure are less
4. The only drawback is it is not as efficient as open inter-stage cooler system.

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CLOSED INTER-STAGE COOLER WITH SUB COOLING COIL



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CLOSED INTER-STAGE COOLER WITH SUB COOLING COIL- SPECIFICATIONS

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COIL FOR CLOSED INTER STAGE COOLER

For +40˚C and -25˚C operation with 10K Sub - Cooling - Intermediate temperature from KC42 chart would be – minus 3.5^{0}C with capacity of 162kW each, for two compressors, it would be 324kW
Qs (sub cooling)required would be 12.33kW or 12330W

LMTD between +40^{0}C and Tm 3.5^{0}Cwould be = 22.78 K

Coil area Fn-required would be =12330/ (3568 x 22.78) = 0.151m^2

Select coil VS1 having area as 0.75m^2 with diameter of 197mm

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CLOSED FLASH INTERSTAGE COOLER

ADVANTAGES

1. ALTERNATE TO OPEN FLASH COOLER EXCLUDING DISADVANTAGES
2. FULL PRESSURE DROP ACROSS EXPN. VALVE
3. NO RISK OF FLASH GAS IN LIQUID LINE TO EVAPORATOR

DRAWBACKS

1. CAPACITY SOME WHAT LOWER THAN OPEN FLASH COOLER(3-5%)
2. SPECIFIC POWER CONSUMPTION SOMEWHAT HIGHER
3. COMPRESSOR PRICE PER UNIT CAPACITY –HIGHER
4. INTERSTAGE COOLING MORE EXPENSIVE THAN OPEN FLASH COOLER DUE TO BUILT-IN COIL
5. ALL DRAWBACKS OF OPEN FLASH COOLER LIKE EXTRA LP OIL SEPERATOR

APPLICATION

FOR MEDIUM TO LARGE SIZE AMMONIA INSTALLATIONS

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LOW PRESSURE/LOW TEMPERATURE

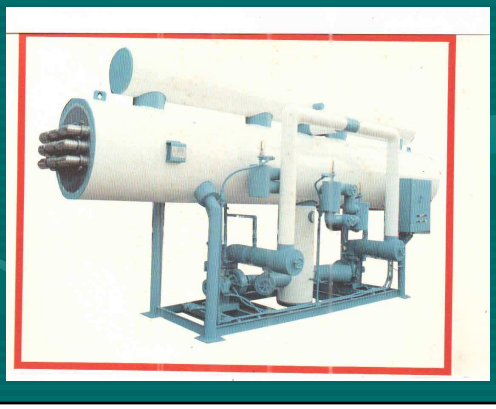
AMMONIA STORAGE (L.P.) VESSEL FOR

FORCED FEED PUMP CIRCULATION SYSTEMS

(there is separate module-5 for pump circulation systems)

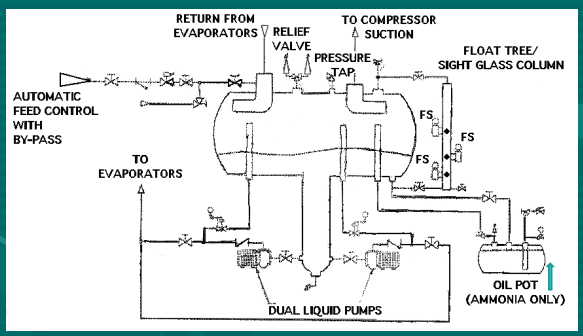
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PUMP CIRCULATION LP PACKAGE



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AMMONIA PUMP CIRCULATION ARRANGEMENT



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L.P. VESSEL SIZING

1. L.P. vessel to be sized for entire liquid volume of all air coolers/freezers+ Wet return line and liquid supply line quantity+20% as minimum level in the L.P. vessel when operating at full load.
2. The L.P. vessel should also have space known as surge volume for maximum level and minimum level for ballast volume

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SURGE VOLUME:

The volume above working liquid level provided in the vessel, is known as surge volume and serves the purpose of accommodating liquid that might be forced out of evaporators during defrosting of one or more evaporators.

Another aspect needs to be considered is the liquid in the wet return line from evaporators to L.P. vessel, which may drain in the L.P. vessel if power shut down takes place and the liquid refrigerant pump becomes inoperative.

In normal circumstances the amount of liquid + vapour returning is same as pump circulation feed to evaporators, but when pump is not working, the extra liquid quantity gets added to vessel from the wet suction line and vessel design needs to take this quantity to account in addition to defrost liquid quantity.

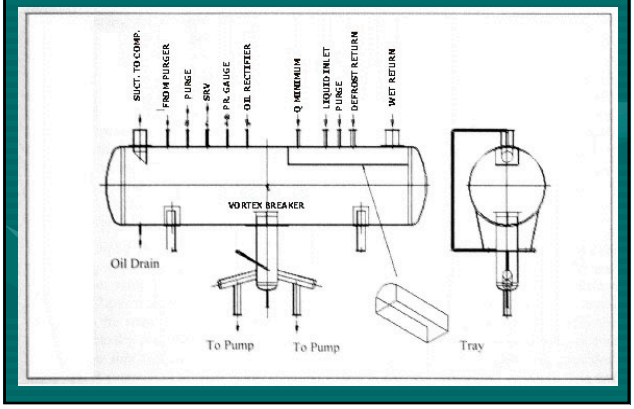
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BALLAST VOLUME:

1. The other important level on the lower side of the operating liquid level is the liquid required either during start up after a pump down cycle or if additional evaporators are taken on line for operation. During this period the liquid drawn from the vessel is at a higher rate than it is returning to L.P. vessel.
2. The alarm indication for this minimum level should be provided making the operator aware about the falling liquid level.
3. The ballast volume is generally calculated for 5 minutes period meaning pump flow rate multiplied by 5.

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LOW PRESSURE VESSEL DETAILS



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LOW PRESSURE VESSEL SIZING

Capacity in TR @ °F Evaporating Temperature							
Size-diameter x L	Single stage-Temperature	Two Stage-Temperature					
	0	-10	-20	-30	-40	-50	-60
20x10	24	20	20	18	12	11	11
24x10	49	46	37	30	29	25	23
30x10	120	100	95	78	72	58	50
36x12	212	182	174	149	125	118	100
42x12	330	293	275	280	202	177	165
48x12	466	415	396	341	300	252	225
54x12	624	549	528	463	400	312	300
60x12	774	675	583	575	481	425	376

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SURGE DRUM ON SHELL & TUBE FLOODED CHILLER

DIFFERENCE BETWEEN SURGE DRUM & ACCUMULATOR

SURGE DRUM HAS ONLY GAS WHEREAS

ACCUMULATOR HAS LIQUID+GAS

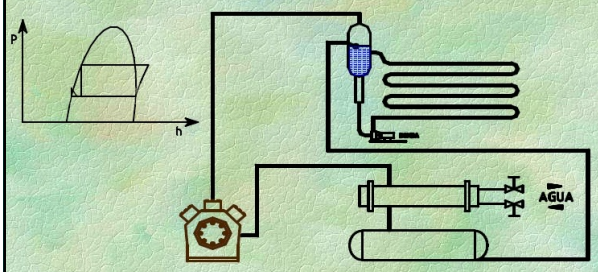
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IMPORTANT CONSIDERATIONS

1. Surge drum is provided to prevent liquid entering the suction line and thereafter in the compressor.
2. The nozzles connections should be selected to maintain very low velocity of vapours likely to enter the suction line
3. Observe the photos and see how many connections have been provided between chiller and surge drum to lower the vapour velocity
4. There is no need to unnecessarily provide large diameter surge drum if proper care is taken in designing the suction velocities-around 0.5 to 0.7m/s

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Flow Control Methods Recirculated Systems



ACCUMULATOTR SELECTION FOR AIR COOLER

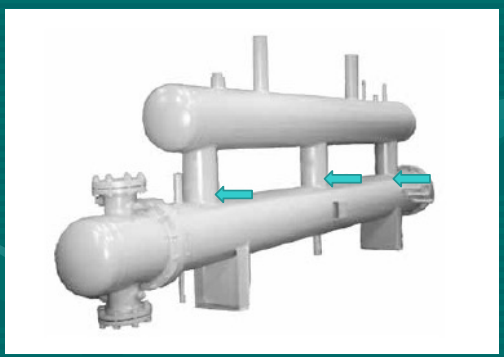
50

SURGE DRUM ON CHILLER



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SURGE DRUM ON CHILLER



Arrows show more than one connection to reduce pull velocity

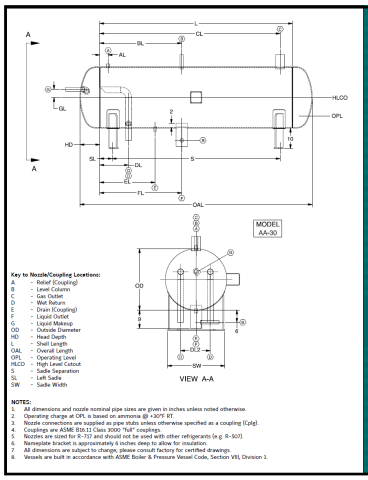
52

SURGE DRUM SELECTION

- Diameter is selected based on 75 to 100 fpm velocity, to Ensure Smaller Liquid Droplets subjected to More Gravity Pull Than Compressor Suction Gas Pull/Drag Lift Velocity
- (Ref ASHRAE 2006 Refrigeration handbook & W.F. Stocker Hand Book)
- 12" diameter x 6ft long mounted directly on Chiller

With 3 Connecting pipes of 6" diameter

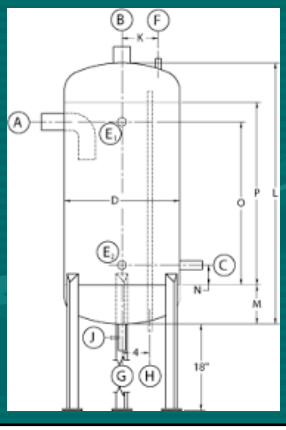
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HORIZONTAL SURGE DRUM

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VERTICAL SURGE DRUM



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ACCUMULATOR FOR GRAVITY FLOODED CHILLERS/AIR COOLERS

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ACCUMULATOR SELECTION

The design, selection and installation of accumulator has remained till date a nightmare for system designers as there are no clear cut guide lines and evaporator or many air cooler manufacturers indicate their inability to suggest proper size and method for installing the same and indicate that this is a system designer's responsibility.

The accumulator in the flooded system has to serve three functions

1. To separate liquid from vapour coming from expansion device
2. To absorb changes in the volume due to load fluctuations and due to sudden variation in suction pressures.
3. To ensure liquid mist or fine droplets are not carried back to the compressor

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ACCUMULATOR SIZING

3.There are two forces acting on the liquid droplet-one is gravity pull and other is compressor suction pull.

4. Many times, if the accumulator design is done on the basis of calculation of droplet size and separation velocity the diameter calculated works out to be quite small, but one must also consider the liquid volume stored in the air cooler which is likely to enter the accumulator if the suction pressure drops suddenly due to load fluctuations. It is therefore essential to accommodate entire internal volume of liquid in the accumulator design over the normal liquid level and hence the diameter or height has to be increased accordingly.

5. Because the static head of the liquid leg results in a higher pressure at the bottom of the leg, the evaporating temperature is higher at the lower section of the evaporator, which reduces heat transfer rate. In low temperature evaporators it becomes more crucial to keep the static head at a minimum.

6. Size the line sizes entering and leaving the accumulatorfor initial load & not for steady load.

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Height of liquid in the Accumulator

The important issue and common confusion exists as to what should be level at which accumulator should be installed with relation to cooling coil & liquid level in the accumulator.

ASHRAE volume on Refrigeration states "The maximum liquid height in the separator is a function of the type of system in which the separator is being used & evaporating temperature. In some systems this can be estimated, but in others, **previous experience** is the only guide for selecting proper height..

In gravity liquid/vapour separators use k value as 0.03 in the formula $v = k(\sqrt{\rho_l - \rho_v / \rho_v})$ where

V= velocity of vapour in m/s

ρ_l = density of liquid in kg/m^3

ρ_v = density of vapour in kg/m^3

K is the factor based on experience

This value of k=.03 gives velocities equivalent to those used for 300mm to 400 mm vertical separation distance for R-717.

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Level of liquid w.r.t. coil top row

Because the static head of the liquid leg results in a higher pressure at the bottom of the leg, the evaporating temperature is higher at the lower section of the evaporator, which reduces heat transfer rate. In low temperature evaporators it becomes more crucial to keep the static head as minimum as possible.

The compromise that emerges is that the Δp that motivates the flow is also proportional to the difference in the elevation between the liquid in the accumulator and that of the bottom of the tubes.

Associated with this compromise is how much elevation of the accumulator above the coil should be provided. A survey of some coil manufacturers revealed that typical distances from the bottom of the accumulator to the top coil to be between 6 to 10 inch.

Physical restrictions force other arrangements, such as placing the accumulator above the roof in order to conserve headroom in the refrigerated space.

One coil manufacturer bases its design on circulating twice as much liquid as evaporates and on the coil pressure drop of approx. 1.4 kPa(0.2 psi) and arrives at a height from the liquid level in the accumulator to the top of the coil of approx. 0.46m (1.5 ft.)

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Level of accumulator w.r.t. coil top

The sizing of outlet connection of accumulator is very important

Size it for the initial cooling load and not for the operating load

Whenever you have an issue of liquid not entering the coil, it means the opposite

Aways worry why the gas/vapour is not coming out in enough velocity and quantity. If the vapours come out, the liquid would enter automatically by gravity

Bruce Griffiths has stated –Always worry why gas is not coming out instead of worrying why liquid is not entering

Most important therefore is to size outlet connection of coil large enough to take care of initial load and vigorous boiling generating large low density vapours

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EXPERIENCE

In one of the installations in Mumbai. We faced a problem of liquid coming to the compressor through the accumulator,. The liquid instead of entering the evaporator coil and getting out was not happening , instead liquid was rising through the accumulator stand pipe and entering the compressor, in spite of level switches, We changed many things

1. Bigger accumulator
2. Raising the level of accumulator
3. Changing the level switches, however the problem remained.
4. The pressure drop through the coil and its inlet outlet connections were not proper
5. As soon as the plant started, the liquid would come to the compressor
6. Finally we decided to lowe the temperature slowly. As soon as the liquid would come to compressor we would stop the plant , the temperature drop achieved was very little say 05 to 1 deg. c
7. We continued to start and stop the plant and in each step a little temperature drop of 1 to 2 deg c was reached
8. 8After doing this for 15 5to 20 times and when the room temperature could come down to -15 deg C then the plant strted behaving normally and we could reach -20deg C thereafter.
9. I contacted Mr. Bruse Griffiths the well known expert and the next slide show his comments and suggestions.

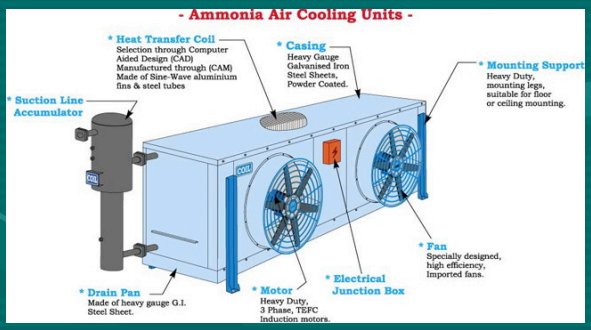
62

SUCTION –LIQUID LINE SIZING-GRAVITY FLOODED ACCUMULATORS

- When we select cooler based on capacity –especially for **low temperature applications**, The initial cool down load is very high when we start cooling form ambient temperature. There is very vigorous boiling and if line sizes are not designed for this load, the liquid instead entering the air cooler, it directly goes to accumulator and then onwards to the compressor in gravity flooded system and one has to stop the plant.
- Always design line sizes of inlet/outlet of air cooler for initial load situation
- In case of problem always think why suction gas is not getting released from air cooler rather than worrying about why liquid is not entering coolers-Liquid will enter automatically by gravity from the accumulator once the space is created by gas going out-Valuable advice by Bruce Griffiths

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ACCUMULATOR FOR AIR COOLER



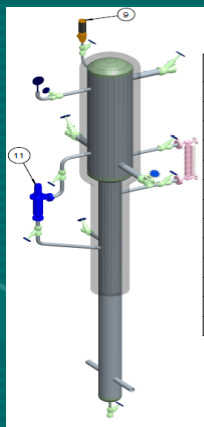
64

ACCUMULATOR FOR GRAVITY FLOODED SYSTEMS



ACCUMULATOR FOR PHE

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ITEM No.	DESCRIPTION	QTY.	REMARKS
1	STRAIGHT SHUT OFF VALVE	ALL VALVES AND ACCUMULATOR ACCESSORIES ARE AVAILABLE ON SITE ON SITE	50 NB
2	STRAIGHT SHUT OFF VALVE		50 NB
3	STRAIGHT SHUT OFF VALVE		25 NB
4	STRAIGHT SHUT OFF VALVE		15 NB
5	STRAIGHT SHUT OFF VALVE		25 NB
6	STRAIGHT SHUT OFF VALVE		25 NB
7	STRAIGHT SHUT OFF VALVE		15 NB
8	STRAIGHT SHUT OFF VALVE		15 NB
9	PRESSURE RELIEF VALVE		15 NB
10	PRESSURE GAUGE		15 NB
11	FLOAT VALVE		25 NB
12	LIQUID LEVEL SWITCH		25 NB
13	PUFF INSULATION		60mm THICK PUFF (UNIFORM THICK ALL AROUND)

CUSTOMER	TITLE	SCALE	DRN BY
	ACCUMULATOR	NTS	P.JANGRA
CRITICAL SYMBOLS	THIRD ANGLE PROJECTION	JOB NO.	DRG.NO.	DATE
CRITICAL SAFETY			2/2	18.11.10
CRITICAL FUNCTION				

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ACCUMULATOR FOR ICE BANK SYSTEM USED IN DAIRIES & FOR BLOCK CAN ICE PRODUCTION



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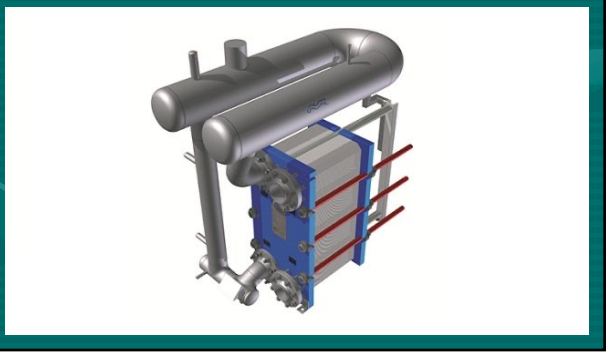
ACCUMULATOR SIZING –VERTICAL-40^{0}C condensing Temperature

Evaporating Temperature-^{0}C	Capacity-kW	Volume Flow rate-m3/hr.	Vessel Dia.-mm	Velocity – m/hr./fpm
0	105	104.84	350	1094.24/59.60
	176	175.74	510	860.72/47.05
	246	245.63	610	840.91/45.97
	404	403.4	760	889.69/48.64
-20	67	147.3	350	1531.78/83.84
	140	307.9	510	1507.99/82.44
	183	402.4	610	1377.62/75.31
	323	710.3	760	1377.62/75.31
-40	42	148.38	350	1543.01/84.35
	88	310.05	510	1518.52/83.01
	105	405.28	610	1387.48/75.85
	158	715.33	760	1577.65/86.24

Average velocity 75fpmfor low temperature/ For high temperature 45 fpm

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'U' TURN ACCUMULATOR



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KNOCK OUT DRUM/SUCTION TRAP



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KNOCK OUT DRUM DESIGN BASICS

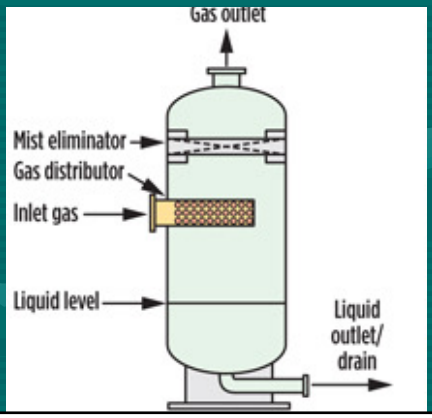
1. Surge drum volume is determined by the maximum potential liquid loading and the amount of fill time desired from the high level alarm(LAH) and high level shut down(LSH).

2. Compressor K.O. Pots are recommended to have minimum 5 minute fill time. The distance between maximum liquid level & the inlet pipe is recommended to be minimum 1ft. To minimize splashing and liquid surge.

3. The distance required from disengaging the liquid from gas is determined by the particle size and terminal velocity.
'K' value of 0.15 to 0.17 at a disengagement distance of 24" provides adequate separation for basic gravity separation that do not have mist pads Mist eliminators are however recommended for all K.O. drums. With mist eliminators the K value can be no more than 0.35 to control liquid carry over.

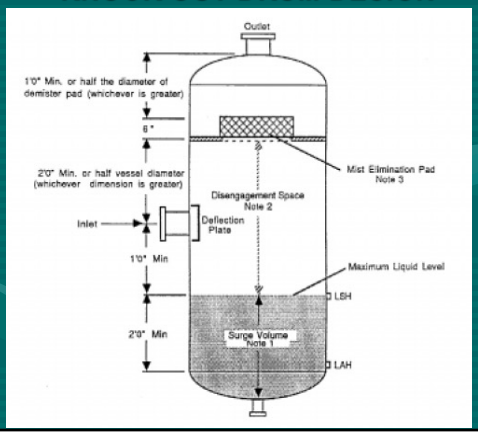
71

KNOCK OUT DRUM-ARRANGEMENT



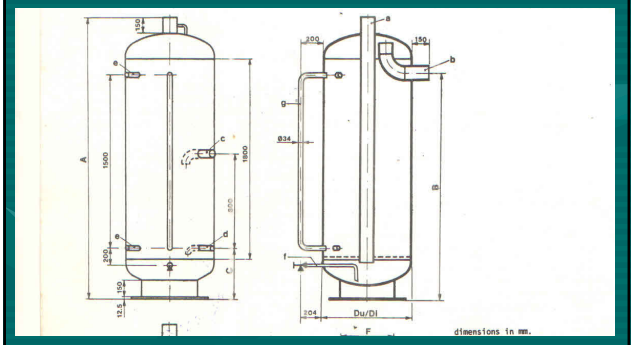
72

KNOCK OUT DRUM DESIGN



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KNOCK OUT DRUM/LIQUID SEPARATOR



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OIL POT

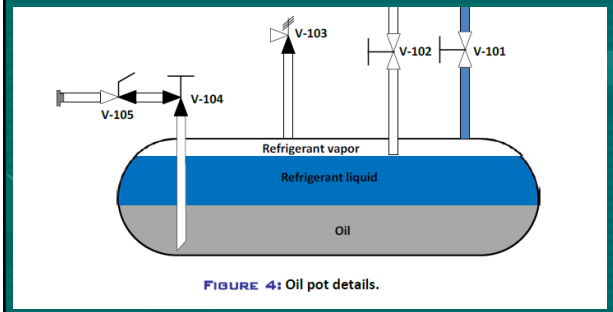


FIGURE 4: Oil pot details.

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VARIOUS LEVELS IN OIL POT

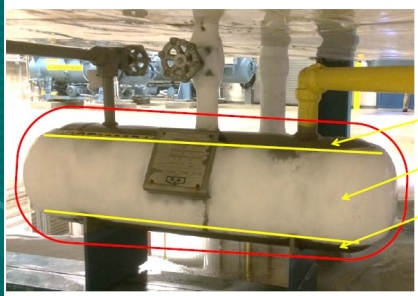
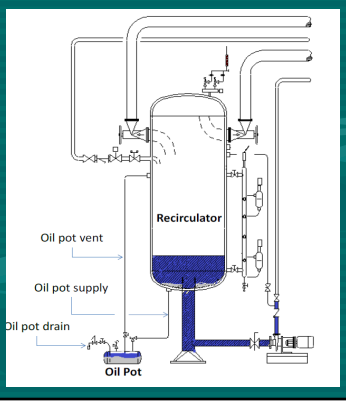


FIGURE 5: Various levels shown on an actual oil pot installed on a recirculator.

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OIL POT INSTALLATION



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OIL POT



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SHELL & TUBE CONDENSER SELECTION CALCULATIONS

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Shell & Tube Condenser



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SHELL AND TUBE CONDENSER



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EQUATION-1

$Q = m \times c_p \times \Delta T$

1. Q=heat transfer rate- Btu/hr
2. m=Mass flow rate lb/hr
3. Cp =Specific heat –Btu/lb^{0}F
4. ΔT=temperature difference between inlet and outlet-^{0}F

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EQUATION-2

Q=U x A x LMTD

1. Q=Btu/hr
2. 'U'- overall heat transfer coefficient- Btu/hr.sq.ft.^{0}F
3. 'A'-heat transfer area in ft^2
4. LMTD- Log Mean Temperature Difference- ^{0}F

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$

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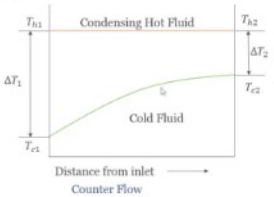
- **Q=U x A x LTD**
- LTD is the difference in temperature between the saturated refrigerant temperature and the leaving water temperature.
- The temperature difference between the refrigerant and the fluid is the driving force to overcome heat transfer resistance.
-

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CONDENSER-LMTD

Log Mean Temperature Difference for "Condenser" (LMTD)

LMTD or $\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$



Parallel Flow

Where,

$\Delta T_1 = T_{h1} - T_{c1}$

$\Delta T_2 = T_{h2} - T_{c2}$

Counter Flow

Where,

$\Delta T_1 = T_{h1} - T_{c2}$

$\Delta T_2 = T_{h2} - T_{c1}$

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Condenser LMTD

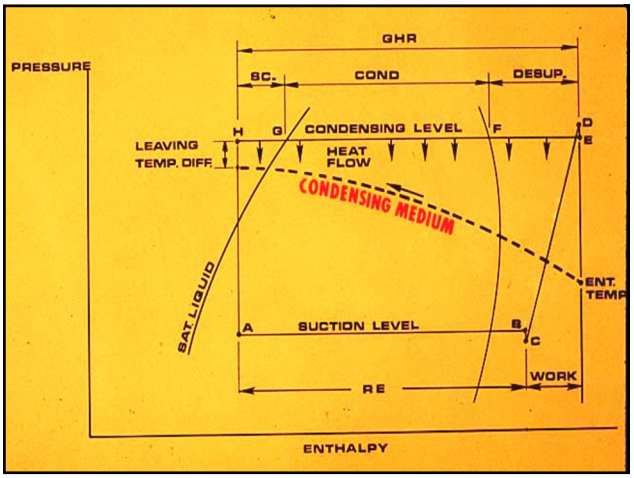
- Since the condensing temperature is constant, the LMTD is between condensing temperature say 40^0C-Water inlet temperature 30^0C, **ΔT1**=10
- Water outlet temperature 35^0C, **ΔT2**=5
- LMTD would be

$$LMTD = \frac{10-5}{\ln(10/5)}$$

$$LMTD = \frac{5}{0.693}$$

$$LMTD \approx 7.2^{\circ}C$$

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CONDENSER SIZING

Compressor	Conditions	Capacity-kW	Power consumed kW	Heat Rejection(Ratio)
KCX2	+40/+5C	100.74	20.72	121.46/100.74=(1.2)
KCX72	+40/-40C	101.51	67.32	168.83/101.51= (1.66)

As we design plants for lower evaporating temperatures, heat rejection and thereby required condenser capacity/size increases and condenser selection therefore needs careful attention.

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- **Resistance to heat transfer consists of four components related to the overall heat transfer coefficient (U):**

1) Refrigerant film resistance

2) Tube wall metal thickness resistance

3) Fouling deposits resistance

4) Fluid/water film resistance at 10 FPS velocity

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FOULING

- Tube fouling/Waterside fouling can be classified into the following categories:
- 1) Particulate
- 2) Precipitation
- 3) Corrosion
- 4) Biofouling
- 5) Chemical-induced corrosion

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- **CONDENSER TUBE FOULING**

1. In HVAC applications, fouling of the chiller/ condenser tubes substantially impacts the power consumption of the compressor.
2. This is why the focal point of any water treatment programme should be the prevention of deposition.
3. A layer of scale that is 1/64 of an inch in thickness on the condenser tubes can increase electrical usage in a chiller by as much as 33%.
4. Biofilms can decrease condenser heat exchange efficiency to an even greater degree.

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- **REFRIGERANT FILM FOULING – REFRIGERANT SIDE FOULING**

- Oil in the refrigerant causes resistance to heat transfer
- . Failure to control excessive oil build-up in a system occurs when the heat exchanger tubes become coated with oil, creating a thermal barrier.
- The heat transfer efficiency is reduced and drastically reduces the cooling effect & adversely affecting capacity and efficiency system .

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- It is of paramount importance to ensure that heat exchange surfaces are maintained in a clean, deposit-free condition.
- Fouling occurs because cooling water contains minerals such as Calcium and Magnesium that precipitate to form deposits on heat transfer surfaces. Cooling water systems are also commonly plagued by biological growth that forms slime (biofilm) or algae on heat transfer areas. Additional foulants include mud, silt, sand, corrosion products and petroleum products, which are ingested into the cooling tower. A 1,000-ton cooling tower will ingest over 3,000 pounds of solids in one year. Some of these solids will inevitably make their way to the condenser tubes. All of the foulants, mentioned above, will reduce the heat transfer efficiency of even the best-designed heat exchangers.

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CONDENSER TUBE AND SHEET FOULED WITH BIOFILM, SCALE, CORROSION, MUD AND SAND



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SHELL & TUBE CONDENSER SELECTION- CALCULATIONS

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CONDENSER SELCETION

Basis of Design

1. Refrigeration capacity-100TR
2. Cooling Water inlet temperature -30^{0}C(86^{0}F)
3. Cooling Water out let Temperature-35^{0}C(95^{0}F)
4. Condensing temperature-40^{0}C(104^{0}F)
5. Evaporating Temperature –Minus 5^{0}C(23^{0}F)

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CONDENSER SELCETION

Step-1-Calculate the Total heat rejection
=Evaporator Tons100x12000Btu/hr+ compressor 128Bhpx2545
=1,52,5760Btu/hr

Step-2-Calculate Water Quantity Required

Total heat rejection
=60x8.33xGPMx(Water leaving Temperature-Water entering temperature)
=500xGPMxΔT
1,52,5760=500xGPM(95-86)

GPM of water required=339GPM

1USGPM=8.33lb/m

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CONDENSER SELCETION

1. **Step-3**-Determine Mean Temperature Difference- LMTD

2. Use table for simplicity-ΔT1=104-86=18
ΔT2=104-95=9

3. LMTD=12.99

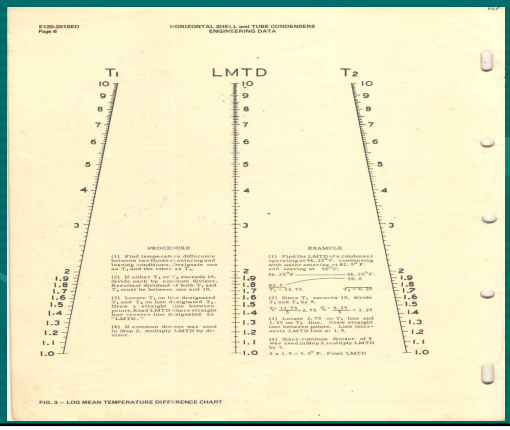
4. Assume Tube per pass=30Nos.

5. GPM/Tube/Pass=339GPM÷30TUBES=11.3

6. **Step-4**- Find Velocity of water by multiplying by
21.8=11.3x21.8=246.34fpm (Allow-250-300fpm. Mx.)
7. 21.8 is velocity factor for 1.06" ID tubes

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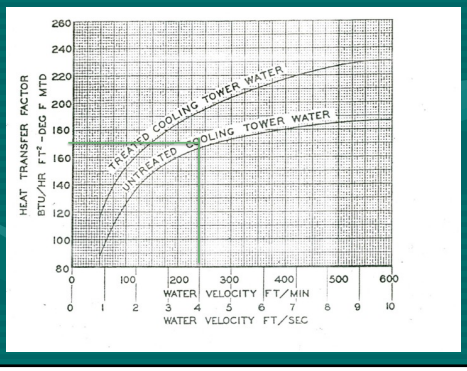
LMTD READY RECKONER



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CONDENSER SELECTION

Heat Transfer Factor



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CONDENSER SELCETION

1. From Graph find out heat transfer coefficient at 250fpm water velocity & untreated water
2. The heat transfer coefficient is- 172 Btu/hr.sq.ft.^{0}F
3. Use Formula-Q-heat rejection='U' x Ax LMTD(12.99)
4. Area Required 'A" =1525760÷172x12.99
5. . Area required=688Sq.ft.
6. Add 10% safety factor for fouling etc. 'A'=757Sq.ft.
7. Select condenser size as 24inch diameterx15ft long having 174 tubes, with area as 855Sq.ft. with 6 pass (29tubes per pass)from the standard sizes available from the catalogue

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Condenser Selection

- Calculation of No. of Passes
- D_{od}1.25-0.19thick=1.06 D_i -inch ID
- 3.14x$(D_i)^2$xfpm÷4x144=
- GPM/tube per passx8.33 lb./gpm÷62.4 lb./Ft^3

$$\text{No. of passes} = \frac{\pi \times (1.06)^2 \times \text{fpm} \times 62.4}{4 \times 144 \times 8.33 \times \left(\frac{\text{gpm}}{\text{tubes}}\right)}$$

- No. of passes = $\frac{\Pi \text{ x1.06x1.06xfpmx62.4}}{\text{4x144x8.33 x (339/174)}}$
- No. of passes-6
- Pr. Drop Water Side-24.4 ft. (0.72kg/cm^2)

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• CHILLER SELECTION

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SURGE DRUM ON CHILLER



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Brines And Their Applications

- The secondary coolants used in Refrigeration systems are known as brines. Below 0^0C water cannot be used and hence brines are used for lower temperatures.

Brine	Temperature range	Application
Sodium Chloride-NaCl	0^0C to Minus 15^0C	Block ice plants, Dairies
Calcium Chloride-CaCl2	-10^0C to-45^0C	Cold storages, Freezers
Ethylene Glycol	-50^0C to -10^0C	Industrial chillers, spiral Freezers
Proplylen Glycol	-50^0C to -10^0C	Food and beverages since it is non toxic

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Brine Heat Transfer Rates

Following Table would give some idea about the Heat Transfer Factors compared at 10fps(3.048m/s) velocity in tubes at average brine temperature of 20^0F(-6.7^{0}C)

Brine	Heat transfer Factor
1. Propylene Glycol	1.000
2. R-11	1.457
3. Ethylene Glycol	1.486
4. Methanol	1.726
5. Sodium Chloride	2.002
6. Calcium Chloride	2.056
7. Methylene Chloride	2.154

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Pumping Energy for Various Brines

The following table would also give idea about relative pumping energy required at 20^0F(-6.7^{0}C).

1. Methanol	1.000
2. Propylene Glycol	1.063
3. Ethylene Glycol	1.147
4. Sodium Chloride	1.192
5. Calcium Chloride	1.330
6. Methylene Chloride	3.425
7. R-11	4.613

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Flooded Ammonia Chiller for Ethylene Glycol Design

- Basis of Design
- 1. Capacity-100TR(351.7kW)
- 2. Brine inlet temperature-0^0C
- 3. Birne outlet temperature-Minus 5^0C
- 4. Average Brine temperature-Minus 2.5^0C
- 5. Brine concentration-30%(Viscosity-μ-2.5Cp)
- 6. Brine Freezing point-Minus15^0C
- 6. Brine Specific heat-3.8kJ/kg.k
- 7. Brine Density-1030kg/m3

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Chiller Selection (Flooded S&T)

- **Capacity-100TR(Brine used-Ethylene Glycol)**
- **Brine Entering Temperature-+32^{0}F(0^{0}C)**
- **Brine Leaving Temperature- +23^{0}F(-5^{0}C)**
- **Average Brine Temperature-+27.5^{0}F(-2.5^{0}C)**
- **Evaporating Temperature-+14^{0}F(-10^{0}C)**
- **Brine Temperature Range- (32-23)=9^{0}F**
- **Leaving Split-(23-14)=9^{0}F**
- **Flow Rate- 220usgpm @400fpm**
- **Specific heat=0.9**
- **Specific gravity- 1.03 @ 30% concentration**

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Brine Chiller Selection

1. Calculate Brine flow rate=Q=m x Cp x ΔT
2. 351.7kW=kg/sx3.8kJ/kg.Kx5
3. m==18.5/1030kg/m3 density
4. =0.018m3/s=65m3/hr
5. Ammonia evaporating temperature-Minus10^{0}C
6. Ammonia latent heat at -10^{0}C=1050kJ/kg
7. Ammonia mass flow rate required-351.7/1050 =0.306kg/s=1100kg/hr

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Brine Chiller Selection

8. Ammonia boiling heat transfer coefficient-1000W/m^2.K
9. ΔT1=10, ΔT2=5. LMTD=7.2^{0}C
10. Area required from equation=Q=U x A x LMTD
11Area=351700÷1000x7.2=48.8m2-say
12. 50sq.m=538sq.ft.
13 Selection 24" diameter x12 ft long 683Sq.ft. area,
14. 174 no of tubes of 1.25inch dia. 4 pass selection

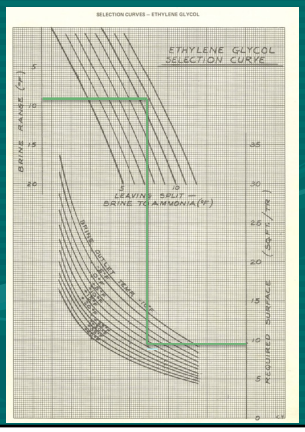
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Brine Chiller Selection Formula

- **Standard formula (FPS units)**
- $Q = 500 \times GPM \times C_p \times SG \times \Delta T$
- **:**Q= Heat load (**BTU/hr**)
- **GPM** = Flow rate (gallons per minute)
- C_p= Specific heat (BTU/lb·°F)
- **SG** = Specific gravity (dimensionless)
- ΔT= Temperature difference (°F)

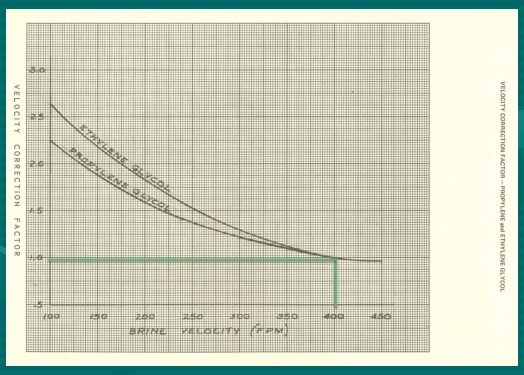
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CHILLER SELECTION



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CHILLER SELECTION



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CHILLER SELECTION- Capacity-83.28 Ton-Tube dia. ¾”

- Area required based on 9^0F range and 9^0F split is 9.5sq.ft/Ton
- 83.28 x 9.5 =791.16 sq.ft.
- ¾” tube external area 0.1964sq.ft/ft.
- For 791.16 sq.ft ÷0.1964 = 4028 rft or 403 Tubes 10 ft long
- Selection-24” Dia x 10 ft long,452 tubes of ¾” having 888 sq.ft.x12 pass,419 fpm

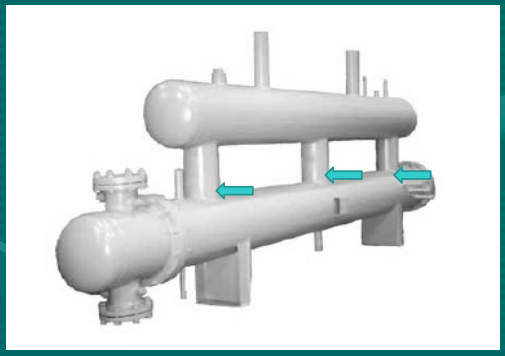
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SURGE DRUM SELECTION

- Diameter is selected based on 75 to 100 fpm velocity, to Ensure Smaller Liquid Droplets subjected to More Gravity Pull than Compressor Suction Gas Pull/Drag Lift Velocity
- (Ref ASHRAE 2006 Refrigeration handbook & W.F. Stocker Hand Book)
- 12” diameter x 6ft long mounted directly on Chiller With 3 Connecting pipes of 6” diameter

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SURGE DRUM ON CHILLER



Arrows show more than one connection to reduce pull velocity

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50 year Distinguish Service Award

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