TOTAL

TEP/DP/EXP/SUR

PROCESS ENGINEERING DESIGN MANUAL

INDEX

Revision : 0

Date : 2/85

Page No : 0

MUHAMMED SALIM SHAIKH.

1. DESIGN CONDITIONS

1. APPLICABILITY

The following design criteria are applicable for both feasibility studies and pre-project studies.

2. PRESSURES

- The design pressure of a vessel shall be taken as of the following ;

Operating pressure barg	Design pressure barg
0 - 10	MOP + 1 bar
10 - 50	MOP + 10 %
50 - 100	MOP + 5 barg
> 100	MOP + 5 %

Note : Pmin = 3.5 barg

MOP = Maximum Process Operating Pressure

- Vessels subject to vacuum during operation shall be designed for the maximum external operating pressure plus a margin of 0.15 bar.

  If the internal pressure is 0.35 bara or less the vessel will be designed for full vacuum.

- Design pressure for pump discharges shall be calculated by taking 120 % of the normal pump Δ P when operating at design conditions.

3.0 DESIGN TEMPERATURES

- Design vessel temperatures shall be as follows :

  Maximum design temperature = max. operating Temp + 15 °C
  Minimum design temperature = min. operating Temp - 5 °C or minimum ambient temperature.

- Consideration for the minimum design temperature must take into account any depressurisation of the vessel that may occur during emergency or shut down situations. (See section on flaring).

4.0 MATERIAL OF CONSTRUCTION

- Details of the required material of construction for various temperatures are given in Table 1.
- Details on corrosion allowances and wall thickness are given in the vessel design section.

TABLE I - MATERIAL SELECTION AS A FUNCTION OF DESIGN TEMPERATURE

	- 102 °C TO - 196 °C	- 69 °C TO - 101 °C	- 47 °C TO - 68 °C	- 30 °C TO - 46 °C	+ 20 °C TO - 29 °C	+ 427 °C TO + 21 °C	MAX. 468 °C	MAX. 538 °C	MAX. 593 °C	ABOVE 593 °C
STEEL GRADE	SS 18/8	3.5 % Ni	2.25 % Ni	Killed Carbon steel	Carbon steel	Carbon steel	0.5 % Mo 1C-0.5Mo	1.125 Cr 1.125 CrO, 5 Mo	2.25Cr-1Mo 5Cr-0.5Mo	Stainless steel
ROLLED SHEET	A 240 304 A 240 304L A 240 316 A 240 316L A 240 321 A 240 347	A 203 Grade D or Grade E	A 203 Grade B Tc = - 60 °C for walls > 50 mm	A 516 Grade 55 Grade 60 Grade 65 Tc = - 40 °C Grade 65 for walls > 50 mm	A 516 Grade 55 Grade 60 Grade 65 Grade 70	A 515 Grade 55 Grade 60 Grade 65 Grade 70 or A 516	A 204 Grades A or B (0.5 Mo) A 387 Grade 12 Class I 1Cr-0.5Mo	A 387 Grade 11 Class I	A 387 Grade 22 Grade 5 Class I	Refractory Grade
	9 Ni									
	A 333									
TUBES	A 312 304 A 312 304L A 312 316 A 312 316L A 312 321 A 312 347	A 333 Grade 3	A 333 Grade 7	A 333 Grade I or Grade 6	A 106 Grade A or B	A 106 Grade A or B	A 335 - P1 (0.5 Mo) A 335 - P12 1Cr - 0.5Mo	A 335 - P11	A 335 - P22 2.25Cr-1Mo A 335 - P5 5Cr - 0.5Mo	
	9 Ni : A 333 Grade 8									

2. VESSELS

1. APPLICABILITY

Virtually all process schemes use phase separation of some description. The design and sizing of a separator with acceptable accuracy is required for both the feasibility and pre-project phases.

Consideration is given in this section to the specification of vertical and horizontal separators for vapour-liquid and vapour-liquid-liquid separation. Details are also given concerning vessel internals.

Separation of solids from gas or liquids is not covered in this design guide. Generally a vendor will be consulted for details of a proprietory designed vessel.

2. SEPARATOR APPLICATIONS AND CONSIDERATIONS

2.1. 2 PHASE SEPARATORS (usualy vertical unless stated)

- Compressor and Fuel Gas KO drums

  Efficient separation of liquid from vapour required. Always consider a mist eliminator. Provide sufficient surge time (1 to 2 minutes) between the HLL and tripping the compressor.

- Relief System KO drum - See section 9.0 Flare Systems

- Unit Feed KO drums

  Required upstream of acid gas absorbers, glycol contactors and dessicant bed dehydrators. Can be incorporated into base of tower for weight and space saving. Always use demister pads.

- Production separators (Vert or horizontal)

  Liquid separation from gas not as critical as compressor KO drum unless a compressor is located immediately downstream of separator. Always consider start-up, shut-down and process slugs when designing.

2.2. 3 PHASE SEPARATORS

- 3 phase production separators are generally horizontal. If good liquid-vapour de entrainment is required demisters are usually stated. Oil separation from the water phase must be sufficient so as not to overload water treatment units. Chemical additives (demulsifiers, anti-foam, pour point depressants) may be added to aid separation.

3. HORIZONTAL OR VERTICAL DESIGN

- Provided sufficient L/D ratio is selected a horizontal separator is more efficient than a vertical for the same flow area.
- Vapour velocity in a horizontal drum can exceed the liquid settling velocity provided L/D > 1. For vertical drums the velocity cannot.
- Horizontal drums are more effective and geometrically more practical for a heavy liquid phase removal than vertical drums.
- A rising liquid level in a vertical drum does not alter the vapour flow area. Consequently vertical drums are preferred for compressor and fuel gas KO drums.
- Vertical drums utilise a smaller plot area and are easier to instrument with alarms and shutdown controls. For floating installations are preferred as less "sloshing" occurs.
- For high volume flowrates a split flow horizontal drum is preferable as smaller drum diameters can be used. The preferred split flow arrangement is a single centre entry nozzle with two end exists. Head exits can be used where plot space is limited.
- Each design case must be evaluated separately but in general the following can be used as a guideline :

- Vertical drums	Compressor KO drums	Degassing boots
	Fuel gas KO drums	Absorber feed KO drums
	Floating installations	
- Horizontal drums	Production separators HP	Reflux drums
	3-phase separation	Flare KO drums

- Try to avoid vessels with wall thickness greater than 100 mm as these require special fabrication and can prove expensive.

4. CALCULATION THEORY AND EQUATIONS (for use in calculation sheets)

(Valid only for pure gravity settlers with no internals to enhance separation)

4.1. LIQUID-VAPOUR SETTLING VELOCITY

① $$Vs = K\left[\frac{\rho_L - \rho_V}{\rho_V}\right]^{1/2}$$

② $$K = 0.003615\left(\frac{D}{C}\right)^{1/2}$$

③ $$\underline{\underline{Vs = 0.003615\left(\frac{D}{C}\right)^{1/2}\left[\frac{\rho_L - \rho_V}{\rho_V}\right]^{1/2}}}$$

$\rho_{L,V}$ - liquid or vapour density kg/m^3
Vs - settling velocity m/s
K = correlating parameter m/s
D - particle diameter -microns
C - drag coefficient
μ - vapour viscosity - centipoise
m/s

- For medium and low pressure with gases of viscosity less than 0.01 cp Figure 1 can be used to estimate Vs.

- For higher pressures (> 50 bar) or viscosities in excess of 0.01 cp it is necessary to calculate Vs. The drag coefficient C is calculated using Figure 2 (curve 2) where :

$$CR_e{}^2 = \frac{1.3072 \times 10^{-11} . \rho_V . D^3 . (\rho_L - \rho_V)}{\mu^2}$$

Equation 3 is then used to calculate Vs.

4.2. LIQUID-LIQUID SETTLING VELOCITY

(based on Stokes law of terminal settling)

The following equation can be used for calculating the settling velocity of water in oil or the upward "settling" of oil in water. The important fact is to use the viscosity of the continuous phase i.e : for oil settling upwards through water use the water viscosity, for water settling in oil use the oil viscosity.

$$Ut = \frac{g.D^2(\rho_H - \rho_L)}{18.\mu_c}$$

U_t =	terminal velocity	m/s
g =	gravitation accel	m/s2
ρH =	density heavy fluid	kg/m3
ρL =	density light fluid	kg/m3
μc =	viscosity (continuous)	kg/m.s

Setting the particle size to 125 microns and using more useful units gives :

⑤ $$Ut = 0.513 \left(\frac{\rho_H - \rho_L}{\mu_c}\right)$$

U_t mm/min
μ_c in centipoise

- The above equation is valid for REYNOLDS no of 0.1 - 0.3
- If calculated settling velocity is > 250 mm/min use 250 max

4.3. VESSEL VOLUMES

- Partial volumes of a horizontal cylinder can be calculated using the partial volume charts in Figure 3 or estimated using the following equations :
(for vessels with a diameter < 1.2 m ignore head volumes)

(see page 2.13 for sketch)

$$A_L = \frac{D^2}{4}\cos^{-1}\left[\frac{D-2h}{D}\right] - \left(\frac{D}{2} - h\right)\left(Dh - h^2\right)^{0.5} \quad m^2$$

in radians

HORIZONTAL CYLINDER $Vc = A_L.L \quad m^3$

2 DISHED HEAD $Vdh = 0.21543\ h^2\ (1.5\ D - h)\ m^3$

2 ELLIPTICAL HEADS $Vel = 0.52194\ h^2\ (1.5\ D - h)\ m^3$ (most common)

2 HEMISPHERICAL HEADS $Vhh = 1.047\ h^2\ (1.5\ D - h)\ m^3$ (gives extra vol)

.. VOLUME UPTO BAFFLE
for depth h
(elliptical heads)

$$= \frac{0.52194\ h^2\ (1.5D-h)}{2} + AL.B$$

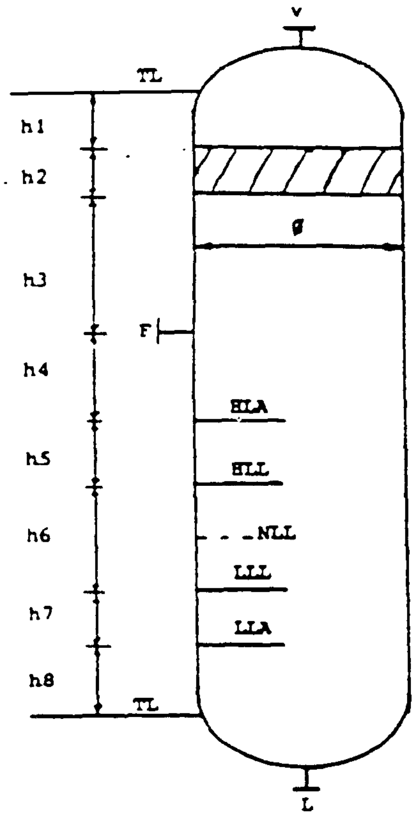
- These formula are accurate enough for general design and are easily programmed on to a calculator for time saving.
- More accurate formula are available, see ref list, but are often too complicated to be useful for multiple calculations.
- For greater accuracy the length L should be the tan-tan length and not the flowpath length between nozzles. This is especially true with large vessels and a tight design.

4.4. CALCULATION PROCEDURE VERTICAL VESSEL (vapour-liquid separation)

A guide for filling in the attached calculation sheet.

- Decide if Figure 1 can be applied i.e $P < 50$ bara, $\mu < 0.01$ cp
- If applicable use the 500 micron curve to evaluate settling velocity (this assumes a mist eliminator will be installed) or 150 micron with no mist eliminator. It is recommended to install a mist eliminator for most applications. If not calculate Vs using equ 3.
- Derate the calculated settling velocity by 85 % design margin to give a maximum allowable vapour velocity.
- Calculate drum internal diameter and round to nearest 50 mm. (note further adjustment of ID : OD can be made to suit standard head dimensions).
- Check if wall thickness is less than 100 mm (See 4.8).

- calculate vessel height based on following criteria :



h1 - max (15 % of Ø or 400 mm)

h2 - 100 mm if mesh selected
150 mm for compressor KO

h3 - max (50 % of Ø or 600 mm)

If no mesh use h1 + h2 + h3 = 60 % Ø or 800 mm

h4 - 400 mm + d/2 : d = inlet nozzle Ø

h5 - calculate based on 1-2 minutes residence time at maximum liquid inflow - min 200 mm

h6 - base on following hold up times : (min 350)

- reflux drums	4 min	
- product drums	5 min	with pump
	3 min	no pump
- heater feed	8 min	
- HP sep. to LP sep.	4 min	

h7 - 1-2 min residence time - minimum 150 mm

h8 - 150 mm for bottom connected LC
300 mm for side connected LC

Note : For compressor suction drums that are normally dry set HLL at 450 mm above tan line and use bottom connected LC. This will reduce vessel height if required. No specific HLL-LLL hold up time required.

4.5. CALCULATION PROCEDURE HORIZONTAL VESSEL (Vapour-liquid)

A guide on how to fill in the attached calculation sheet.

1. Calculate settling velocity Vs for partical size 150μ , use Fig. 1 or equ. 3.
2. Derate this by F = 0.85 and calculate required vapour velocity V m/s

   Vm = F x Vs x (L/D) m/s use L/D of 3 to 4 max (3 initial est)
3. Evaluate required vapour cross sectional area, Av
4. Assume drum is 70 % full i.e h/D = .7 and evaluate drum Ø to give required Av (to nearest 50 mm). For "dry" vessels use h/D = .35
5. For required liquid surge volume, calculate vol at HLL, if insufficient adjust D or L (note if L/D changes significantly recheck Av using new Vm).
6. Set position of LLL in drum and confirm required surge vol between HLL-LLL. If volume is insufficient increase Ø, L or h. Include volumes in heads.
7. When setting LLL height take into account any LSLL, LSL alarms and vortex breakers which may set minimum value usable. Usually 300-350 mm.
8. Rationalise all heights and dimensions to nearest 10 mm.

NOTES :

- For high volumetric flows of gas with small liquid volumes consider using split flow arrangement. Design is as above but with half vapour volume flow.
- Normal design is with top entry, exit nozzles. However if space is limiting (primarily offshore) head mounted nozzles can be used to increase flowpath.
- L is designated as the flow path length i.e distance between inlet and outlet nozzle. L' is the tangent-tangent lengh. For 1st estimates L' = L + 1.5 Øi + 1.5 $Ø_2$

  Øi = inlet nozzle diameter Ø2 = outlet nozzle diameter
- "Normal" liquid levels are taken as midway between the high and low levels.

4.6. CALCULATION PROCEDURE HORIZONTAL VESSEL 3 PHASE (See Figure 4)

- Sufficient residence time to allow separation of the oil-water mixture as well as the oil surge and vapour flow areas must be provided.

1. Proceed with steps 1 to 4 as for a two phase separation. Use L/D = 3 (1st estimate) and evaluate L.
2. Provision now has to be made to accomodate both oil and water surge volumes. Use Tan-Tan length L' and not nozz-nozz distance L.
3. Calculate LLL required to give approx 4 mins oil surge capacity (minimum). Inspection will reveal whether sufficient height exists below LLL to include the interface levels. If not, adjust the vessel Øor L to give sufficient room.

   Note : If the water cut is very small, consideration may be given to using a water boot instead of a baffle arrangement see step 10.

4. Having determined HLL and LLL now set both position and height of baffle. Calculate terminal settling velocity of water droplet (equ 5 sect 4.2) at both HLL and LLL. Volumetric flow of liquid is in both cases the oil plus the water. Calculate fall distance of a droplet across length of the drum. Baffle height and position can now be set noting :

   - the baffle should be at least 75 mm below the LLL
   - the baffle should be at least 2/3 down the length of the drum from the inlet
   - in some cases the water droplets will settle to the floor in a short distance. The baffle should still be set at a minimum of 2/3 along the vessel.

5. Set the HIL at baffle height - 75 mm. The LIL according to height determined by vortex breaker + LSLL use a minimum of 300-350 mm.
6. Check if an oil droplet will rise through the water layer (from drum floor) to LIL before reaching water outlet. Use area at LIL with normal oil + water flowrates. (This criteria is very rarely governing but must be checked).
7. Calculate water surge time Vol HIL - Vol@ LIL, and residence time Vol NIL - Vol.@ outlet. Remember to use only one head volume, and length of drum upto baffle. Minimum acceptable times are 4-5 mins. If calculated times are very long consider using a water boot arrangement.
8. Rationalise all dimensions and "tidy" levels to standard values if possible i.e : 150 mm, 200, 250, 300 etc. This allows use of standard displacers.
9. Recalculate all residence times based on "tidied" levels (if required).

   Note : In calculating the final residence times make sure that the vessel tan-tan length is used and not the nozzle to nozzle distance L.

10. Boot calculation (See Fig. 5)

- If the water volumetric flow is so small as to not warrant a separate baffled settling compartement as detailed above a water boot should be used instead. To design proceed as follows :

1. Proceed as previous upto step 3.
2. Calculate settling distance of water droplet when vessel is operating at LLL. Water droplet should reach floor of drum before oil outlet. Remember that the oil exit nozzle will be raised above the floor as a standpipe. Adjust drum Ø or L to achieve settling.
3. Check that settling is also possible when operating at HLL, droplet to fall below drawoff nozzle level.
4. Size water drawoff boot Ø (try to use standard pipe diameters). Calculate rising velocity of the oil in water, set downward velocity of water in boot at 90 % of this and evaluate boot Ø. Boot length by inspection (use standard displacers).

Note : Boot Ø must be less than 35 % of vessel Ø
When heavy walled vessels are used a remote boot may be more economical to prevent large cuts in the main vessel.

4.7. NOZZLE SIZING (see section 10.0 also)

Inlet nozzle

- Size based on normal volumetric flow + 10 % (liquid + vapour flow)
- Limit inlet velocity to 7 - 13 m/s
- Round nozzle diameter up or down to nearest standard size

Gas outlet

- Size on normal flow
- Velocity limit 15-30 m/s
- Manholes : 450 mm or 600

Liquid outlet

- Normal flow + 10 %
- Velocity limit 1-3 m/s HC
  2-4 m/s water
- Min. diameter = 2" (avoid plugging)

4.8. VESSEL WALL THICKNESS

Calculate vessel wall thickness using the ASME VIII div. 1 formula. The wall thickness should be calculated immediatly after D is known to confirm if $t < 100$ mm.

$$t = \frac{PD}{2SE - 1.2P} + C$$

D = diameter mm
t = wall thickness mm
P = design pressure barg
E = joint efficiency
use 1 for seamless shells .85 otherwise
S = max. allowable stress bar
use 1220 bar for CS plate
1000 bar for SS plate

C = corrosion allowance mm
- use 3 mm unless stated
otherwise by EXP/TRT

for $t < 100$ mm : no fabrication problems
$100 < t < 150$ mm : vendor advice may be needed
$t > 150$ mm : Major fabrication problems

In order to meet standard vessel head sizes and wall thicknesses the following ranges should be observed :

Vessel diameter : 250 - 1250 mm in increments of 50 mm i.e. 250, 300, 350...
1300 - 4000 mm in increments of 100 mm i.e. 1300, 1400, 1500...

Standard wall thicknesses : 1 - 30 mm in increments of 1 mm i.e. 1, 2, 3, 4...
30 - 60 mm in increments of 2 mm i.e. 30, 32, 34, 36...
60 - 140 mm in increments of 5 mm i.e. 65, 70, 75, 80...

4.9. VESSEL WEIGHT

Vessel weights either horizontal or vertical can be estimated using Fig. 5. This figure is for the steel shell including manholes, nozzles, fittings etc but not the removable internals or support skid. The heads can be estimated by using weight of 2 heads = φ (m) x wall thickness (mm) x 20 kg.

5. VESSEL INTERNALS

5.1. MIST ELIMINATORS

- Mist eliminators or mesh pads are located under the vapour outlet nozzles of all compressor suction drums and fuel gas KO drums. For production separators it is always good practice to install an exist mesh pad.

- For dirty or and high viscosity liquids the efficiency falls to approx. 75 %. Consult vendor for futher data.

- Mesh is usually made from 304 SS. YORK DATA as follows :

Types of pad :	York n°	kg/m3	Thickness mm	Residual* entrainment PPM
General purpose	431	144	100	1.0 - 1.2
High efficiency	421	192	100	.55 - .61
	326	115	100	.17 - 0.19
Dirty service	931	80	150	1.6 - 1.8
	644		150	.8 - .87

- The engineer should specify type, diameter and thickness of pad required on the vessel data sheet.

- For particle sizes of 5 microns or less use two pads spaced 300 mm apart eg : glycol contactor.

5.2. INLET INTERNALS

Inlet internals can be specified to aid feed distribution and promote vapour-liquid separation. Generally for pre-project stage details are not required.

5.3. LIQUID PHASE INTERNALS

- Vortex breakers should be detailed for each oil/condensate and produced water outlet where the oulet flow direction is vertical.

- Vendors will sometimes specify internal packs of tilted plates or baffles or other arrangements to promote phase separation.

- Sand jetting facilities should be provided for on services where there is a risk of silting or sediment build up in the vessel. Generally jetting facilities are not required on gas-condensate systems.

6.0 REFERENCES AND USEFUL LITERATURE

6.1. LUDWIG VOL I CHAPTER 4

6.2. PERRY CHAPTER 6

6.3. Program calculates partial volumes Pierre Koch
OGJ Dec. 3 1983

EQUIPMENT N° D 1234 DEGASSING BOOT

Operating data :

Pressure (operating)	bara	=	1.04				
Temperature (operating)	°C	=	34				
Gas MW		=	51.4	Liquid description :			CRUDE OIL
Gas flow rate Wg	kg/hr	=	7290	Liquid flow rate	kg/hr	=	109 900
Gas density T, P	kg/m3	=	2.1	Liquid density (T,P)	kg/m3	=	810
Actual volume flow Qg	m3/s	=	1	Actual volume flow	m3/min	=	2.23
				Particle size	microns	=	150

Mesh pad [✓ Yes] : . Estimate Vs using Figure 1 and 500 micron line

No : . If P < 50 bar and μ < 0.01 use Fig. 1 and 150 microns

. If P > 50 bar or μ > 0.01 use calculation for Vs

1. **Vapour-liquid settling velocity** : from Fig 1 Vs = 1.6 m/s

or calculated C = ; Vs = m/s

2. **Derating** % = 85 maximum velocity **Vm = 1.36 m/s**

3. **Actual volumetric gas flow** = 1 m3/s

Drum flow area = 0.74 m2

Calculated drum Ø = 1000 mm *

* Vapour area not governing *

SELECTED DIAMETER = 2500 mm Flow area = 4.9 m2

BASED ON LIQUID RETENTION

HLL-LLL

4. **Required liquid hold-up times** :

h5 :	HLA - HLL	=	2	min	=	4.4	m^3	=	900	mm height
h6 :	HLL - LLL	=	5	min	=	11.15	m^3	=	2250	mm
h7 :	LLL - LLA	=	2	min	=	4.4	m^3	=	900	mm

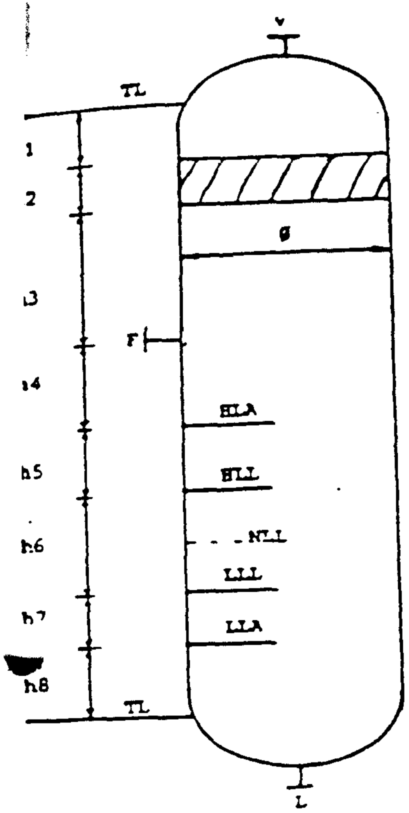
5. **Mesh pad** : [✓ Yes]/no thickness = 100 mm

Sheet 1 of 2

TOTAL TEP/DGP/DIP/EXP/SUR	PROCESS CALCULATION SHEET		
	VERTICAL VAPOUR-LIQUID SEPARATOR	ITEM : DEGASSING BOOT NO : D 1234	
BY | CHK	DATE | JOB TITLE : EXAMPLE	JOB N° : EXAMPLE	REV

. Height calculation

Ø = 2500 mm



h1 : 15 % of Ø or 400 mm (Use max) = 400 mm

h2 : mesh pad = 100 mm

h3 : 50 % of Ø or 600 mm = 600 mm

With mesh : h1 + h2 + h3 = 1100 mm

No mesh : h1 + h2 + h3 : 60 % Ø or 800 = mm

h4 : 400 mm + d/2 : d = inlet nozz Ø = 550 mm

h5 : From step 4 or 200 mm = 900 mm

h6 : From step 4 or 350 mm = 2250 mm

h7 : From step 4 or 150 mm = 900 mm

h8 : 150 mm for bottom LC
300 mm for side LC = 150 mm

For "dry" vessel

h6 + h7 + h8 = mm

TOTAL VESSEL HT TAN/TAN = 5850 mm

7. Wall thickness

- DESIGN PRESSURE P = 2·5 barg Diameter D = 2500 mm
- CORROSION ALLOWANCE C = 3 mm

Max stress :

S = 1220 bar CS

1000 bar SS S = 1220

Joint efficiency (.85) E = 0·85

$$t = \frac{P \times D}{2 \times S \times E - 1.2\,P} + C$$

= 6·8 mm

tmin = D/800 + C + 3 mm ⟶ = 9 mm

8. Vessel weight (Fig. 6)

t = 9 mm

L = 6·9 m (5·9+1)

D = 2·5 m

Shell weight = 7000 kg

Head weight = 450 kg

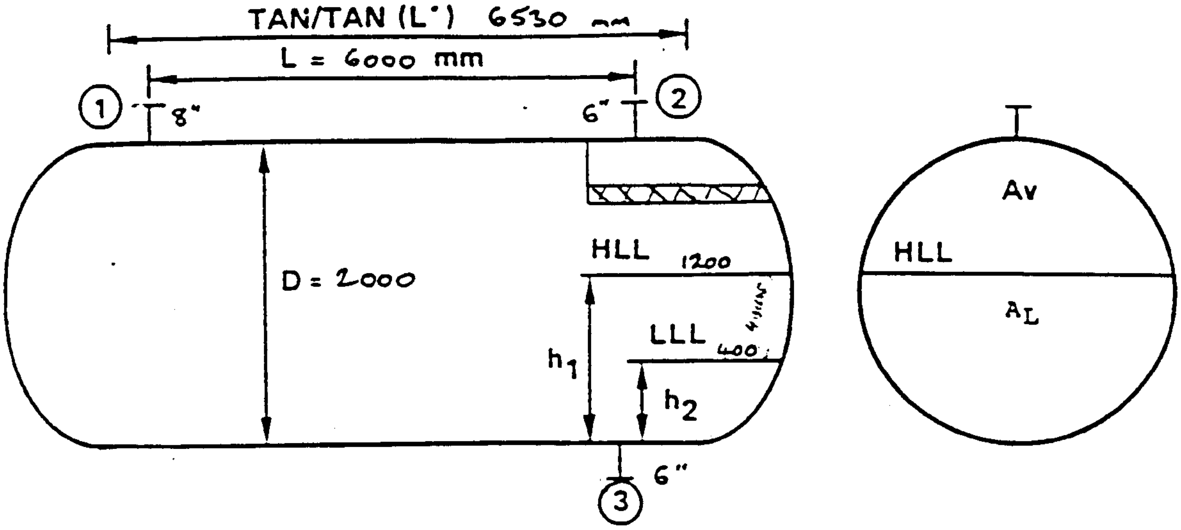
(t x D x 20)

TOTAL WEIGHT = 7500 kg

Sheet 2 of 2

TOTAL				PROCESS CALCULATION SHEET					
TEP/DOP/DIP/EXP/SUR				VERTICAL VAPOUR-LIQUID SEPARATOR			ITEM : DEGASSING BOOT NO : V 1234		
BY		CHK		DATE		JOB TITLE EXAMPLE	JOB N° : EXAMPLE	REV	

CALCULATION SHEET FOR HORIZONTAL 2 PHASE



Head type 2:1 elliptical/~~hemispherical~~ *

Indicate on sketch if demister mesh required

* Delete as applicable

EQUIPMENT N° : V 2010

DESCRIPTION :

1st STAGE SEPARATOR

Operating data :

Operating pressure bara = 20

Operating temperature °C = 18

Gas molecular wt		= 20·4	Liquid nature : CRUDE		
Wg gas flow rate	Kg/hr	= 17950	Liquid flowrate	Kg/hr	= 121650
Gas density T, P	Kg/m3	= 18·0	Liquid density T,P	Kg/m3	= 765·8
Qg actual vol flow	m^3/s	= 0·277	QL actual vol flow	m3/min	= 2·65
Gas viscosity	cp	= 0·01085	Particle size	microns	= 150

1. Vapour-liquid settling velocity : from Fig. 1 150 micron Vs = 0·23 m/s

   or calculated C = ; Vs = m/s

2. Max. vapour velocity $Vm = Vs \times F \times \frac{L}{D}$ Vm = 0·586 m/s

   L/D = 3

3. Actual vapour volumetric flow Qg = 0·277 m3/s $Av = \frac{Qg}{Vm}$ = 0·473 m2

TOTAL TEP/DOP/DIP/EXP/SUR	PROCESS CALCULATION SHEET	
	CALCULATION FOR HORIZONTAL 2 PHASE	ITEM : EXAMPLE NO :
	DATE	JOB N° : REV

4. Nozzle sizing vel limits : m/s inlet 7-13, Exit 15-30, liquid 1-3

Øi : Inlet flow = 0·321 m3/s Nozzle ID = 8 " Actual vel = 10·8 m/s
(+ 10 %) 0·35

Ø2 : Gas exit = 0·28 m3/s Nozzle ID = 6 " Actual vel = 15 m/s
Liquid outlet = 0·04 m3/s Nozzle ID = 6 " Actual vel = 2·1 m/s

1·5 × (Ø1 + Ø2) = 530 mm

5. Drum sizing

For trial 1 t_{res} = 4 mins vol required = 4 x QL = 10·6 m^3

		TRIAL	1	2	3	4
elected h/D			0·7	0·6		
Vapour area	Av	m2	0·473	1·174		
% Total area (Fig. 3)			0·25			
Total area	Ar	m2	1·89	3·142		
Liquid area	Al	m2	1·42	1·968		
Calculatd drum Ø		mm	1550			
Selected drum Ø	D	mm	1750	2000		
L/D (3 - 4)			3	3		
Flowpath length	L	mm	5250	6000		
a) Tan/Tan length	L'	mm	5780	6530		
HLL height		mm	1225	1200		
b) VOL @ HLL		m3	10·5	14·2	1·968 x 2 +	ud elliptical head
LLL height		mm	400	400		
Vol @ LLL		m3	2·36	3·13	[illegible] × 3·142 ×	6·53 + [illegible] volume
ΔVOL		m3	8·14	11·07		
Calculated tres		min	3·1	4·2		
NOTES :			too short try D = 2m	OK		

SELECTED DRUM : DIAMETER Ø 2000 mm x 6530 mm tan/tan

a) Tan/tan length L' = L + 1.5 x Øi + 1.5 Ø2 (ignore this correction if D < 1.2 m and use L for volume calcs. For trial 1 use L and ignore heads).

b) If VOL HLL is less than required surge increase D, L or h/D or reduce tres (by inspection).

PROCESS CALCULATION SHEET

TOTAL
TEP/DGP/DIP/EXP/SUR

CALCULATION FOR HORIZONTAL
2 PHASE

ITEM :
NO : EXAMPLE

BY | CHK | DATE | JOB TITLE : | JOB N° : | REV

6. Wall thickness

- DESIGN PRESSURE P = 20.9 barg Max stress CS = 1220 bar
- CORROSION ALLOWANCE C = 3 mm SS = 1000 bar

S = 1220

Joint efficiency E = 0.85

$$t = \frac{P \times D}{2SE - 1.2P} + C = 25 \text{ mm}$$

8. Vessel weight (Fig. 6)

t = 25 mm

L = 6.53 m

D = 2 m

Shell weight = 10 800 kg

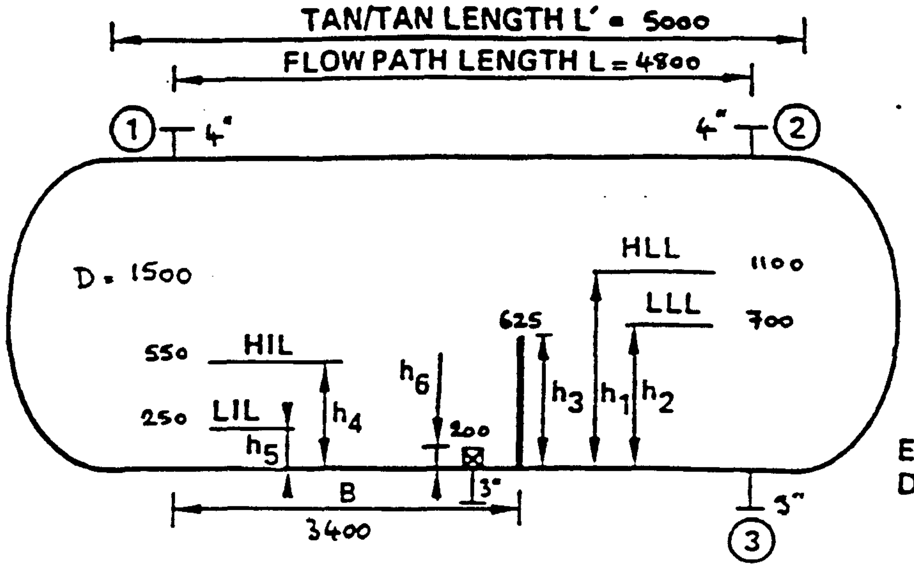
Head weight = 1000 kg

(t x D x 20)

TOTAL WEIGHT = 12 000 kg

NOTES:

TOTAL TEP/DOP/DIP/EXP/SUR	PROCESS CALCULATION SHEET	
	CALCULATION FOR HORIZONTAL 2 PHASE	ITEM : EXAMPLE NO :



- Amend sketch if boot required instead of baffle
- Indicate on sketch if mesh required
- Heads : 2 : 1 elliptical ~~/hemispherical~~

EQUIPEMENT No : D 5010
DESCRIPTION :
OFFSHORE TEST SEPARATOR

Operating data :

Operating pressure	bara	= 40	CONDENSATE		Flowrate	Kg/hr	= 31 000
Operating temperature	°C	= 50	ρc		Density T,P	Kg/m3	= 728.4
			QL		Vol flow T,P	m3/min	= 0.71
GAS MW			μc		Viscosity	cp	= 0.75
Wg flowrate	kg/hr	= 4509 = 125.85 m³/hr [illegible]					
Density @ T,P	Kg/m3	= 35.0	WATER CUT		Flowrate	Kg/hr	= 9955
Qg Vol flow	m3/h	= 129 = 0.036 m³/s	ρw		Density T,P	Kg/m3	= 988
μ		= 0.0103 cP	QW		Vol flow T,P	m3/min	= 0.168
Particle size	microns	= 150	μw		Viscosity	cp	= 0.54

1. Vapour-liquid settling velocity : from Fig. 1 — Vs = 0.175 m/s

   or calculated C = ; Vs = m/s

2. Maximum vapour velocity L/D = 3 — $Vm = Vs \times F \times \frac{L}{D}$ — Vm = 0.446 m/s

3. Liquid-liquid settling

Oil in water $U_t = 0.513 \left[\frac{\rho_w - \rho_c}{\mu_w}\right]$ mm/min ∴ $U_t$oil = 247 mm/min

Water in oil $U_t = 0.513 \left[\frac{\rho_w - \rho_c}{\mu_c}\right]$ mm/min ∴ $U_t$water = 177.5 mm/min

TOTAL TEP/DOP/DIP/EXP/SUR	PROCESS CALCULATION SHEET 1		
	CALCULATION FOR HORIZONTAL 3 PHASE	ITEM : EXAMPLE NO :	
BY CHK	DATE	JOB TITLE :	JOB N° : REV

4. Nozzle sizing : vel limits   m/s : inlet 7-13, Exit 15-30, liquid 1-3

	m^3/s :	nozzle id :	actual vel M/S
1. Inlet flow : (+ 10 %)	0.055	4" (100 mm)	6.8
2. Gas exit :	0.036	4"	4.4
3. HC outlet :	0.012	3" (75 mm)	2.6
4. Water outlet :	0.0028	3"	0.6

5. Vessel sizing

For trial 1 use hold up time oil (HLL-LLL) = 4 mins = 2.84 m^3
= 4 × Q_L

OIL SECTION

0.036/[illegible]

		TRIAL	1	2	3	4
Selected h/D			0.7	0.64		
Calculated (Qg/Vm)	Av	m2	0.08		0.38	
Av as % AT (Fig. 3)			25	32	21.5	
Total area	AT	m2	0.32	1.227	1.767	
Liquid area	AL	m2	0.24	0.834	1.387	
Calculated Ø		mm	650	↑		
Selected Ø	D	mm	1000	⇒ 1250	⇐ 1500	
L/D (3 - 4)			3	4	3.2	
Flowpath length	L	mm	3000	5000	4800	
a) Tan/Tan length	L'	mm	3250	5250	5060	
HLL height	h1	mm	600	800	1100	
VOL at HLL		m3	1.77	4.67	7.67	
LLL height	h2	mm		400	700	
VOL at LLL		m3		1.9	4.44	
△VOL		m3		2.8	3.23	
Calculated tres		min		3.9	4.54	
Notes or comments :			too small	tight LLL too low for water	ok Check H2O ⇒	

a) tan-tan length L' = L + 1.5 x (Ø$_1$ + Ø$_2$) mm - Ignore if D < 1.2 m

TOTAL
TEP/DOP/DIP/EXP/SUR

PROCESS CALCULATION SHEET 2

CALCULATION FOR HORIZONTAL
3 PHASE

ITEM : EXAMPLE
NO :

BY | CHK | DATE | JOB TITLE : | JOB N° : | REV

WATER SECTION

Trial 1 B = 2/3 x L = 3450 mm (rounded)

		TRIAL	1	2	3	4
Total liquid vol flowrate Qw + QL		m3/min	0.88			
Baffle distance	B	mm	3450	3400		
Liquid area at HLL	AL	m2	0.834	1.387		
Horizontal vel at HLL	V1	mm/min	1055	634		
Ut water (step 3)		mm/min	177.5	177.5		
Vertical fall from HLL = B x Ut/V		mm	580	952		
Final settled h = HLL - vert fall		mm	220	150		
Liquid area at LLL	AL	m2	0.447	0.825		
Horizontal vel at LLL	V2	mm/min	2110	1067		
U_t water (step 3)		mm/min	177.5	177.5		
Vert fall from LLL = B x Ut/V2		mm	290	565		
Settled baffle height	h3	mm	400	625		
Selected HIL level (adjust h3 and B if necessary)	h4	mm	350	550		
Check oil rise :						
Horizontal vel at LLL	V2	mm/min	2110	1067		
Ut oil (step 3)		mm/min	247	247		
Vert rise avec dist B = B x Ut/V2 = max outlet height		mm	400	787		
h5 selected LIL level		mm	250	250		
h6 selected outlet height		mm	200	200		
q1 water vol at HIL (upto baffle)		m3	1.01	2.13		
q2 water vol at LIL (upto baffle)		m3	0.62	0.69		
q3 water vol at NIL (upto baffle)		m3	0.81	1.36		
q4 water vol at outlet (")		m3	0.45	0.49		
q surge = vol (q1 - q2)		m3	0.39	1.44		
surge time q/Qw		mins	2.3	8.5		
residence time q3-q4/QW		mins	2.14	5.17		
calculated oil residence time (upto baffle) Vol (NLL - NIL)/Q_L		mins	1.8	4 mins		
			too small	OK.		

TOTAL

TEP/DOP/DIP/EXP/SUR

PROCESS CALCULATION SHEET 3

CALCULATION FOR HORIZONTAL 3 PHASE

ITEM : EXAMPLE
NO :

BY | CHK | DATE | JOB TITLE : | JOB N° : | REV

6. Wall thickness

- DESIGN PRESSURE P = 42·9 barg Max stress CS = 1220 bar
- CORROSION ALLOWANCE C = 3 mm SS = 1000 bar

S = 1220

Joint efficiency E = .85

$$t = \frac{P \times D}{2SE - 1.2P} + C = 35 \text{ mm}$$

8. Vessel weight (Fig. 6)

t = 35 mm Shell weight = ~~10800~~ 9000 kg

L′ = 5 m Head weight = 1050 kg

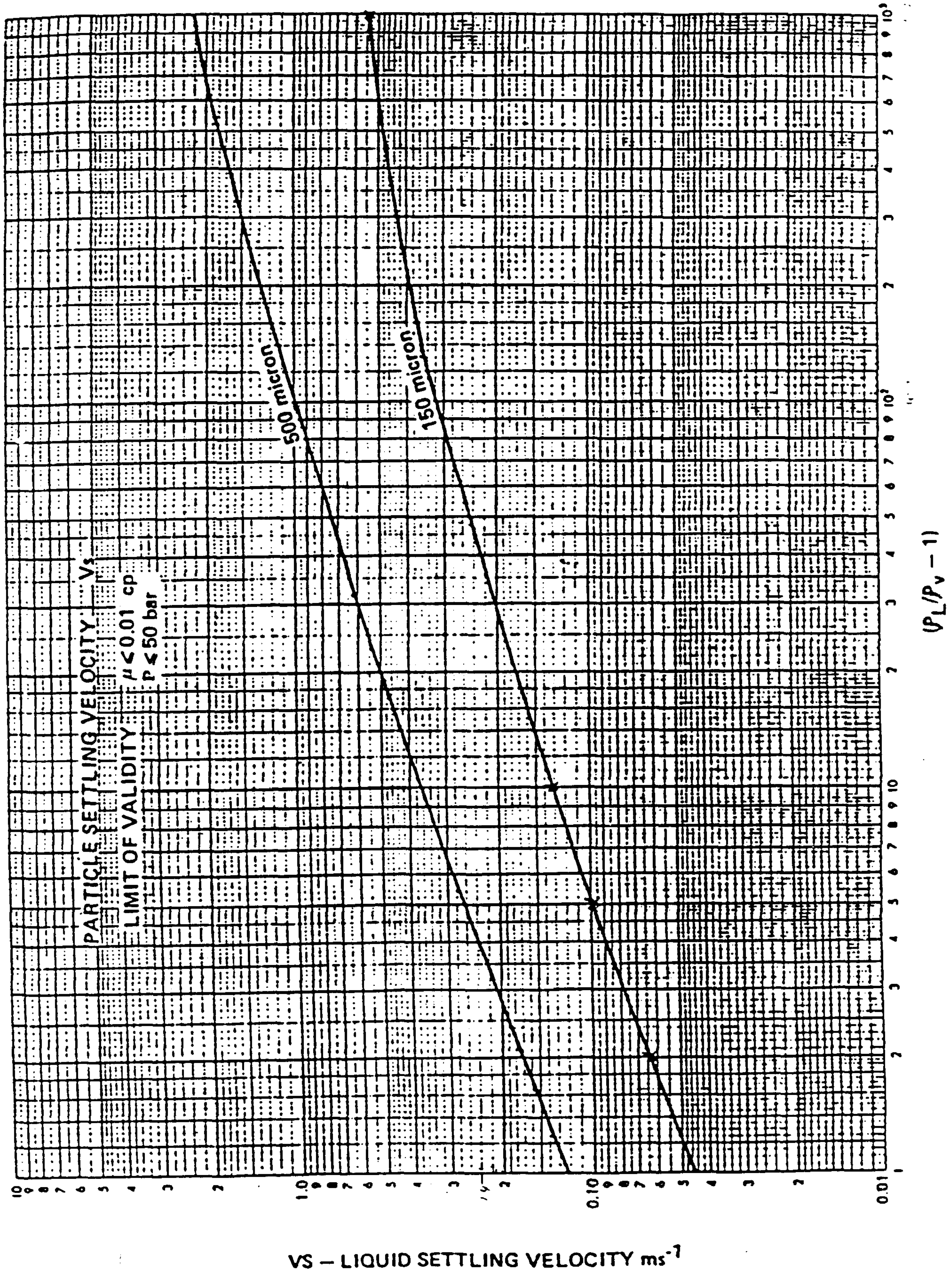
D = 1·5 m (t x D x 20)

TOTAL WEIGHT = ~~12000~~ 10050 kg

TOTAL TEP/DOP/DIP/EXP/SUR	PROCESS CALCULATION SHEET 4	
	CALCULATION FOR HORIZONTAL 3 PHASE	ITEM : EXAMPLE NO :
BY CHK DATE	B TITLE :	JOB N° : REV

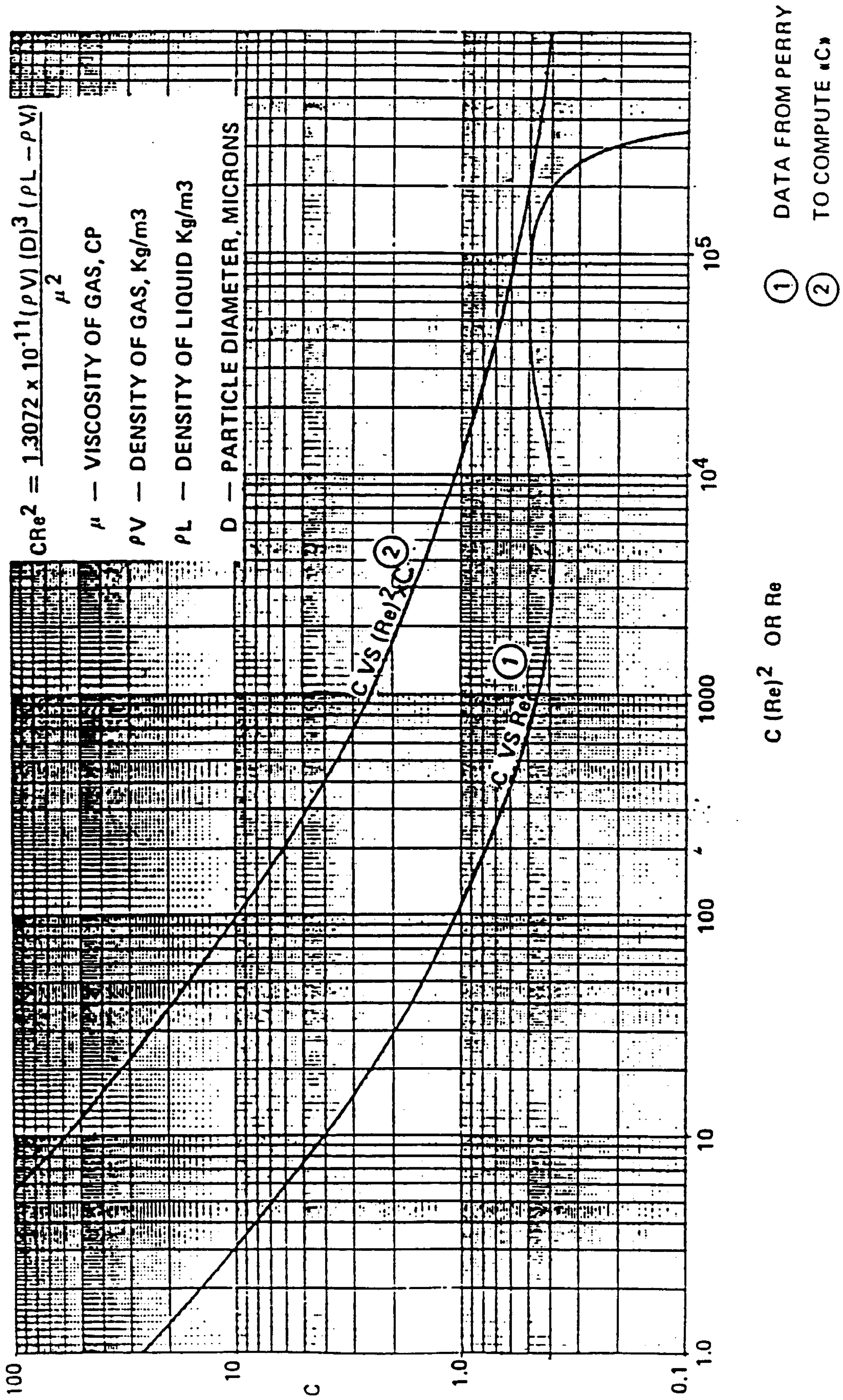
2.19

FIGURE 1



REV

FIGURE 2
DRAG COEFFICIENT (C)
VS Re or C (Re)^2



$$CRe^2 = \frac{1.3072 \times 10^{-11} (\rho V)(D)^3 (\rho L - \rho V)}{\mu^2}$$

μ — VISCOSITY OF GAS, CP
ρV — DENSITY OF GAS, Kg/m3
ρL — DENSITY OF LIQUID Kg/m3
D — PARTICLE DIAMETER, MICRONS

② TO COMPUTE «C»

FIGURE 3
RELATIONSHIP BETWEEN CHORDAL HEIGHT AND CIRCULAR SEGMENTAL AREA

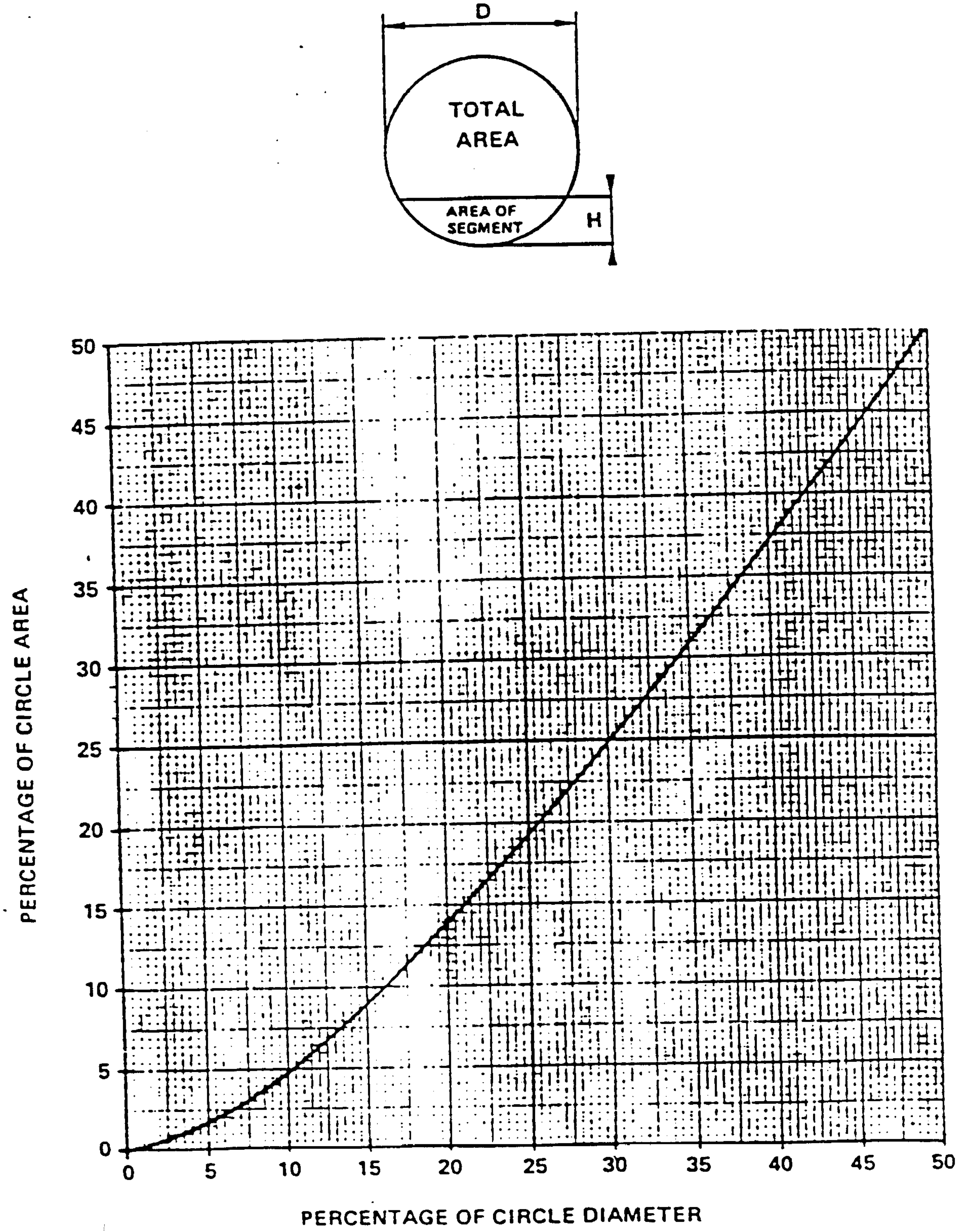
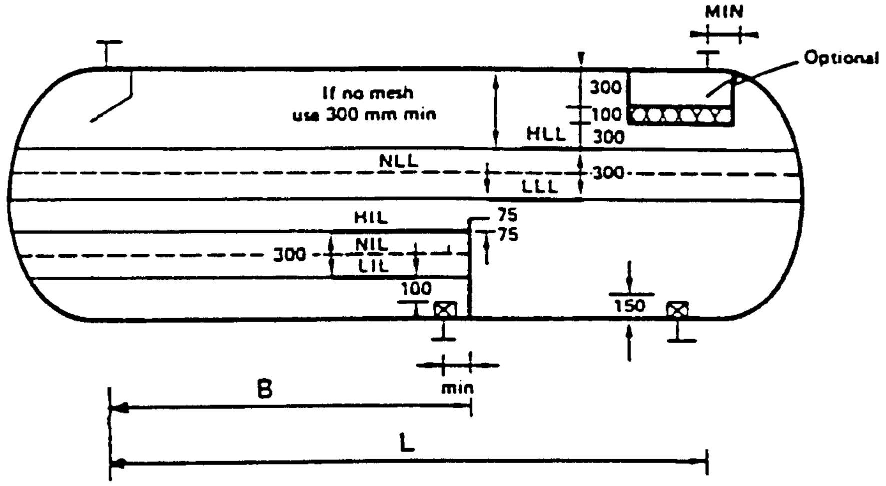


FIGURE 4

3 PHASE SEPARATOR

ALL DIMENSION SHOWN ARE MINIMUM RECOMENDED



- OIL RESIDENCE TIME - Volume between NLL-NIL upto baffle only
  use residence time of 3-6 minutes for design
- OIL SURGE TIME - Volume between HLL and LLL across full length of vessel
  use 4-5 minutes if feeding to another column/vessel
  5 minutes if pumping to storage
  3 minutes if flowing to storage (no pump)
  8 minutes if sole charge to fired heater
- WATER RESIDENCE TIME - Volume between NIL and outlet
  Use 4 minutes minimum
- WATER SURGE TIME - Volume between HIL and LIL
  Use 4-5 minutes minimum

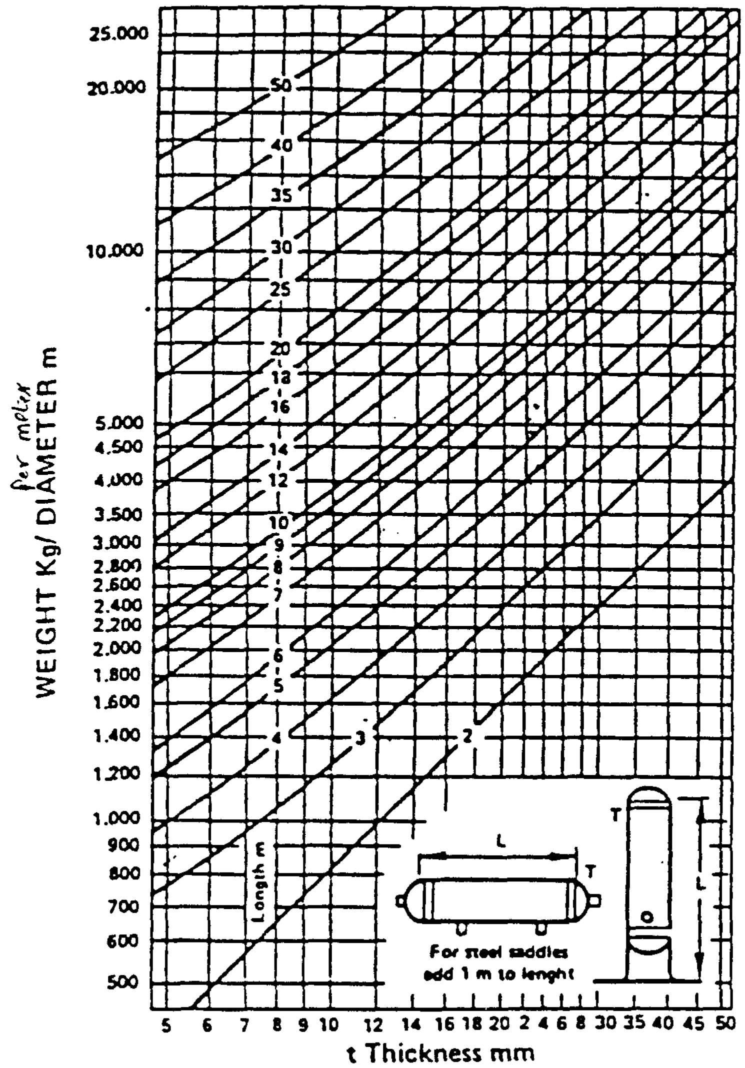


FIGURE 6
Vessel weight estimation

ωt of two heads = Ø (m) x t (mm) x 20 Kg

From hydrocarbon processing
August 1981

essel

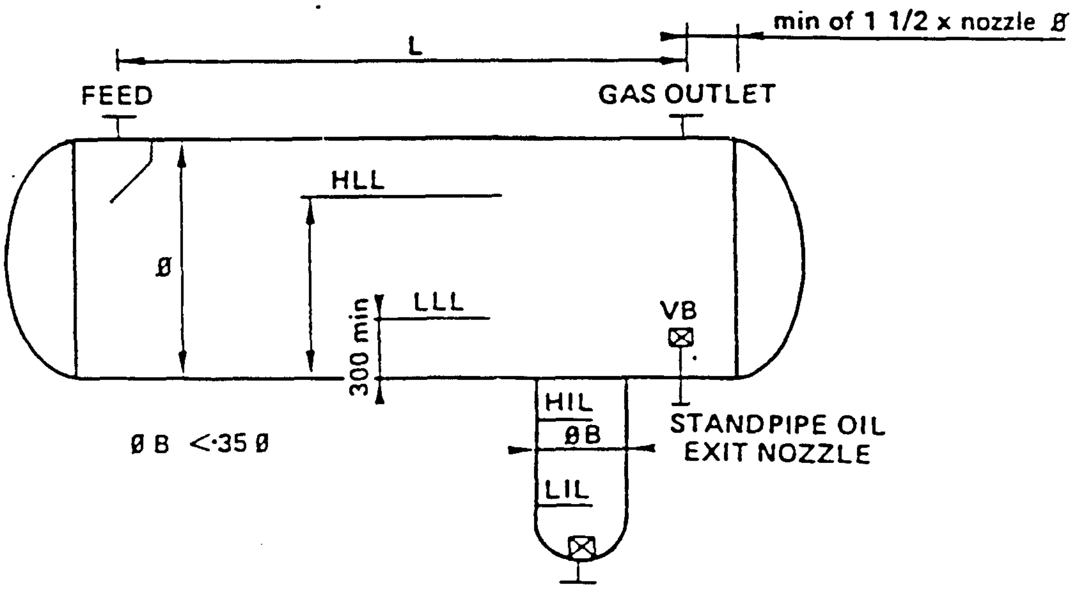


FIGURE 5
3 PHASE SEPARATOR WITH WATER BOOT

TABLE 2

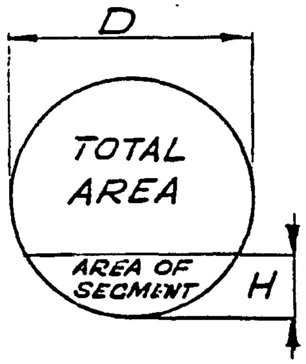
FOR GAUGING HORIZONTAL CYLINDRICAL TANKS - FLAT ENDS

%d - Percentage of Total Diameter of Tank

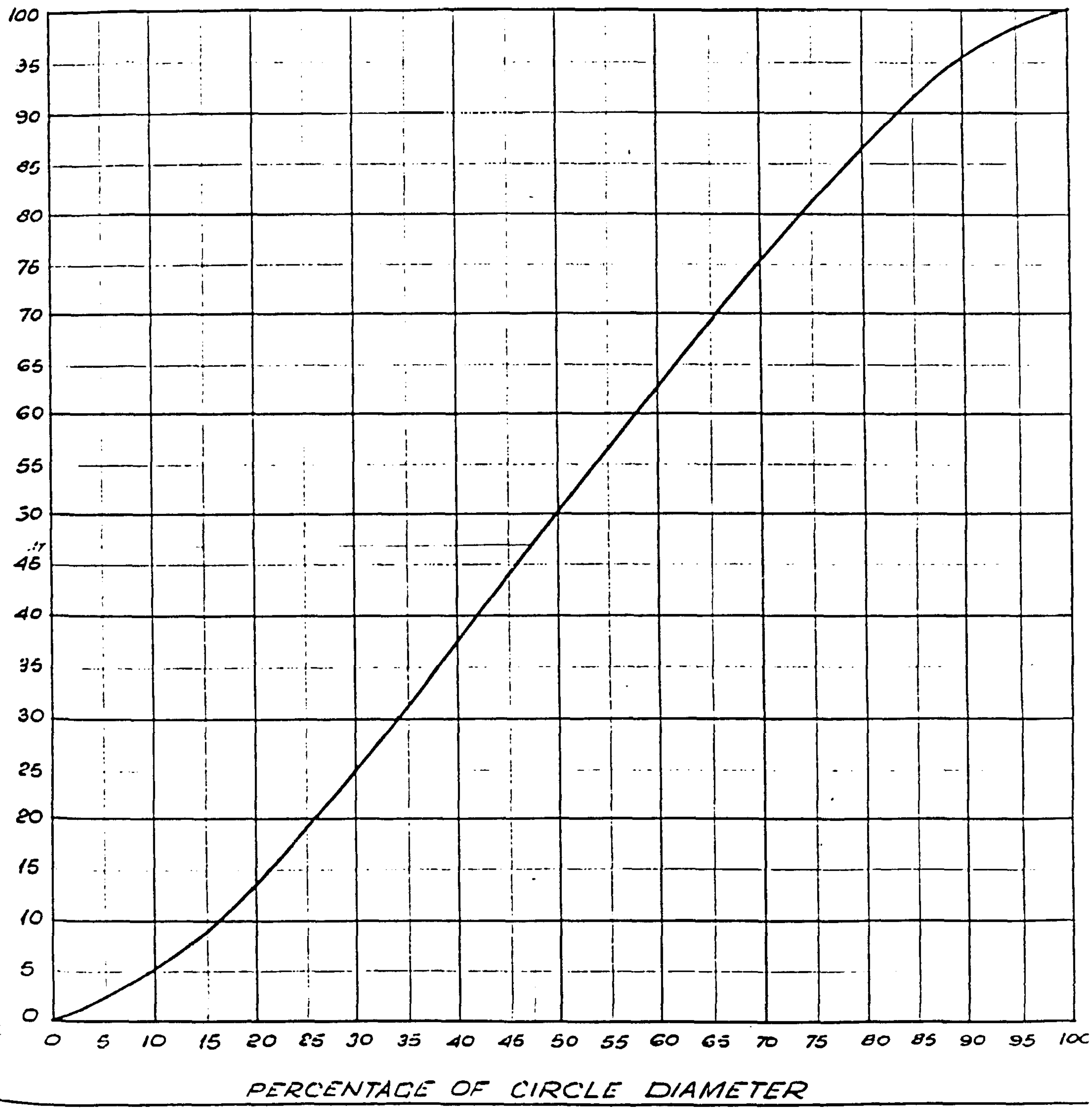
%c - Percentage of Total Capa of Tank

%d	%c	%d	%c	%d	%c	%d	%c
0.1	0.0053	8.8	4.3131	17.6	11.851	28.5	23.49
0.2	0.0152	9.0	4.4582	17.8	12.046	29.0	24.07
0.4	0.0429	9.2	4.6045	18.0	12.240	29.5	24.65
0.6	0.0788	9.4	4.7525	18.2	12.437	30.0	25.23
0.8	0.1212	9.6	4.9015	18.4	12.633	30.5	25.81
1.0	0.1692	9.8	5.0523	18.6	12.831	31.0	26.40
1.2	0.2223	10.0	5.2040	18.8	13.030	31.5	26.99
1.4	0.2800	10.2	5.3580	19.0	13.229	32.0	27.58
1.6	0.3419	10.4	5.5122	19.2	13.429	32.5	28.18
1.8	0.4077	10.6	5.6690	19.4	13.630	33.0	28.78
2.0	0.4773	10.8	5.8258	19.6	13.832	33.5	29.38
2.2	0.5501	11.0	5.9848	19.8	14.035	34.0	29.98
2.4	0.6263	11.2	6.1445	20.0	14.238	34.5	30.58
2.6	0.7061	11.4	6.3060	20.2	14.444	35.0	31.19
2.8	0.7886	11.6	6.4685	20.4	14.649	35.5	31.80
3.0	0.8742	11.8	6.6320	20.6	14.854	36.0	32.41
3.2	0.9625	12.0	6.7970	20.8	15.060	36.5	33.02
3.4	1.0533	12.2	6.9630	21.0	15.267	37.0	33.63
3.6	1.1470	12.4	7.1305	21.2	15.475	37.5	34.25
3.8	1.2432	12.6	7.2990	21.4	15.683	38.0	34.87
4.0	1.3418	12.8	7.4680	21.6	15.892	38.5	35.49
4.2	1.4429	13.0	7.6390	21.8	16.101	39.0	36.11
4.4	1.5461	13.2	7.8110	22.0	16.312	39.5	36.73
4.6	1.6515	13.4	7.9840	22.2	16.524	40.0	37.35
4.8	1.7594	13.6	8.1580	22.4	16.737	40.5	37.98
5.0	1.8693	13.8	8.3330	22.6	16.949	41.0	38.60
5.2	1.8914	14.0	8.5090	22.8	17.161	41.5	39.2
5.4	2.0956	14.2	8.6360	23.0	17.376	42.0	39.86
5.6	2.2116	14.4	8.8645	23.2	17.590	42.5	40.4
5.8	2.3297	14.6	9.0440	23.4	17.806	43.0	41.1
6.0	2.4497	14.8	9.2240	23.6	18.022	43.5	41.7
6.2	2.5715	15.0	9.4060	23.8	18.240	44.0	42.3
6.4	2.6952	15.2	9.5880	24.0	18.457	44.5	43.0
6.6	2.8211	15.4	9.7710	24.2	18.675	45.0	43.6
6.8	2.9483	15.6	9.9560	24.4	18.892	45.5	44.2
7.0	3.0771	15.8	10.142	24.6	19.110	46.0	44.9
7.2	3.2082	16.0	10.327	24.8	19.330	46.5	45.5
7.4	3.3408	16.2	10.515	25.0	19.551	47.0	46.1
7.6	3.4749	16.4	10.703	25.5	20.106	47.5	46.8
7.8	3.6105	16.6	10.893	26.0	20.661	48.0	47.4
8.0	3.7480	16.8	11.082	26.5	21.222	48.5	48.0
8.2	3.8869	17.0	11.273	27.0	21.785	49.0	48.7
8.4	4.0276	17.2	11.465	27.5	22.353	49.5	49.3
8.6	4.1696	17.4	11.657	28.0	22.923	50.0	50.0

PREPARED BY	DATE	APPROVED BY	DATE	RELEASED	CLASSIFIC
W. F. Chaple	122458	[illegible]	6-?-59	06/1959	Confiden



RELATIONSHIP BETWEEN CHORDAL HEIGHT AND CIRCULAR SEGMENT AREA



1. APPLICABILITY

It is not expected that a hand calculation of a tray distillation or absorbtion column be performed by the engineer. For the purpose of a feasibility or pre-project study any required rigorous column sizing would be performed using SSI PROCESS simulator, or similar.

Should, however, a quick estimation of tower diameter and height be required one of the most common methods of hand calculation for valve trays is the "GLITSCH METHOD". An example of the procedure for this method is given in Section 3.

A detailed mechanical design of a tray column is beyond the scope of this guide. For details on glycol towers see package units.

2. DESCRIPTION AND NOTES

2.1. TRAYS

There are basically three types of tray used in distillation columns ; sieve, bubble cap and valve trays. Each type has specific applications and flexibilities dependant on the process criteria. Some of the major aspects are detailed as follows :

Bubble caps

Operation : Vapour passes through "risers" into the bubble cap then bubbles into the surrounding liquid on the tray. Bubbling action effects liquid-vapour contact. The liquid exits the tray via outlet weir and downcomer arrangement to the tray below.

Capacity : Moderately high efficiency (minimum 50 %) is maintained at varying rates due to weir maintaining liquid head.

Efficiency : For many years was the most common type of tray-consequently many published tray efficiencies are available from vendor sources. Note : most expensive type of tray.

Application : All major services excepts coking, polymer formation or other high fouling conditions. Ideal for use in low flow conditions where tray must remain flooded to maintain a vapour seal.

Tray spacing : 18" is normal. Consider 24" to 36" for vacuum conditions.

Sieve trays

	With downcomers	Without downcomers
Operation :	Vapour rises through 1/8" to 1" holes and bubbles through liquid. Liquid flows across tray over weir via downcomer to tray below.	Vapour rises through holes in and bubbles through liquid. Liquid head forces liquid countercurrent through same holes to tray below. Flow is generally random and does not form continuous streams from each hole.
Capacity :	As high as or higher than bubble cap trays for design rates or down to 60 % of design. At lower rates efficiency falls and performance is poor. Generally unacceptable to operate below 60 % capacity.	
Efficiency :	As high as bubble caps at design capacity. Efficiency becomes unacceptable below 60 % design capacity. Not suitable for variable load columns.	
Application :	Systems where high capacity near design rates are to be maintained in continuous service. Handles suspended solid particles well, flushing them down to tray below. Can be problem to run with salting-out systems where trays run hot and dry, holes may plug. Not recommended for oil + gas service due to poor flexibility.	
Tray spacing :	15" average, 9" to 12" acceptable. Use 20" to 30" for vacuum.	12" average, 9" to 18" acceptable. Use 18" to 30" for vacuum.

Valve trays/ballast cap

Generally the same aspects as for sieve trays. Most valve trays are specialist proprietry design for specific operation problems and capacities. Specialist vendors include Glitsch, Koch (flexitray), Nutter, Union Carbide. Best choice of tray for distillation application.

Tray layouts

Not only may the type of bubble cap/valve/sieve hole be specified for a particular design but also the tray hydraulics c liquid path. Common arrangements are shown in Figure 1.

Tray efficiencies

General tray efficiencies to use :

Absorbers		Stripping	
Hydrocarbon oils + vapour	35-50 %	Hydrocarbon oils + vapour	50-80 %
Amine units	15-20 %	(Amine towers usually have 20 actual trays)	
Distillation columns	60-80 %		

2.2. CONDENSERS

. Condensers are usually installed on the overhead of fractionation towers to recover liquid product and provide internal tower reflux. Design of condensers is covered in shell + tube exchanger section.

. Basically two types of overhead condenser exist, partial and total. When using a total condenser the heat load is equal to the latent heat of the saturated overhead vapour. The resultant bubble point liquid is split with some returning as reflux and the remaining portion as distillate product.

For a partial condenser the vapour withdrawn from the accumulator is in equilibrium with the returning reflux and consequently the condenser is acting as an "external" additional tray. The vapour is normally withdrawn under pressure control with all or part of the liquid returning as reflux to the column.

2.3. REBOILERS

. Generally three types of reboiler exist for light hydrocarbon fractionators.

internal reboiler	thermosyphons
external "kettle" type	
external "heat exchanger" type	furnace, electrical

In most cases the "heat exchanger" type is preferred for efficiency.

. The heat exchanger should be located 2-3 m below the exist nozzle from the column so that sufficient head is available for thermal circulation.

. Reboilers may be heated by direct fire, electrical coil, steam, closed heating medium or process fluid exchange.

. Values of U overall (incl. fouling factor) for various types of reboiler and design methods are given in the heat exchanger design guides.

3. CALCULATIONS

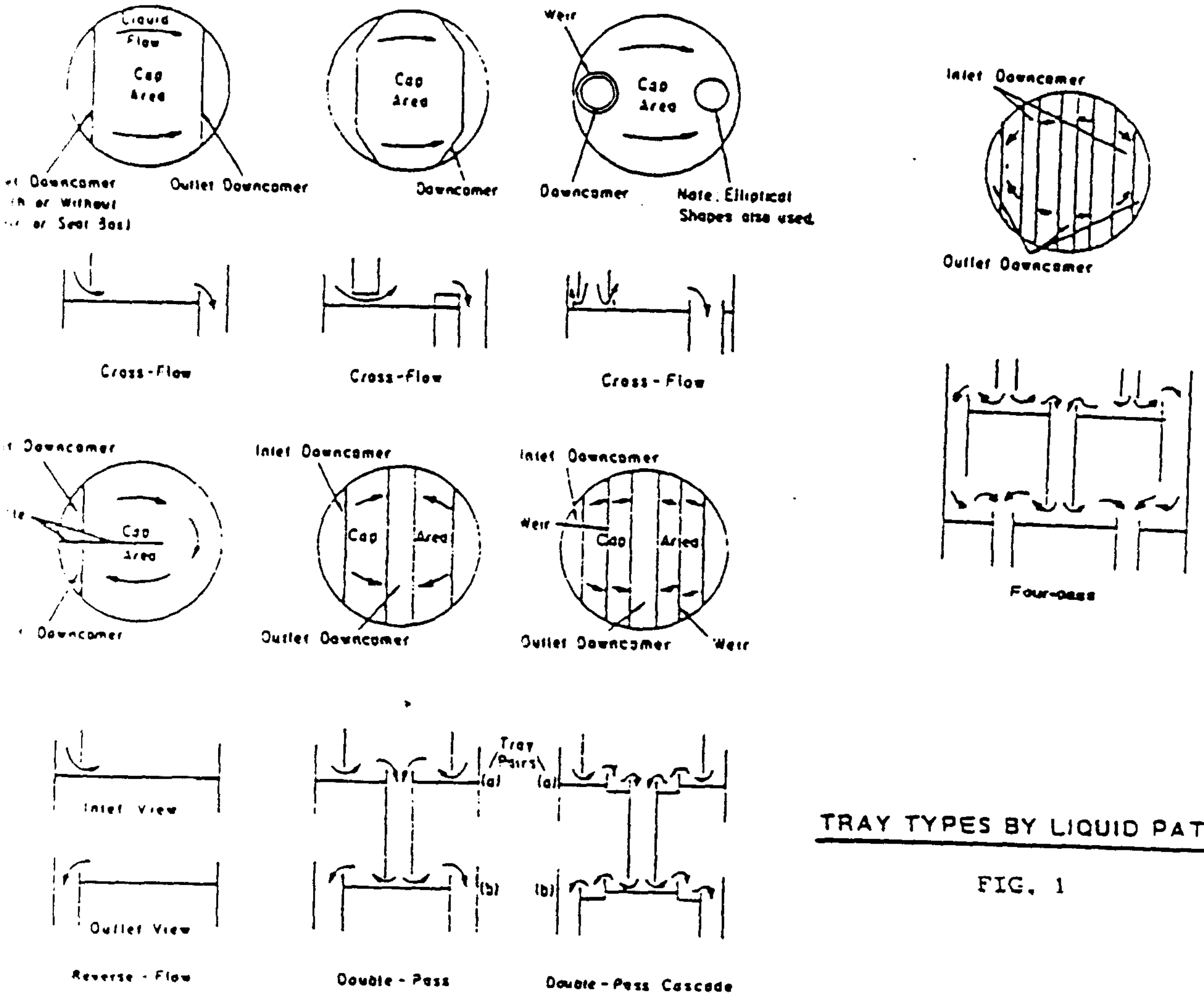
See following pages : 3.5 to 3.11

4. REFERENCES AND USEFUL LITERATURE

4.1. Distillation

Part 1 :	Distillation Process Performance	LUDWIG VOL II - CHAPTER 8
Part 2 :	Hydrocarbons Absorption + stripping	
Part 3 :	Mechanical Designs for Performance	

4.2.	Absorption and fractionation fundementals	CAMPBELL VOL I - CHAPTER 13
4.3.	Gas liquid systems	PERRY - CHAPTER 18
4.4.	Thermosyphon reboiler piping design	W.F. ABBOTT MOBIL - Hydrocarbon Pr. March 1982



TRAY TYPES BY LIQUID PATHS

FIG. 1

No :

3.4

TRAY CALCULATION DATA SHEET

Column item: D 2016

Tray number: 8 Number of passes: 2

1. VAPOR TO TRAY

Fluid	kg/h	MW	kmol/h	t_c	P_c	
HYDROCARBON: LIQUID VAPOUR	98 000 168 800	32.8 31.4	2987 5375	308.7 K	48.42	
TOTAL	266800		8362			

13

t_v = 13.17 °C

$T_v = t_v + 273 = 286.17$ K

$T_c = t_c + 273 = 308.7$ K

$T_r = \frac{T_v}{T_c} = 0.927$

P = 27.6 ATM. abs

Pc = 48.42 ATM. abs

$P_r = \frac{P}{P_c} = 0.57$

Then Z = 0.6771

Vapor density

$$D_v = \frac{12.03 \times MW \times P\,(\text{atm})}{z \times T_v\,(°K)} = \frac{12.03 \times 31.4 \times 27.6}{0.6771 \times 286.17} =$$

$$= \underline{53.806} = \quad \text{kg/m}^3$$

Vapor actual rate

$$C_v = \frac{\text{kg/h}}{D_v} = \frac{168\,800}{53.806} = \underline{3137.2} \quad \text{m}^3/\text{h}$$

HS

Sheet 1 of 4

TOTAL TEP/DGP/DIP/EXP/SUR	PROCESS CALCULATION SHEET		
	TRAY COLUMNS	ITEM : NO :	EXAMPLE
BY CHK	DATE	JOB TITLE :	JOB N° : REV

2. LIQUID FROM TRAY

t_L = 13·2 °C

d_4^{15} = 0·389 d_L at t_L = 0·411 kg/l or x 10^3 = 411 kg/m^3

Liquid flowrate = 98 000 kg/h

$$C_L = \frac{kg/h}{d_L \text{ at } t_L\ (kg/l)} = \frac{98\,000}{411} \cancel{\times 10^{-3}} = \underline{\underline{238.4}}\ m^3/h \text{ at } t_L$$

3. DOWNCOMER DESIGN VELOCITY VD_{dsg}

TS = 18" = 450 mm "TRAY SPACING"

DL - Dv = 357 kg/m^3

VD_{dsgo} = 320 $m^3/h/m^2$ (Fig.2) (610 $m^3/h/m^2$ maxi)

System factor K1 = 1 (Table 1)

VD_{dsg} = VD_{dsgo} x K1 = $\underline{\underline{320}}$ $m^3/h/m^2$

4. VAPOR CAPACITY FACTOR CAF

TS = 450 mm

CAF o = 0·38 (Fig.3)

System factor K_2 = 1·0 (Table 2)

CAF = CAF_o x K_2 = 1·0 x 0·38 = $\underline{\underline{0{\cdot}38}}$

5. VAPOR EFFECTIVE LOAD V Load

$$V\ Load = C_v\sqrt{\frac{Dv}{D_L - Dv}} = 3137.2 \times \sqrt{\frac{53.806}{357}} = 3137.2 \times \sqrt{0.1507}$$

$$= 3137.2 \times 0.388 = 1218\ m^3/h$$

6. APPROXIMATE COLUMN DIAMETER D_T = 2·5 m (Fig.4)

Sheet 2 of 4

TOTAL
EP/DDP/DIP/EXP/SUR

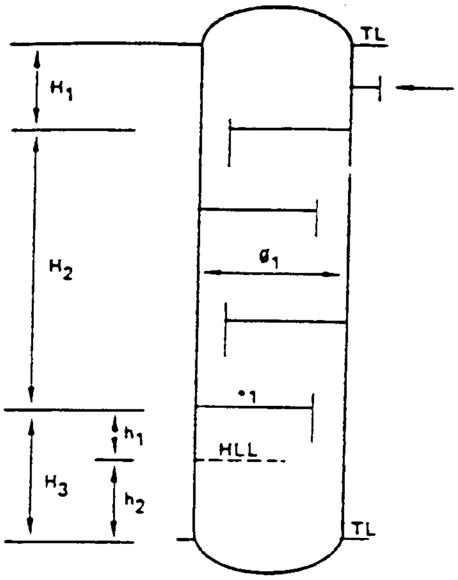
PROCESS CALCULATION SHEET

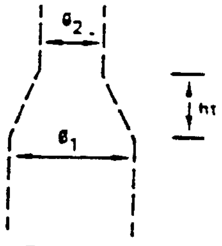
TRAY COLUMNS

ITEM : EXAMPLE
NO :

Y CHK DATE JOB TITLE : JOB N° : REV

3.2. COLUMN HEIGHT ESTIMATION





a. H1 : See design details on vertical vapour-liquid separators. Minimum distance for H1 will be one tray spacing. Minimum distance between inlet nozzle and to tray 300 m.

Selected H1 = 600 mm

b. H2 :

H2 : tray spacing x (number of actual trays - 1)

No actual trays = theoretical trays/η

for η see section 2.1 = 50%

Actual trays = 16 ESTIMATE

Note : if the column diameter changes over the length, the transition piece will be

$h_t = \frac{\sqrt{3}}{2}(Ø1 - Ø2)$ long and H2 will increase by this amount

Selected H2 = 6750 mm

Sheet 3 of 4

TOTAL TEP/DDP/DIP/EXP/SUR	PROCESS CALCULATION SHEET	
	TRAY COLUMNS	ITEM : NO : EXAMPLE

BY | CHK | DATE | JOB TITLE : | JOB N° : | REV

c. H3 :

H3 = h1 + h2

h1 = tray spacing x 2 = 900 mm

h2 - see vertical separator sizing = h6 + h7 + h8

h6 = hold up time volume

For production flowing to :

.	another column	t = 15 min
.	storage	2
.	a furnace	10
.	another unit	5
.	reboiler/heat exch.	5

h6 = 2000 h7 = 500 h8 = 300

$\longrightarrow$ h2 = 2800 mm

H3 = h1 + h2 = 3700 mm

Selected H3 = 3700 mm

TOTAL COLUMN HEIGHT = H1 + H2 + H3 = 6750 mm

TABLE 1

SYSTEM FACTORS

Service	System Factor
Non foaming, regular systems	1.00
Fluorine systems, e.g., BF_3, Freon	0.9
Moderate foaming, e.g., oil absorbers, amine and glycol regenerators	.85
Heavy foaming, e.g., amine and glycol absorbers	.73
Severe foaming, e.g., MEK units	.60
Foam-stable systems, e.g., caustic regenerators	.30

TABLE 2

SYSTEM FACTORS

Service	System Factor
Non foaming, regular systems	1.00
Fluorine systems, e.g., BF_3, Freon	0.9
Moderate foaming, e.g., oil absorbers, amine and glycol regenerators	.85
Heavy foaming, e.g., amine and glycol absorbers	.73
Severe foaming, e.g., MEK units	.60
Foam-stable systems, e.g., caustic regenerators	.60

TABLE 3

Column diameter mm	Minimum recommended Tray spacing : 75 mm
Ø < 1 200	450
1 200 < Ø < 2 500	600
2 500 < Ø < 4 200	700
Ø > 4 200	950

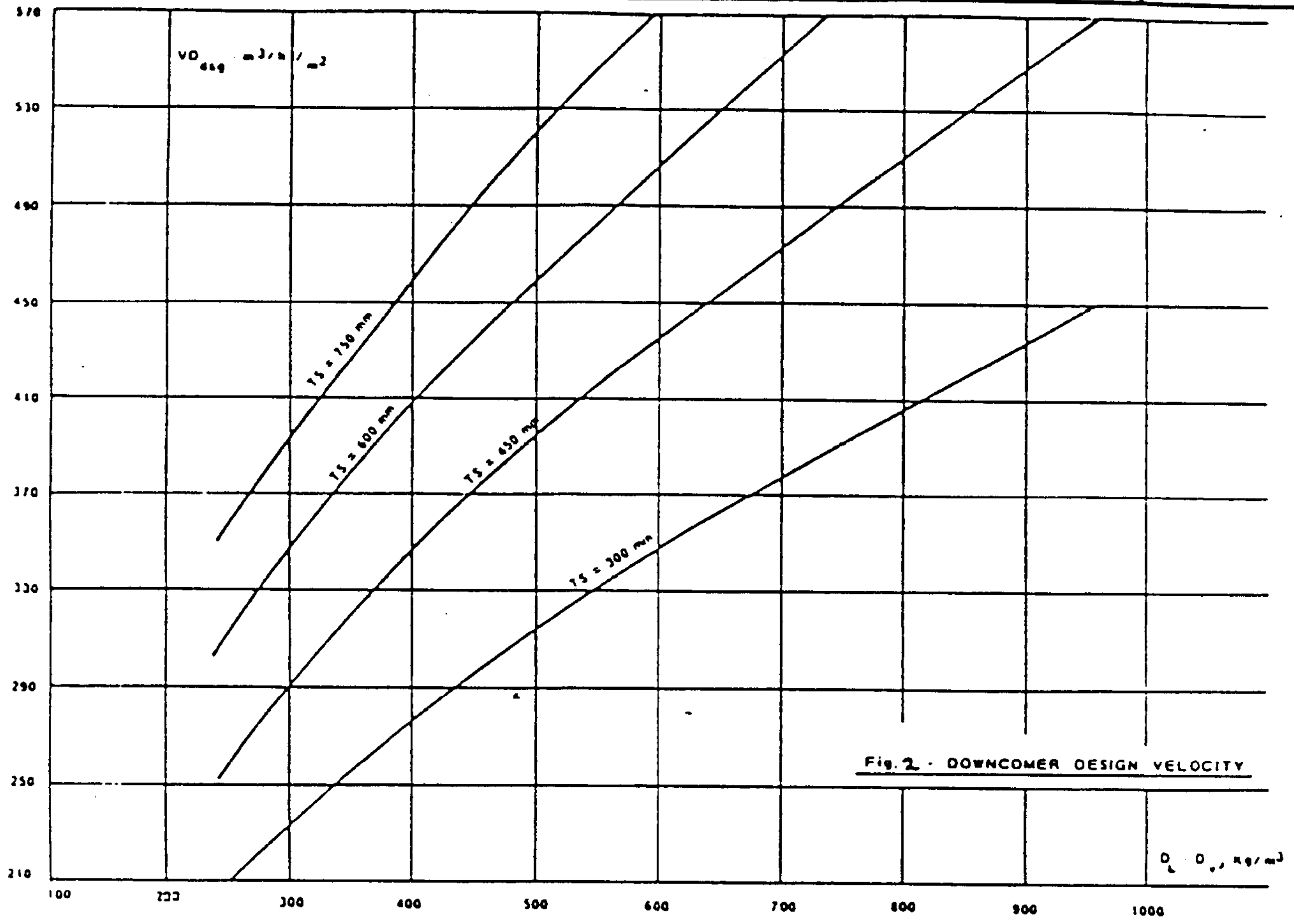


Fig. 2 - DOWNCOMER DESIGN VELOCITY

Fig. 3 FLOOD CAPACITY OF BALLAST TRAYS

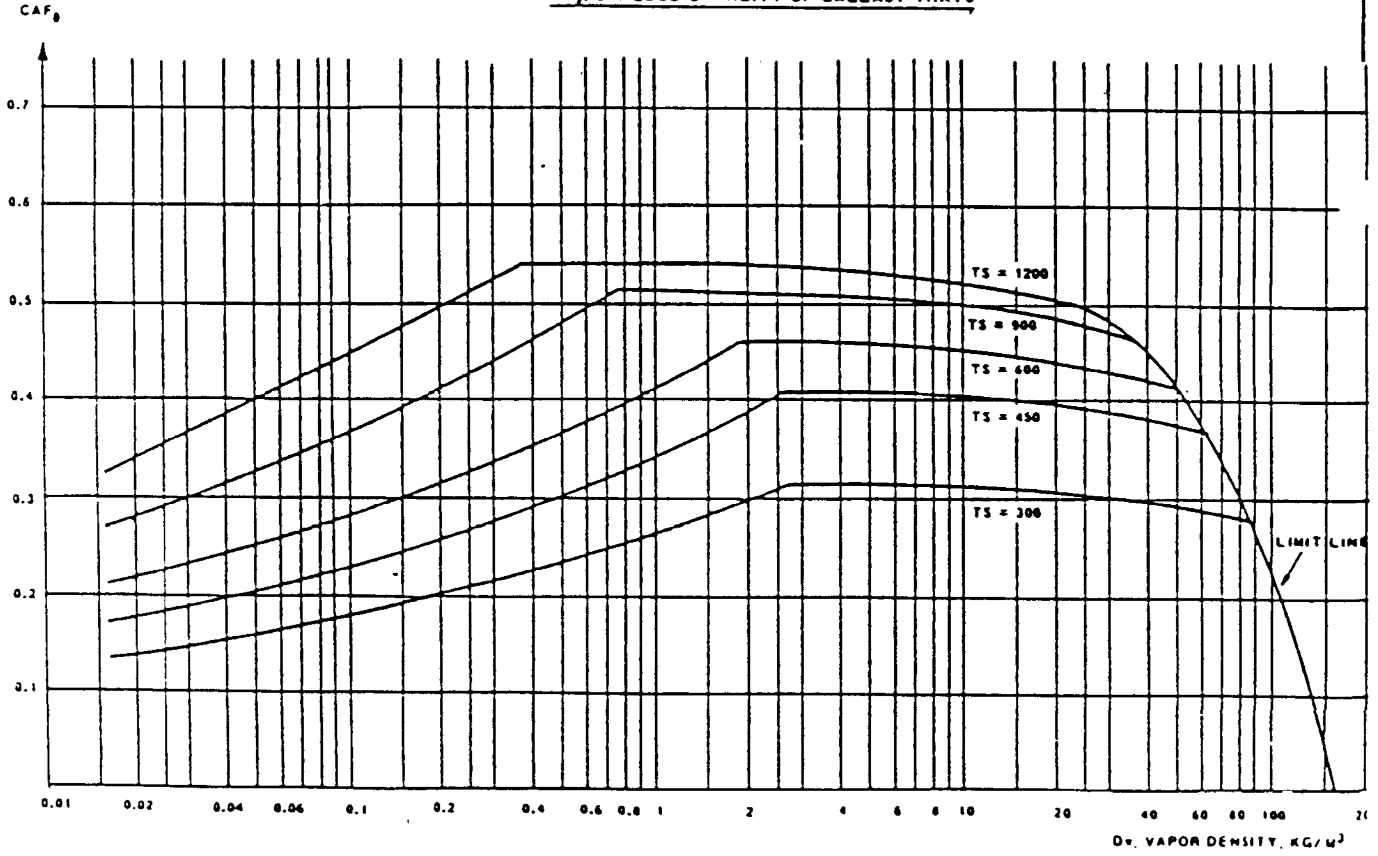
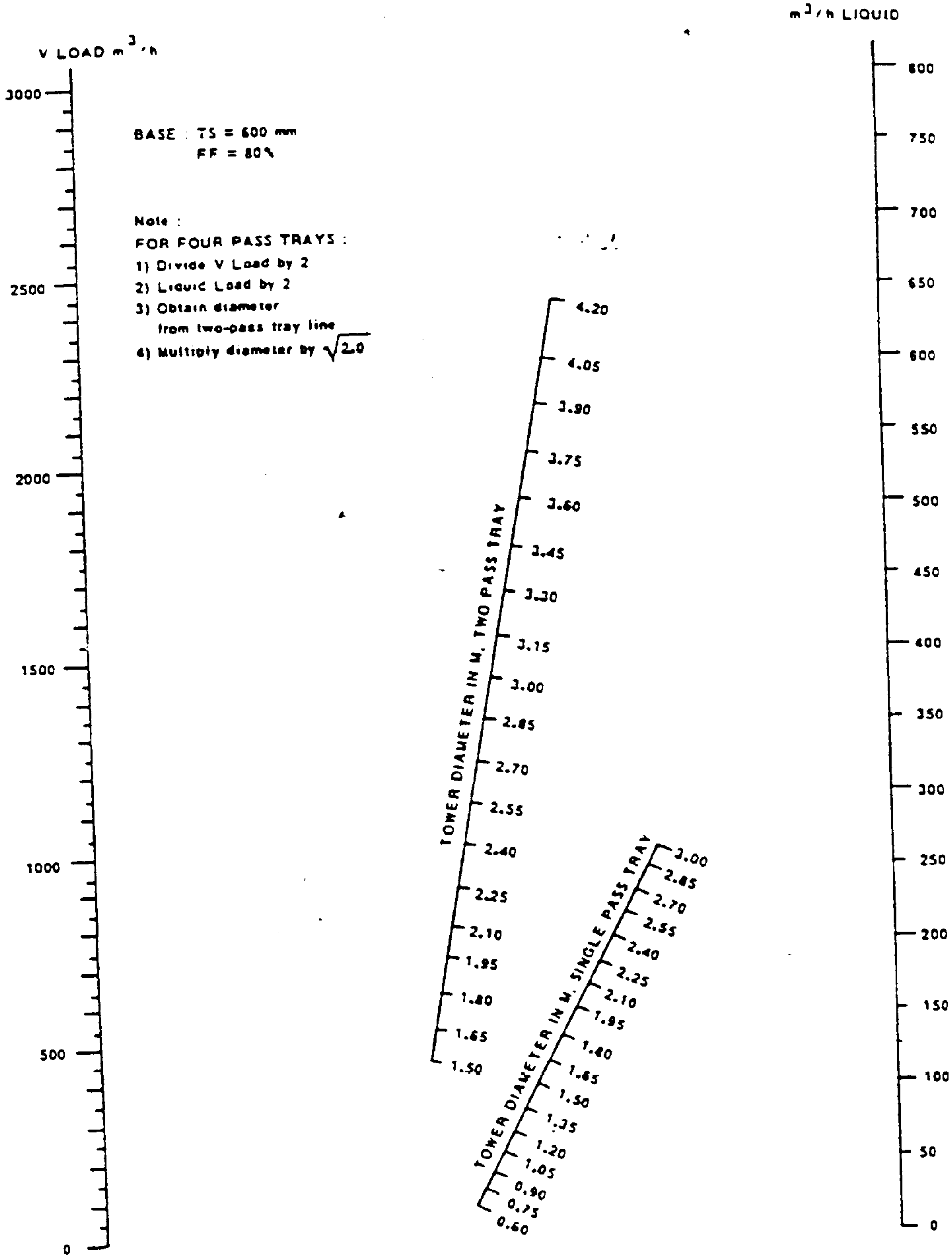


Fig. 4 BALLAST TRAY DIAMETER
(FOR APPROXIMATION PURPOSES ONLY)



BASE : TS = 600 mm
FF = 80%

Note :
FOR FOUR PASS TRAYS :
1) Divide V Load by 2
2) Liquid Load by 2
3) Obtain diameter from two-pass tray line
4) Multiply diameter by $\sqrt{2.0}$

1. APPLICABILITY

FEASIBILITY STUDY : PRE-PROJECT

Under normal circumstences the design of a packed tower would be detailed by a vendor based on process data supplied by the engineer. The detailed design of packed towers is complex and requires specific information regarding both packing type and size and mass transfer data for the fluids contacted.

For the purpose of this design guide details are given on the general arrangement of packed towers, various types of packing and loading and pressure drop correlations.

The determination of the height of a packed tower should be evaluated by a vendor or determined by the engineer if required using methods outlined in design literature (see references). A detailed description is beyond the scope of this guide and is normally unnecessary for feasibility and pre-project level.

2. PACKED TOWER DESCRIPTION + NOTES

A general arrangement of a packed tower is shown in Figure 1.

Packing

The correct selection of a tower packing will normally be made by the vendor based on the required process, flowrates and pressure drops stated. Details on packing are given in :

Table 1 - Packing service applications.

NOTES AND GUIDELINES

- Carbon steel towers may be lined for corrosive service with rubber, plastic or brick depending on the nature of the fluids being processed and the temperatures encountered.
- Towers are generally loaded by dumping the packing rather than stacking. Stacking is more expensive and gives inferior liquid distribution but smaller pressure drop. Certain packing types will be stacked at vendor request.

- Packing heights per support plate/grid should not exceed 12' (3.6 m) for Raschig rings of 15-20' (4.5 - 6 m) for other packing types. Individual bed heights are normally limited to 8 column diameters or 6 m maximum.

- Good liquid distribution over the packing is necessary to promote adequate phase contact within the bed. The streams of liquid should enter the bed on 3" - 6" square centres for small towers with D < 36". For larger towers the number of streams should not be less than $(D/6)^2$.

- Liquid redistributors should be installed after approx. 3 tower diameters for Raschig rings and 5-10 diameters for other packing types. Redistributors are not generally required for stacked packing as the downward liquid flow is vertical.

- In order to reduce ceramic and carbon packing breakage occuring during flow surges hold-down or floating bed limiters are installed on top of the packing. The limiter must be heavy enough to hold down the bed and be able to resettle as the bed moves.
  For plastic or metal packing the bed limiter is bolted in place and does not rest on the packing.

- Packed towers are not recommended for dirty service fluids nor for glycol dehydration. Packed tower should be considered in preference to tray towers for :

  i) small columns with Ø < 2 ft
  ii) acids or corrosive liquids
  iii) highly foaming liquids
  iv) low hold up times
  v) low pressure drop requirement

4. REFERENCES AND USEFUL LITERATURE

4.1 Applied Process Design for chemical + Petrochemical plants - VOL II pp 129-239	LUDWIG
4.2 Design Information for Packed Towers Bulletin DC-11	NORTON Co.
4.3 Tower Packings	Bulletin TP-78 "
Packed Tower Internals	TA-80R "
Hy-Pack	MY-40 "
Interlox saddles	CI-78
4.4 Design Techniques for sizing Packed Towers	- John S. ECKERT Chem. Eng. Process Sept. 1961 VOL 57
4.5 No mystery in packed bed Design	John S. ECKERT Oil and Gas Journal Aug. 24 1970
4.6 Calculator Program for Designing Packed Towers	V.I. PANCUSKA Chem. Eng. May 5 1980
4.7 Packed column Design on a Pocket Calculator	T.J. HIXSON Chem. Eng. Feb. 6 1984
4.8 Packed Columns	Perry Chemical Eng. Handbook pp 18.19 ⟶ 18.47

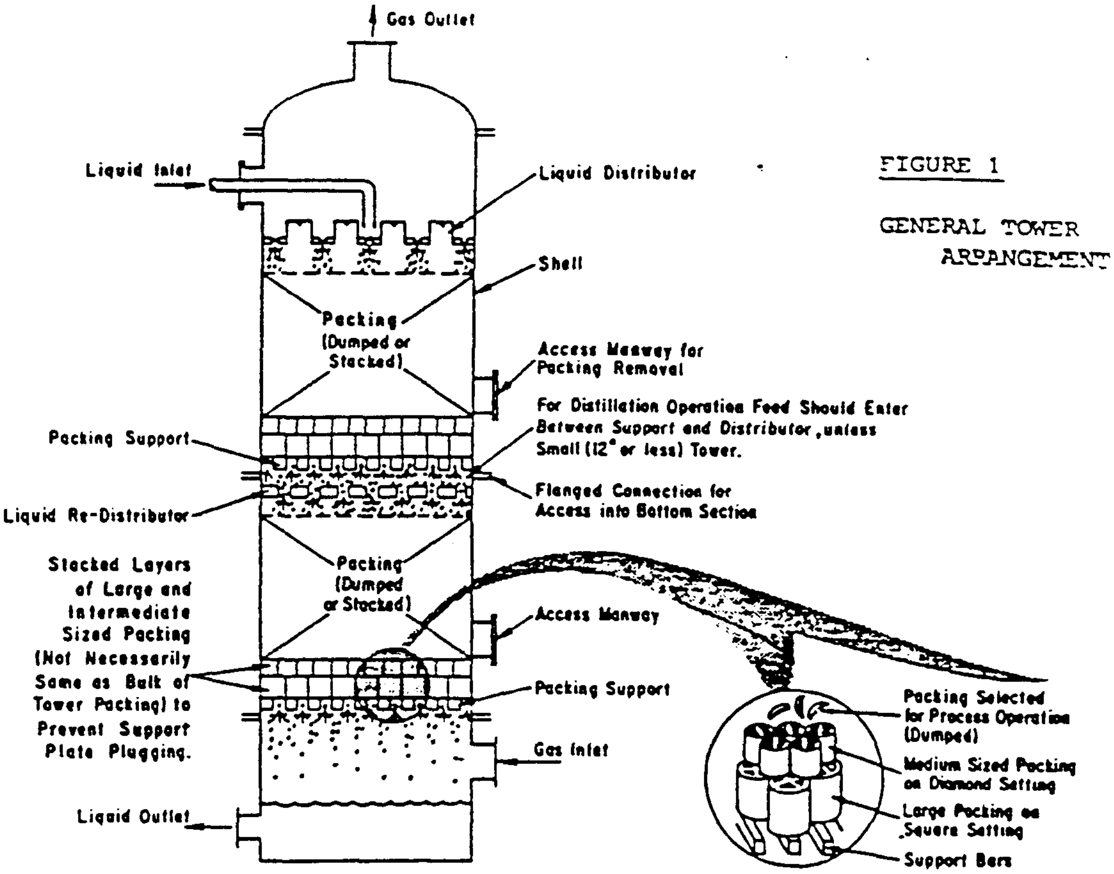


FIGURE 1

GENERAL TOWER ARRANGEMENT

Packing Service Application

Packing Material	General Service Application	Remarks
Glazed and unglazed, Porcelain or Chemical Stoneware	Neutral and acid conditions except hydrofluoric, solvents. Not good in hot caustic (above 70° F.)	Unglazed usual type specified except special requirement of low adsorption on surface. Special ceramics available for mild caustic. Porcelain stronger and more resistant than stoneware.
Carbon	Hot alkali, all acids except nitric, no oxidizing atmospheres.	Stand Thermal shock, low cubic weight
Plastic	Alkali, salts, aqueous and acids depending on resin	Light weight
Steel and other light gauge metals	Hot alkali for steel, other service to suit metals	May be heavier than ceramic, more expensive

TABLE 1

PACKING SERVICES AND SIZES

Packing Size

This affects contact efficiency; usually, the sma packing is more efficient; however, pressure drop creases.

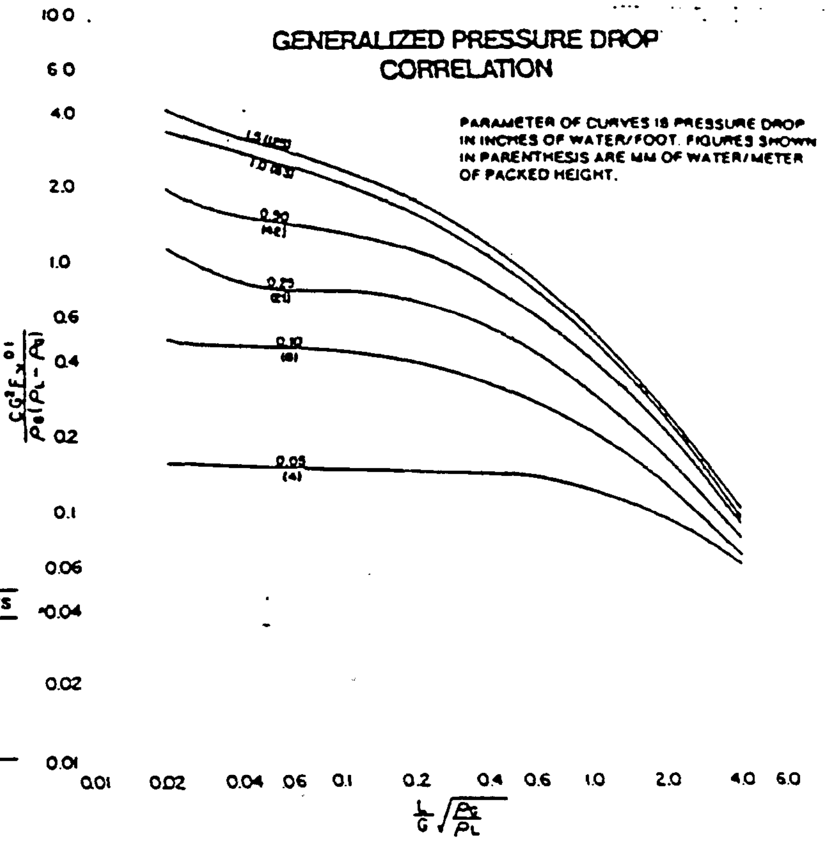
As a general guide, use:

Packing Size, Nominal, inches	Column Diam., inch
½" — ⅝"	6" — 12"
⅝" — 1"	12" — 18"
1" — 1½"	18" — 24"
1½" — 2"	24" — 48"
2" — 3"	36" — larger

3.0

Design Techniques for Sizing Packed Towers

Reproduced from NORTON
'Design Information for Packed Towers'
Bulletin DC-11



GENERALIZED PRESSURE DROP CORRELATION

PROPERTY	SYMBOL	BRITISH UNITS	METRIC UNITS
Gas Rate	G	Lbs/ft² sec	KG/M² s
Liquid Rate	L	Lbs/ft² sec	KG/M² s
Gas Density	ρ_G	Lbs/ft³	KG/M³
Liquid Density	ρ_L	Lbs/ft³	KG/M³
Liquid Viscosity	ν	Centistokes	Centistokes
Conversion Factor	C	1 000	10.764
Packing Factor	F	—	—

Packing Factors

(DUMPED PACKING)

Packing Type	Mat'l.	Nominal Packing Size (Inches)										
		¼	⅜	½	⅝	¾	1 or #1	1¼	1½	2 or #2	3	3½ or #3
Hy-Pak™	Metal						43			18		15
Super Intalox* Saddles	Ceramic						60			30		
Super Intalox Saddles	Plastic						33			21		16
Pall Rings	Plastic				97		52		40	24		16
Pall Rings	Metal				70		48		33	20		16
Intalox* Saddles	Ceramic	725	330	200		145	92		52	40	22	
Raschig Rings	Ceramic	1600	1000	580	380	255	155	125	95	65	37	
Raschig Rings	1/32" metal	700	390	300	170	155	115					
Raschig Rings	1/16" metal			410	290	220	137	110	83	57	32	
Berl Saddles*	Ceramic	900		240		170	110		65	45		

Packing factors determined with an air-water system in 30" I.D. tower.
*Data by Leva

1. To design a packed tower, first it is necessary to know the amount of liquid or gas to be handled and from this determine the liquid-gas ratio (L/G). The densities of both liquid and gas should be known and the term $\sqrt{\frac{\rho_G}{\rho_L}}$ is calculated, then the abscissa, $X = \frac{L}{G}\sqrt{\frac{\rho_G}{\rho_L}}$

2. After calculating the value of X consult the generalized pressure drop correlation above. It will be noted that there are a series of marked parameters ranging from 0.05 to 1.5 inches of water pressure drop per foot of packed depth (4 to 125 mm of water pressure drop per meter of packed depth).
Normally, a packed tower should be designed to operate at a maximum economical pressure drop. The design engineer must determine the best balance between higher capital investment vs. lower operating costs for low pressure drop towers, and low capital investment vs. higher operating costs for towers operating at higher pressure drop. Ordinarily, packed towers are not operated above 1.0 inch of water pressure drop per foot of packed depth (83 mm of water pressure drop per meter of packed depth). Higher pressure drops are possible when instrumentation is such as to maintain a constant pressure drop. Most absorbers and regenerators are designed for low pressure drop operation, i.e., somewhere between 0.20 and 0.60 inches of water pressure drop per foot of packed depth (17 and 50 mm of water pressure drop per meter of packed depth). Atmospheric or pressure distillations are designed for pressure drops of 0.50 to 1.0 inches of water pressure drop per foot of packed depth (42 mm to 83 mm of water pressure drop per meter of packed depth). Vacuum distillations run the complete range of pressure drop and are dependent on what is to be accomplished and whether the vacuum is solely for improved separation or whether it is to reduce temperature of separation to improve product quality.
The designer should keep in mind that the pressure drop parameters shown on the generalized pressure drop correlation are in inches of water (mm of water). Therefore, when designing columns operating with other liquids, special consideration should be given, especially when the specific gravity of the liquid is substantially less than that of water. For example, an absorber handling a hydrocarbon with a specific gravity of 0.5 will exhibit the properties of a tower with a hold-up volume corresponding to a pressure drop approximately 80% greater than that for which it was designed.

3. After having determined the value of X as the abscissa in Step 1, and selected an operating pressure drop in Step 2, the value of the ordinate, Y, may be determined by the use of the generalized pressure drop correlation. Locate the value of the abscissa on this chart; move vertically until the proper pressure drop parameter is contacted; then move horizontally from this point to the left hand edge of the chart and read the value of the ordinate. Make the value equal to this group of variables:

$$Y = \frac{C\,G^2 F \nu^{0.1}}{\rho_G(\rho_L - \rho_G)}$$

4. Then $G = \left(\frac{Y \rho_G(\rho_L - \rho_G)}{C F \nu^{0.1}}\right)^{0.5}$

The value of all variables is known except for the viscosity of the liquid, the packing factor F and the gas rate G. The viscosity of the liquid can be determined from literature, experiment or approximation. The packing factors of all sizes of packing are given in the table on page 4. Broadly speaking, packings smaller than 1 inch size are intended for towers one foot or smaller in diameter, packings 1 inch or 1½ inch in size for towers over one foot to three feet (0.3 to 0.9 meters) in diameter and 2 or 3 inch packings are used for towers three or more feet (0.9 meters) in diameter. The designer should select the proper size of packing, and therefore the proper packing factor in this first calculation.

5. Now that all variables have assigned values, G may be calculated and the diameter of the tower determined by using the equation:

$$D = \left(\frac{4A}{\pi}\right)^{0.5} = 1.13A^{0.5}$$

where $A = \frac{G' \text{ total lbs./sec.}}{G \text{ lbs./sq. ft./sec.}}$ as determined from Step 4.

This establishes the diameter of the tower which, when filled with the packing selected and operated at design liquid and gas rates, will develop the selected pressure drop.

6. The depth of the bed required will be dependent upon the approach to total mass transfer required with 100% mass transfer theoretically requiring a bed of infinite depth. Therefore towers are always designed to operate at less than total mass transfer. In gas absorption problems, the bed is usually calculated from the mass transfer co-efficient:

$$K_Ga = \frac{N}{H\,A\,P\,\Delta Y_{LM}}$$

because the drive is from the gas to the liquid phase.
Or if a stripping operation is involved then the mass transfer co-efficient becomes:

$$K_La = \frac{N}{H\,A\,\Delta X_{LM}}$$

because the drive is from the liquid to the gas phase.
The definitions of the terms for the above equations for K_Ga and K_La are as follows:

K_Ga = Mass transfer co-efficient, lb. moles/ft.³ Hr. Atm.
K_La = Mass transfer co-efficient, lb. moles/ft.³ Hr.
N = Lb. moles transferred/Hr.
H = Packed depth of tower packing, ft.
A = Tower cross sectional area, ft.²
P = System pressure, atmospheres
Y_i = Gas phase mole fraction, component i
Y_i^* = Gas phase mole fraction of component i in equilibrium with liquid bulk phase mole fraction of component i, X_i
X_i = Liquid phase mole fraction, component i
X_i^* = Liquid phase mole fraction of component i in equilibrium with gas bulk phase mole fraction of component i, Y_i

For counter-current gas-liquid flow:

$$\Delta Y_{LM} = \frac{(Y_i - Y_i^*)_2 - (Y_i - Y_i^*)_1}{L_n\left[(Y_i - Y_i^*)_2 / (Y_i - Y_i^*)_1\right]}$$

where the subscripts 1 and 2 refer to the top and bottom of the column respectively. The equation of ΔX_{LM} is analogous to the equation for ΔY_{LM} given above.
K_Ga and K_La data are available for most absorption and stripping operations. Because the data on absorption of CO, with caustic soda solution are so complete for the various packings, it is not at all unusual to use the data as a ratio information source for design with other packings and other rates than those for which direct information exists.
Distillation units are generally designed on the basis of HETP (height equivalent to a theoretical plate). Hundreds of distillation experimental studies have caused us to conclude that the properties of a system have little to do with the HETP value, provided that good distribution is maintained and the packed bed is operated with pressure drops of at least 0.20 inches of water pressure drop per foot of packed depth (17 mm of water pressure drop per meter of packed depth). Mass transfer taking place in packed beds, where any substantial amount of pressure drop exists, will occur predominately as a result of turbulent contact of gas and liquid rather than as a diffusional operation governed by film resistances at the interface.
Once the total bed depth has been determined, the depth of individual beds must be established. Generally, individual bed depth is held to eight column diameters or 20 ft., although under certain conditions 30 ft. beds are permissible.
Proper tower internals are required to realize the full potential of the packing in any application. (See engineering manual TA-80R.)

4. HEAT EXCHANGERS

1. APPLICABILITY

It is not expected that a hand calculation of shell and tube exchangers be performed by the engineer. For the purpose of a feasibility or pre-project study any required rigorous calculation would be performed using computer programs HTRI or HTFS.

A quick estimation of heat exchange area, shell diameter and tube length should be done by hand calculation. An example of the procedure is given in Section 3.

A detailed mechanical design is beyond the scope of this guide.

2. DESCRIPTION AND NOTES

2.1. DESCRIPTION

The flow of fluids inside the exchanger varies according to requirements and can be single or multi-pass on either tube or shell side. Figure 1 shows the types of tubu heat exchanger manufactured to TEMA standards ;

The following types are frequently found :

- Exchangers (Heaters)
- Condensers
- Reboilers (Thermosyphon or forced circulation)
- Evaporators (Kettle)
- Chillers (using refrigerants)

2.2. SELECTION OF SHELL OR TUBE SIDE FOR THE FLUIDS

a. Tube side :

really ??

- Most of time highest pressure fluid
- Cooling water, steam
- Fouling or corrosive fluid
- Sea water (it is always recommanded to install the sea water on tube side)

b. Shell side :

- Fluid with the highest viscosity
- Condensation
- Evaporation (refrigerants in chiller)
- Least fouling fluid
- Most of time lowest pressure fluid

2.3. SELECTION OF TUBES

- Standard length : 12', 16', 20' but longer tube lengths are possible (upto 40')
- Diameter commonly used : 3/4", 1"
- Pitch commonly used : triangular or square. External tube cleaning is possible with square-pitch only.

2.4. TUBE SIDE VELOCITIES

- The tube side velocity for most materials and services should be held between about 1.3 to 2.5 m/s.
- Below 1 to 1.2 m/s fouling will be excessive, much above 2.5 m/s erosion can become a problem.

2.5. CHARACTERISTICS OF TUBES

BWG = BIRMINGHAM WIRE GAGE

External diameter (in et mm)	BWG	THICKNESS (mm)	Internal diameter (cm)	SECTION (cm^2)	AREA (m^2/m)		WEIGHT (kg/m)
					External	Internal	
1/2 in (12.7 mm)	14	2.10	0.848	0.565	0.0399	0.0266	0.600
	16	1.65	0.940	0.694		0.0295	0.490
	18	1.24	1.021	0.819		0.0321	0.384
3/4 in (19.05 mm)	10	3.40	1.224	1.177	0.0598	0.0384	1.436
	12	2.77	1.351	1.434		0.0424	1.216
	14	2.10	1.483	1.727		0.0466	0.963
	16	1.65	1.575	1.948		0.0495	0.774
	18	1.24	1.656	2.154		0.0520	0.597
1 in (25.4 mm)	10	3.40	1.859	2.714	0.0798	0.0584	2.024
	12	2.77	1.986	3.098		0.0624	1.696
	14	2.10	2.118	3.523		0.0665	1.324
	16	1.65	2.210	3.836		0.0694	1.057
	18	1.24	2.291	4.122		0.0720	0.811
11/4 in (31.75 mm)	10	3.40	2.494	4.885	0.0997	0.0783	2.604
	12	2.77	2.616	5.375		0.0822	2.158
	14	2.10	2.743	5.909		0.862	1.682
	16	1.65	2.845	6.357		0.0894	1.340
	18	1.24	2.921	6.701		0.0918	1.024
11/2 in (38.1 mm)	10	3.40	3.124	7.665	0.1197	0.0981	3.185
	12	2.77	3.251	8.300		0.1021	2.634
	14	2.10	3.378	8.962		0.1061	2.039
	16	1.65	3.480	9.512		0.1093	1.622
	18	1.24	3.556	9.931		0.1171	1.237

2.6. TEMPERATURE APPROACH AND PINCH

- minimum temperature approach 5 °C.
- minimum pinch for condenser or chiller 3 °C.

2.7. DESIGN MARGIN

- 10 % on area is recommended.

2.8. PRESSURE DROP

- Allowable Δ P varies with the total system pressure and the phase of fluid.
- Liquid pressure drops of 0.7 to 1.0 bar per exchanger are common. The equivalent gas drop is about 0.2 to 0.5 bar.
- Some exchangers have low pressure losses and as reboiler and condenser (less than 0.1 bar) especially those in vacuum system.

2.9. CHOICE OF HEAT EXCHANGER TYPE (Figure 1)

a. Front end stationary head types

- Type A : Used for frequent tube side cleaning due to the ease of dismantling the cover.
- Type B : Cheaper than Type A but the dismantling of the bonnet is more difficult. To be used for clean products.
- Type C : Cheaper than Type A for low pressure. The price increases quickly with the pressure. This type is practically never used.
- Type D : Special for high pressure P > 200 bar.

b. Shell types

- Type E : In general the most commonly used.
- Type F :
  - Advantage : Fluids flow at perfect counter current (F = 1).
  - Disadvantage :
    - leakage between the longitudial baffle and shell -decreases in value.
    - mechanical problems from expansion.
    - low pressure drop eg : < 1 bar (risk of damage of the longitudinal baffle).

This type will be used only to avoid a great number of Type E shells in series.

- Type G & H : Used for low $\Delta P \approx 50$ mbar as for thermosyphon reboiler. Vertical baffles are not installed for these types and due to that the length of the shell must be limited.
- Type J : Used for high flow or high ΔP for Type E and also sometimes on condensates to avoid the use of vapor belt.
- Type K : Used for vapor separation is required ie chiller, some reboilers...

c. Rear end, head types

- Types L, M and N : Fixed tube sheet, used for clean fluid on shell side and for low $\Delta T < 30$ °C. If $\Delta T > 30$ °C use other head types or install an expansion joint cn the shell.

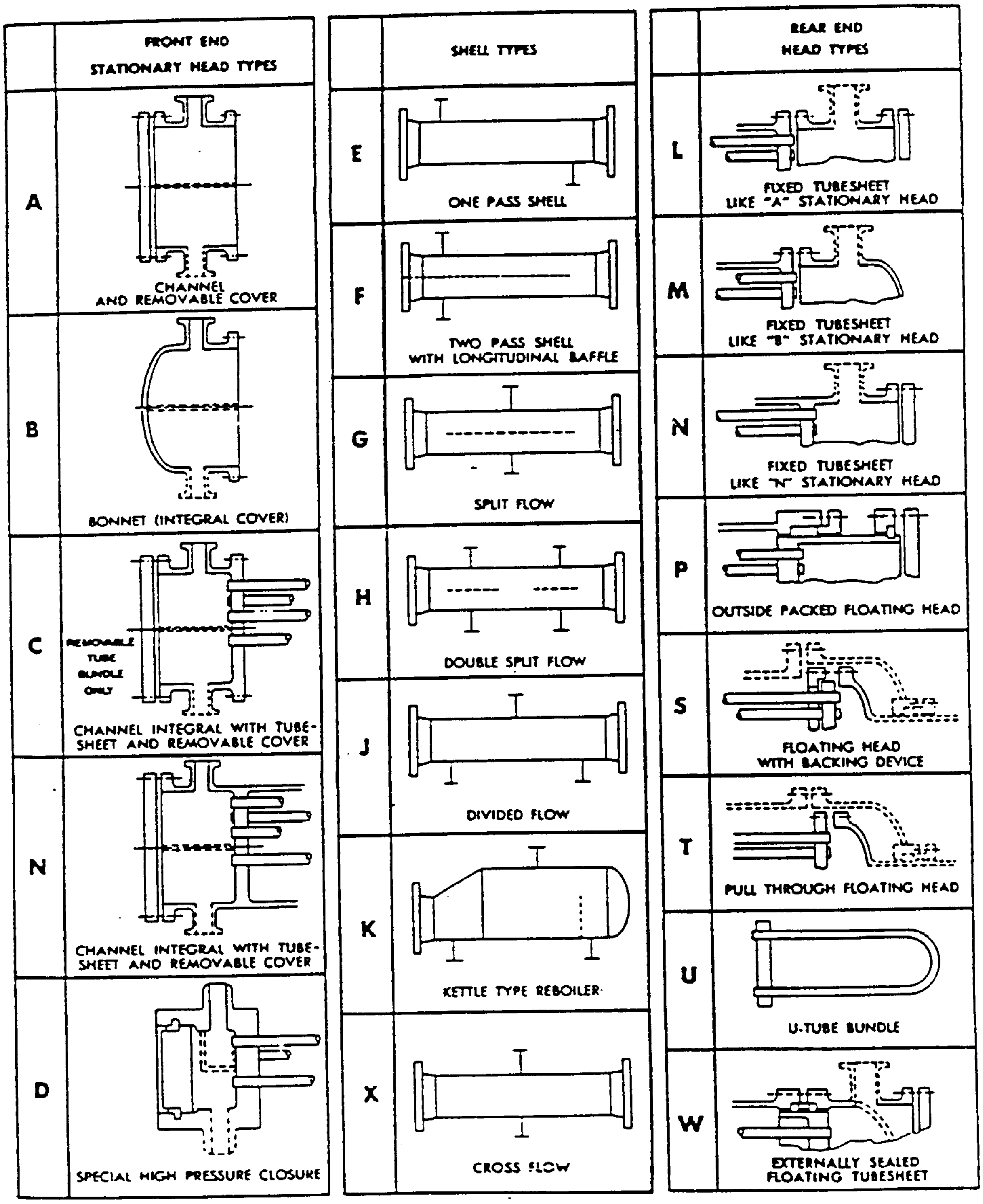
  Type L and N will be used for dirty fluid on tube side. For the other cases the type M will be used it is the cheapest.
- Type P : Generally not used.
- Type S : Used very frequently, no restrictions.
- Type T : For frequent dismantling, expensive, shell diameter larger than type S for same number of tubes generally not used.
- Type U : For clean fluids on tubeside no other restrictions, low cost.
- Type W : Generally not used.

d. Conclusion

The most frequently used types are : BES, BEW, AES, BEM, divided flow, BEU.

Heat-exchanger nomenclature

FIGURE 1



Page No :

4.3

ımantling

is more

s quickly

l).

and shell

damage

ies.

e to that

times on

ers...

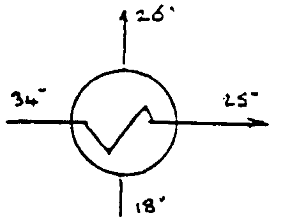
e and for

expansion

cases the

than type

BEU.

ITEM :		VALUE		NOTES :
DUTY Q (2)	kcal/hr	0.5×10^6		Indicate temperatures 
HOT FLUID				
Inlet temperature T1	°C	34		
Outlet temperature T2	°C	25		
COLD FLUID				
Inlet temperature t1	°C	18		
Outlet temperature t2	°C	26		
T1 - t2	°C	8		
T2 - t1	°C	7		
LMTD (1)	°C	7.5		
t2 - t1	°C	8		
T1 - t1	°C	16		
T1 - T2	°C	9		
$P = \frac{t2 - t1}{T1 - t1}$		0.5		
$R = \frac{T1 - T2}{t2 - t1}$		1.125		
F = LMTD correction factor (2)	Fig. 2	0.95 ✓	0.74	
Number of shell passes (3)	Fig. 2	2	1	
Number of tube passes	Fig. 2	4	2	
HEAT TRANSFER COEFF. U	kcal/hr m2 °C	~ 400		including fouling factor
HEAT TRANSFER AREA $A = \frac{Q}{U.F.LMTD}$	m2	175		F = 0.95
ESTIMATED TUBE LENGTH ESTIMATED SHELL DIAM	FT ins	20 29" (740 mm)		3/4" □ tube pitch 16 GWG. ≡ 480 tubes
ESTIMATED WEIGHT Bundle Shell Total	tonnes tonnes tonnes	2.76 5.18 8.0		EXCHANGER TYPE : BEU

TOTAL

TEP/DOP/DIP/EXP/SUR

BY | CHK

PROCESS CALCULATION SHEET

SHELL AND TUBE HEAT EXCHANGER

ITEM : FEED/PRODUCT EXCHNGR

NO : E 1234

DATE 4/85

JOB TITLE : DESIGN MANUAL

JOB N° : EXAMPLE

REV

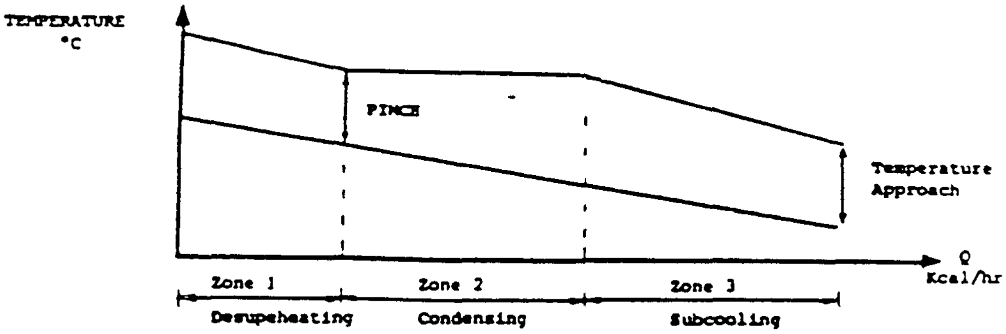
(1) Use following formula

$$LMTD = \frac{(T2 - t1) - (T1 - t2)}{Ln \frac{T2 - t1}{T1 - t2}} \qquad \text{if } T2 - t1 > T1 - t2$$

$$LMTD = \frac{(T1 - t2) - (T2 - t1)}{Ln \frac{T1 - t2}{T2 - t1}} \qquad \text{if } T1 - t2 > T2 - t1$$

Remark : If the heat exchange curves are not linear the LMTD should be determined step by step with the linearisation of the curves and with the ponderation of the partial LMTD by the partial duty on each linear step.

(2) For total condensing



In this case calculate the heat transfer area for each zone, the sum of these areas is the surface for the exchanger.

(3) See LMTD correction factor (Figures 2)

the number of shell and tubes passes should be chosen in order to have 1 < F < 0.8

If F < 0.8 add shells (2 exchangers in series)

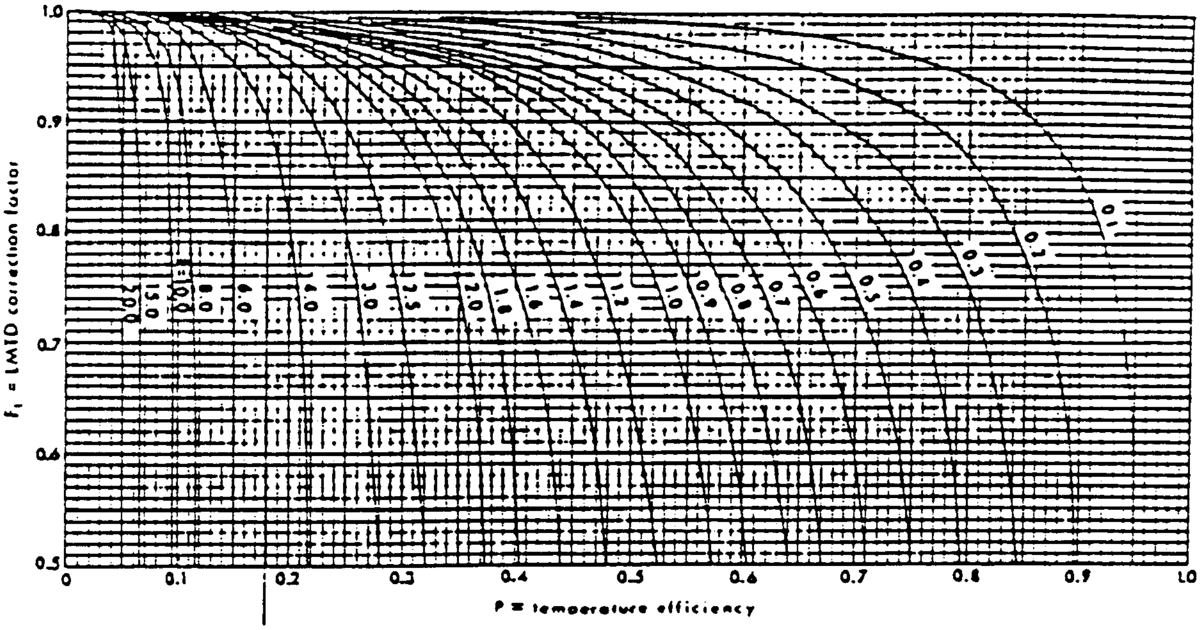
(4) See Section 5 shell and tube heat transfer.

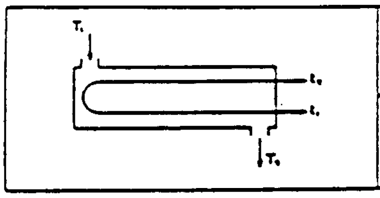
4. ESTIMATION OF SHELL DIAMETER

With the heat transfer area, selected tubes size, pitch, tubes length it is possible to determine the number of tubes and with table 1 or 2 hereafter the approximate shell diameter.

Take maximum shell diameter about 60 inches.

FIGURE 2

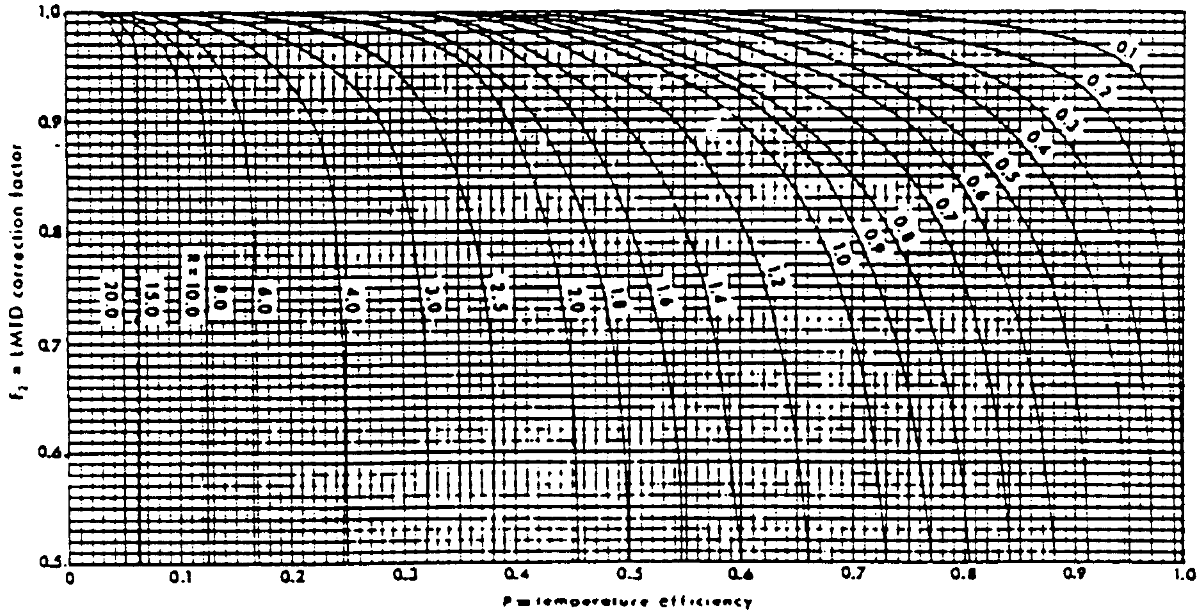


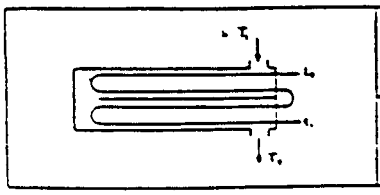


LMTD correction factor

1 shell pass 2 or more tube passes

$P = \frac{t_2 - t_1}{T_1 - t_1}$ $R = \frac{T_1 - T_2}{t_2 - t_1}$ CMTD=(LMTD)(F_1)



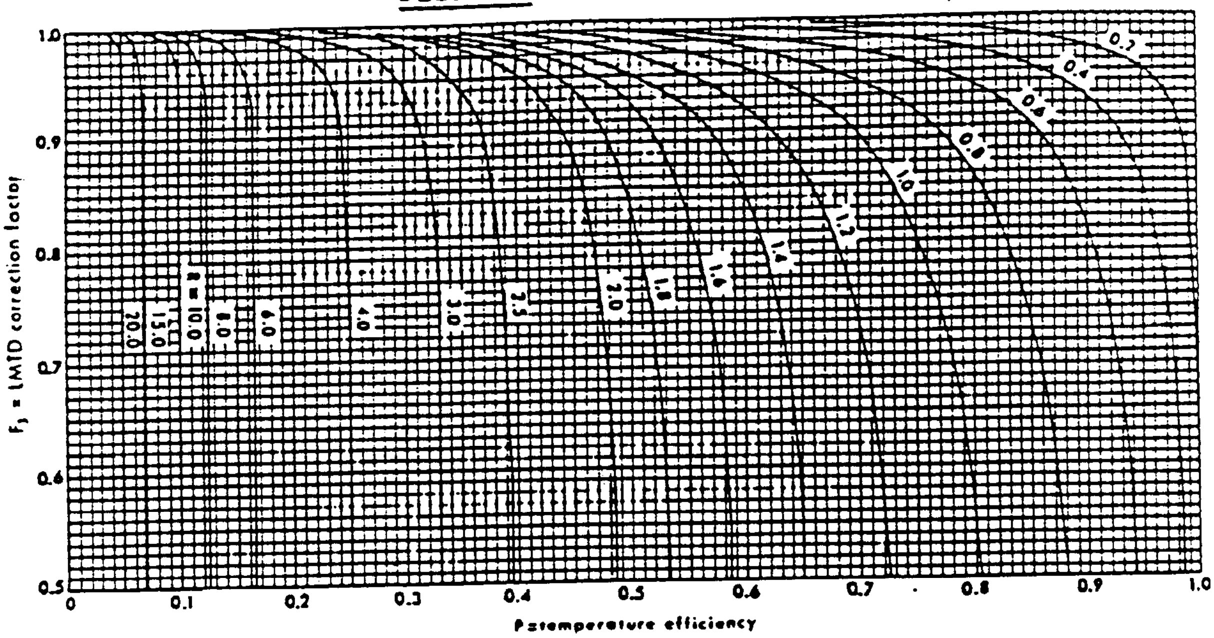


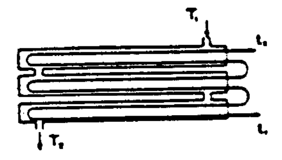
LMTD correction factor

2 shell passes 4 or more tube passes

$P = \frac{t_2 - t_1}{T_1 - t_1}$ $R = \frac{T_1 - T_2}{t_2 - t_1}$ CMTD=(LMTD)(F_2)

FIGURE 2

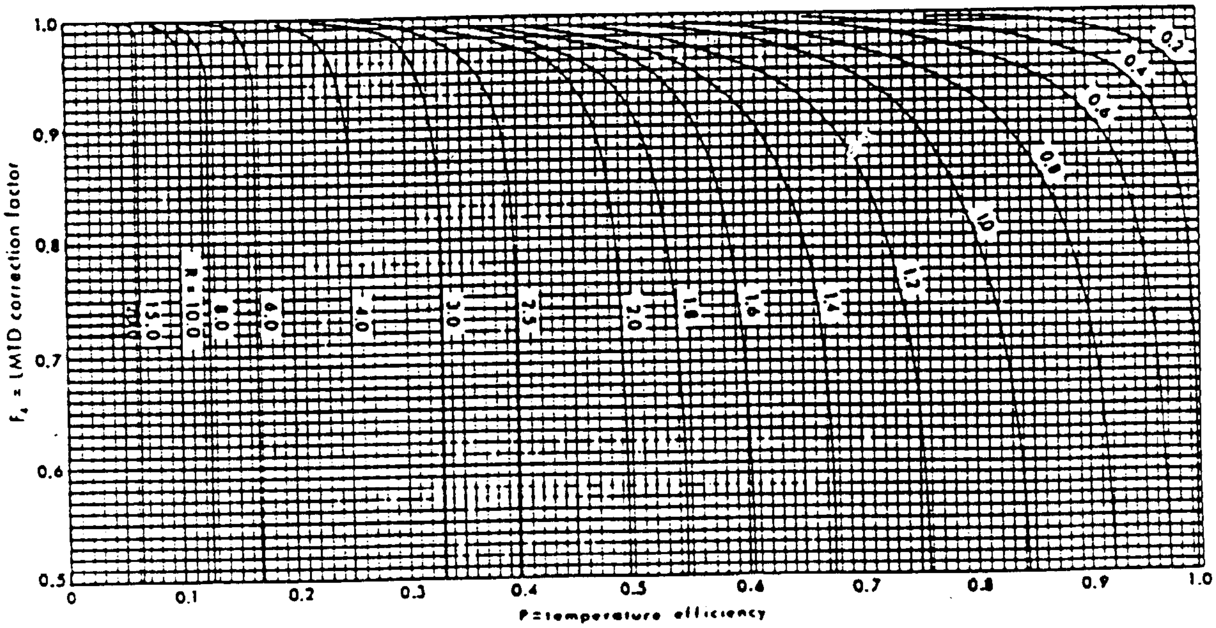


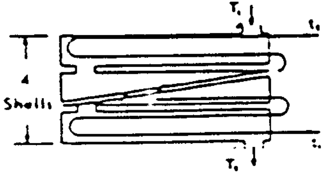


LMTD correction factor

3 shell passes — 6 or more tube passes

$P = \frac{t_2 - t_1}{T_1 - t_1}$ $R = \frac{T_1 - T_2}{t_2 - t_1}$ CMTD=(LMTD)(F_3)





LMTD correction factor

4 shell passes — 8 or more tube passes

$P = \frac{t_2 - t_1}{T_1 - t_1}$ $R = \frac{T_1 - T_2}{t_2 - t_1}$ CMTD=(LMTD)(F_4)

SHELL ⌀i in inches	TUBES ⌀e 3/4"					
	PITCH □ 1"			PITCH △ 1"		
	NUMBER OF PASSES TUBE SIDE			NUMBER OF PASSES TUBE SIDE		
	1	2	4	1	2	4
8	32	26	20	36	32	21
10	52	52	40	60	56	47
12	82	76	68	90	82	76
13 1/4	95	90	82	109	104	90
15 1/4	137	124	116	164	150	137
17 1/4	188	166	158	211	200	183
19 1/4	236	220	204	274	254	241
21 1/4	274	270	246	320	306	279
23 1/4	325	318	308	383	375	350
25	407	394	370	471	452	419
27	465	460	432	556	534	487
29	551	526	480	630	603	556
31	633	616	589	745	726	677
33	740	712	685	855	830	772
35	825	812	770	967	937	882
37	928	901	880	1074	1047	1014
39	1024	1014	983	1206	1173	1129
42	1202	1168	1147	1406	1315	1310
45	1435	1411	1367	1639	1611	1543
48	1620	1598	1553	1872	1845	1766
52	1918	1890	1848	2212	2183	2092
56	2241	2214	2167	2568	2545	2446
60	2587	2556	2510	2987	2945	2827

TABLE 1

MAXIMUM TUBES NUMBER PER SHELL

SHELL ⌀i in inches	TUBES 1"					
	PITCH □ 1" 1/4			PITCH △ 1" 1/4		
	NUMBER OF PASSES TUBE SIDE			NUMBEF OF PASSES TUBE SIDE		
	1	2	4	1	2	4
8	20	16	16	20	16	16
10	30	30	26	37	30	26
12	47	45	41	57	51	47
13 1/4	61	55	51	67	65	57
15 1/4	88	82	76	96	92	86
17 1/4	113	108	102	129	127	117
19 1/4	148	139	137	170	160	150
21 1/4	170	166	158	199	189	179
23 1/4	207	197	197	246	232	215
25	248	246	222	294	281	257
27	287	287	267	349	335	302
29	349	339	320	396	376	539
31	390	390	365	472	456	431
33	458	452	431	538	520	487
35	526	513	483	610	592	561
37	577	557	555	674	664	633
39	643	637	618	766	735	698
42	746	729	709	900	-	-
45	894	875	853	1038	1013	982
48	1029	1010	975	1188	1163	1098
52	1216	1196	1167	1405	1375	1323
56	1420	1400	1371	1638	1605	1549
60	1639	1615	1587	1889	1851	1797

TABLE 2

MAXIMUM TUBES NUMBER PER SHELL

5. SHELL AND TUBE OVERALL HEAT TRANSFER

Approximate overall heat transfer coefficients.
= U
Kcal/hr m² °C
(including fouling factors)

a. Heating/cooling

	U
Water/gas 1 - 35 bars	170 - 250
Water/gas 35 - 70 bars	250 - 390
Water/gas 70 - 100 bars	390 - 500
Water/gas over 100 bars	500 - 700
Water/natural gasoline	340 - 440
Water/MEA	630 - 730
Water/air	70 - 120
Water/Water	800 - 1 000
Gas/gas (< 35 bars)	240 - 340
Gas/gas (about 70 bars)	270 - 360
Gas/C3 chiller	290 - 440
Water/light H.C. viscosity < 0.5 Cpo	290 - 730
Water/average H.C 0.5 Cpo < Viscosity < 1 Cpo	250 - 610
Water/heavy H.C. viscosity > 1 Cpo	50 - 500
Oil/oil	300 - 450
Dowtherm/gas	20 - 200
Dowtherm/heavy oil	30 - 300
Steam/water	540 - 1 100
Steam/gas	25 - 250
Steam/light H.C. Viscosity < 0.5 Cpo	500 - 1 000
Steam/average H.C. 0.5 Cpo < viscosity < 1 Cpo	250 - 500
Steam/heavy H.C. viscosity > 1 Cpo	30 - 300

$$\text{kcal/hrm}^2\,{}^\circ\text{C} \times 0.2047 = \text{BTU/hrft}^2\,{}^\circ\text{F}$$
$$\times 1.162 = \text{W/m}^2\,{}^\circ\text{K}$$

b. Condensing

Steam/water	700 - 1 700
Light H.C./water	420 - 660
Fractionator overhead/water	340 - 390
Steam/oil	340 - 540
Gasoline/water	320 - 630
Heavy H.C./water	190 - 370

c. Vaporization (reboilers)

Hydrocarbons light/steam	440 - 900
Hydrocarbons C4-C8/steam	360 - 720
With hot oil	440 - 590

6. WEIGHT ESTIMATES

See figures 3 and 4 hereafter.

7. REFERENCES AND USEFUL LITERATURE

4.1. TEMA (standards of tubular exchanger manufacturers association)
4.2. KERN
4.3. LUDWIG VOLUME 3 second edition
4.4. CAMPBELL VOLUME 1
4.5. NGPSA Chapter 8
4.6. PERRY
4.7. HTRI Program
4.8. HTFS Program
4.9. QUICK CALCULATION OF HEAT EXCHANGER WEIGHT Process Engineering January 1980

FIGURE 3

TYPE AES
WEIGHT ESTIMATE

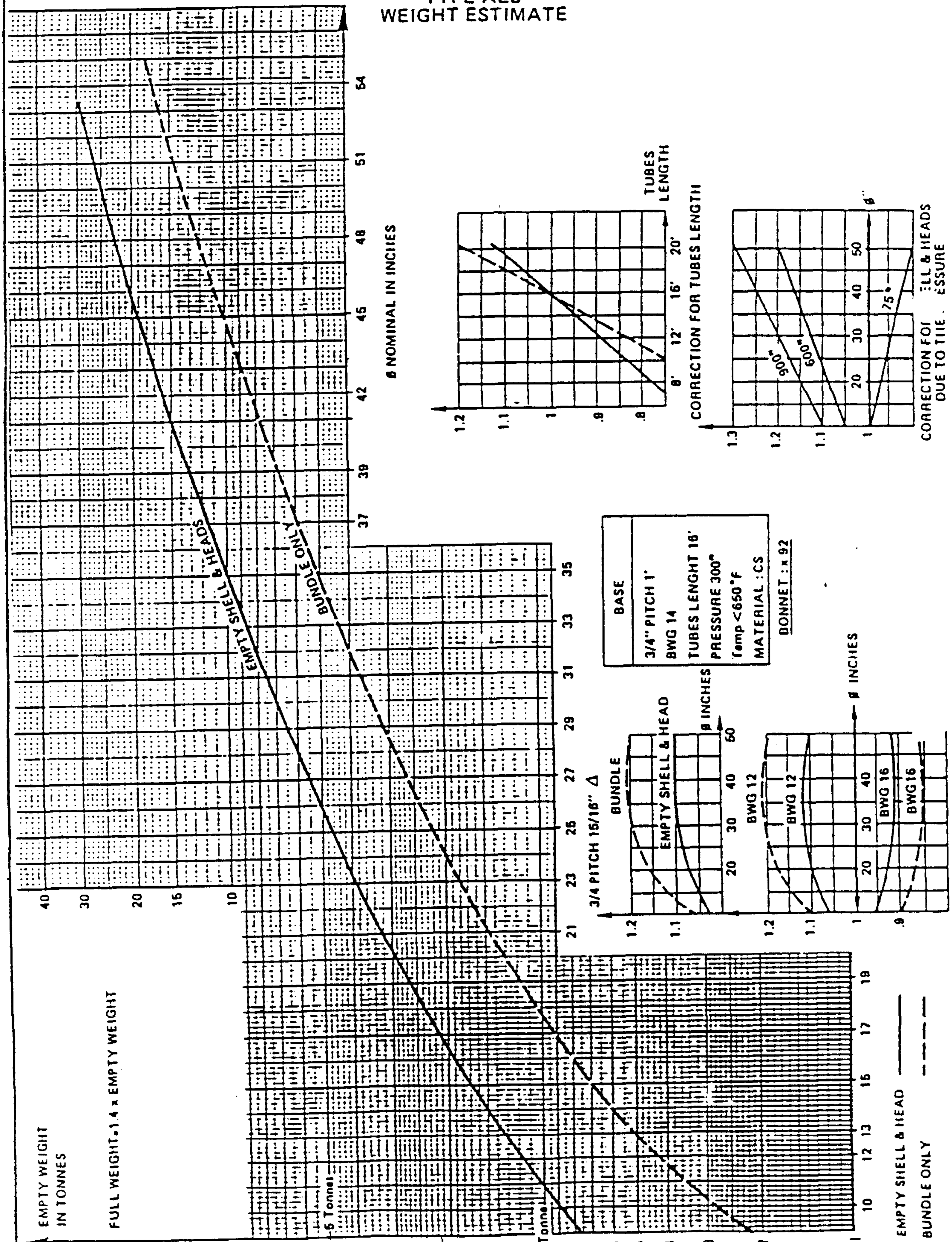
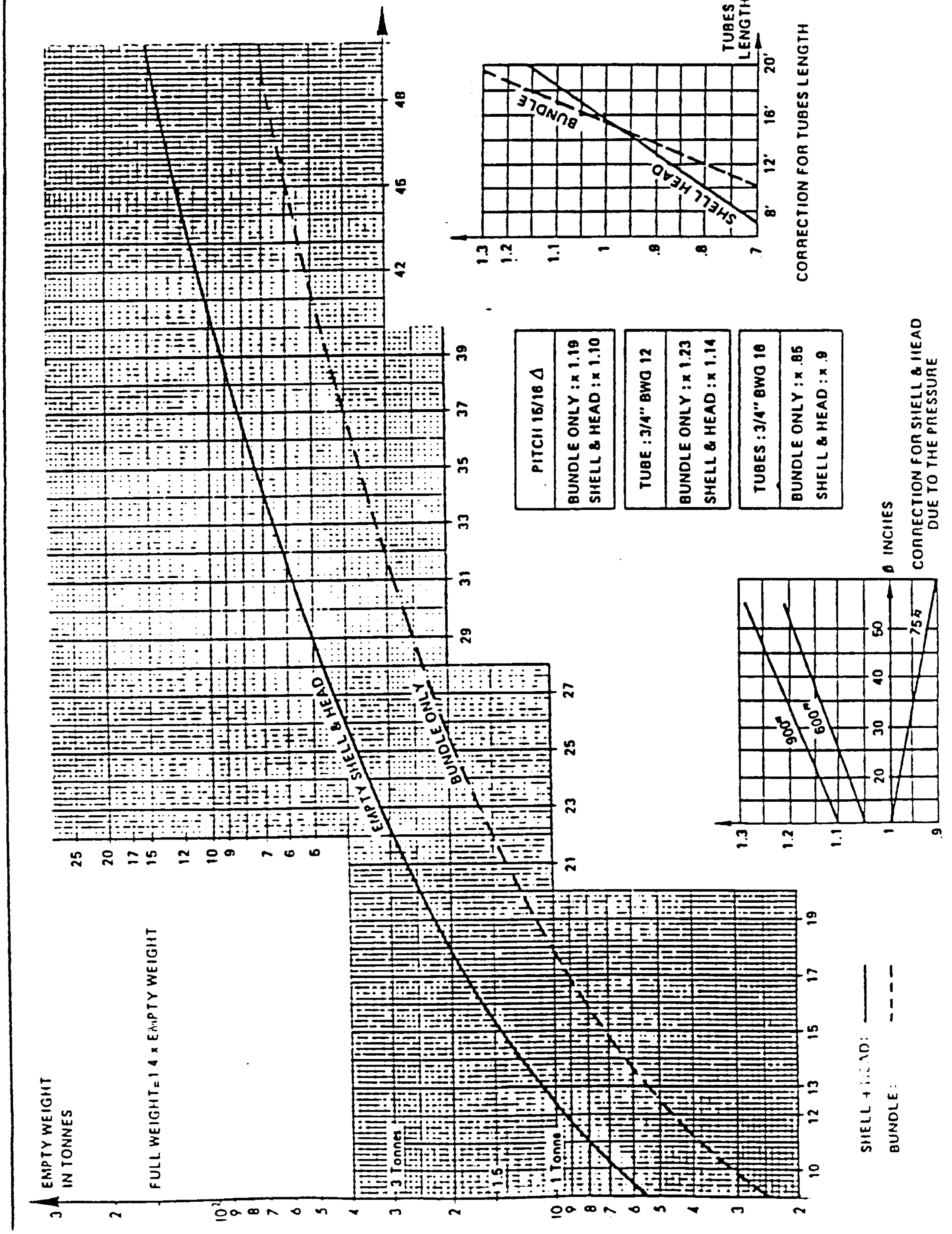


FIGURE 4

TYPE BEU
WEIGHT ESTIMATE

Base : 3/4" BWG 14 PITCH : 1" ◻ TUBES LENGTH 16'
PRESSURE 300# Temp ≤ 650 °C
MATERIAL : CS



1. APPLICABILITY

For both the feasibility and preproject study it would generally be required to state the required duty of the air cooler, the overall dimensions and weight and an estimate o required fan power.

A calculation procedure sufficient for a preliminary estimate is given in section 3.0.

2. DESCRIPTION AND GUIDELINE NOTES

Water or Air Cooling ?

- Air cooling offshore is sometimes prohibited due to the modular layout of the platf   . This may require installation of the air cooler too remote from the associated equipment. Use closed loop water cooling.
- Air cooling is cheaper, simple and flexible when compared to water cooling. The cost and nuisance of water treating is eliminated if air coolers are used.
- In warm climates air cooling will not be as effective as water which will produce a cooler product stream. Air cooling is approx 50-70 % as effective as water.

Forced on induced draft ?

- Forced draft pushes the air at lowest available temperature (highest ρ) hence lower power requirement.
- Accessability to motor and driver are better on forced. Structural and main-tain costs lower.
- Possibility with forced draft of hot air recirculating into suction of fan thereby reducing efficiency.
- Induced draft gives better air distribution due to lower inlet velocity with less chance of recirculating of hot air.
- Induced draft coolers can be easily installed above piperacks or other equipment. Protection is given by induced draft coolers from effects of rain, wind snow on finned tubes. Important if fluid in tubes is sensitive to sudden temp change also freezing of tubes can occur in cold climates or heavy snowfall.

Finned tube elements (see Table 2)

- 1" OD tubing is most common with 0.5" to 0.625" fins. Fin spacing 7 to 11 per inch. Extended surface area is 7 to 20 times bare area.
- Standard tube lengths from 6 ft to 50 ft (2 m to 15 m). Longer tube designs are less costly than short ones.
- Bundle depth may vary from 3 rows to 30 rows of tubes. 4 or 6 rows is common for smaller units. Use 4 as first estimate.
- Fin material most commonly AL. Adequate upto 400 °C operating . Use steel for higher temperatures.

Fans and motors

- Fans are axial-flow large volume low DP devices. Use total fan efficiency 65 %. Driver efficiency 95 %.
- Fan Ø equal to or slightly less than bundle width. Normally 2 fans preferred. Fans have 4 to 6 blades. Max fan diameter 14'-16'.
- Distance between fan + bundle 0.4-0.5 of fan diameter. Ratio of fan ring area to bundle area must not be less than 0.4.
- Fans may be electric, steam, hydraulic or gasoline driven. Individual driver size usually limited to 50 hp, (40 kw), 380 V.
- Face velocity of air across a bundle is 300-700 ft/min (1.5-3.6 ms^{-1}).
- A 10 % change in air flow rate results in ~35 % change in power used.

Temperature control

- For close control of process outlet temperature auto-variable pitch fans, top louvers or variable speed motors are required.
- Variable pitch fans more efficient than louvers.
- Louvers can be manually adjusted for winter or night time operation.
- For process fluids that freeze or gel at temperatures above the winter ambient a recirculation system is necessary to maintain air temp entering the tube bundle.
- General approach temp to ambient air is 20-28 °C. Absolute min is 10-12 °C.

Note : Air coolers are noisy. Keep fan speed as low as possible and consider relative layout carefully.

4.0 REFERENCES AND USEFUL LITERATURE

4.1.	Air cooled heat exchangers	PERRY	pp 11.23 > 11.25
4.2.	Air cooled heat exchangers	LUDWIG	pp 177 > 193
4.3.	Air cooled heat exchangers	GPSA	chapter 9
4.4.	Aerial coolers	CAMPBELL	pp 207-209
4.5.	Design of air coolers - A Procedure for estimation	R. BOWN Chem. Eng, Mar 27 1978, p 109	
4.6.	Estimate air cooler size HP 41CV program	N. SHAIKH Chem. Eng, Dec 12 1983, p 65-70	

OPERATING CONDITIONS AND NATURE OF FLUID : HYDROCARBON GAS COOLER

Duty	Q =	2×10^6	kcal/hr	
Fluid inlet temperature	T1 =	100	°C	
Fluid outlet temperature	T2 =	50	°C	FLUID T_f = T1 - T2 = 50 °C
Fluid inlet pressure	P =	10	bar abs	
Air ambiant temperature	t1 =	30	°C	INLET Tt = T1 - t1 = 70 °C
Overall heat transfer coeff. (See Table 1 and/or attached work sheet) (Based on bare tube area)	U =	200	kcal/hr m^2 °C	NOTES

STEP

1. Optimum number of tube rows	N =	8		(curve N° 4)
2. R = Δt air/Δt m	R =	0.8		(curve N° 4)
T1 - T2/T1 - t1		0.714	°C	
4. Y = Δt air/T1 - t1	Y =	0.35		(curve N° 5)
5. Δtair = Y x (T1 - t1)	Δtair	24.5	°C	
6. Exit air temp t2 = Δtair + t1	t2 =	54.5	°C	
7. Average differential temp. $\Delta tm = \frac{\Delta tair}{R}$	tm =	30.6	°C	
8. Bare tube surface $A = \frac{Q}{U \times \Delta tm}$	A =	326	m^2	
9. Bare tube area/row Fa=A/N	Fa =	41	m^2	
10. Tube length	L =	7.5	M	3, 4, 5, 6, 7.5 or 9 m are common
11. Tubes/row TR = Fa/Lx0.08	TR =	68		(1" OD tubing)
12. Cooler width W=TRx0.0635	W	4.3	m	
13. Total fan power =Fax0.795	Fp	32.4	kW	
14. Number of fans	N_F	2		max. fan diam = 4.6 m
15. Fan diameter	F_D	3.5	M	
16. Power/fan Fp/N_F	P_F	16.2	kW	
17. Estimated weight 4.88 (36.4X9.35 N)xWxL	M	17500	kg	(including motors)

Notes : Curve numbers refer to Process Design Manual Chap. 4.

TOTAL
EP/DOP/DIP/EXP/TUR

PROCESS CALCULATION SHEET

AIR COOLER

ITEM : EXAMPLE
NO :

BY | CHK | DATE | JOB TITLE : | JOB N° : | REV

1. LIQUID COOLING

LIQUID VISCOSITY AT $\frac{T1 + T2}{2}$ = Cp

GLOBAL HEAT TRANSFER COEFFICIENT : U = kcal/hr m² °C
(Read curve n° 1)

2. GAS COOLING

MOLECULAR MASS : MW =

GLOBAL HEAT TRANSFER COEFFICIENT : U = kcal/hr m² °C
(Read curve n° 2)

3. TOTAL CONDENSATION

T1 - T2 = °C

GLOBAL HEAT TRANSFER COEFFICIENT : U = kcal/hr m² °C
(Reat curve n° 3)

4. PARTIAL CONDENSATION

4.1. WITHOUT LIQUID AT INLET

inlet gas flowrate WG1 = kg/hr

outlet gas flowrate WG2 = kg/hr

outlet liq flowrate WL2 = kg/hr

T1 - T2 = °C

GAS MOLECULAR WEIGHT AT $\frac{T1 + T2}{2}$ =

HEAT TRANSFER COEFF. Uc = kcal/hr m² °C
(Read curve n° 3)

HEAT TRANSFER COEFF. Ug = kcal/hr m² °C
(Read curve n°.2)

GLOBAL HEAT TRANSFER COEFF.

$U = \frac{WL2}{WG1} \times Uc = \frac{WG2}{WG1} \times Ug$ = kcal/hr m² °C

SELECTED GLOBAL HEAT
TRANSFER COEFF. : U = kcal/hr m² °C

Curves refer to PDM Chptr. 4.

TOTAL
P/DOP/DIP/EXP/SUR

PROCESS CALCULATION SHEET

AIR COOLERS
HEAT TRANSFER COEFFICIENT

ITEM :

NO :

CHK

DATE

JOB TITLE :

JOB N° :

REV

4.2. WITH LIQUID AT INLET

inlet liquid flow rate $WL1$ = kg/hr

outlet liquid flow rate $WL2$ = kg/hr

LIQUID MOLECULAR WEIGHT AT $\frac{T1 + T2}{2}$ =

LIQUID SPECIFIC HEAT AT $\frac{T1 + T2}{2}$ CPl = kcal/kg °C

$QL = \frac{(WL1 + WL2)}{2} \times CPl \times (T1 - T2)$ = kcal/hr

inlet gas flow rate $WG1$ = kg/hr

outlet gas flow rate $WG2$ = kg/hr

GAS MOLECULAR WEIGHT AT $\frac{T1 + T2}{2}$ =

GAS SPECIFIC HEAT AT $\frac{T1 + T2}{2}$ CPg = kcal/kg °C

$QG = \frac{(WG1 + WG2)}{2} \times CPg \times (T1 - T2)$ = kcal/hr

CONDENSATION HEAT

$Qc = Q - QL - QG$ = kcal/hr

LIQUID VISCOSITY AT $\frac{T1 + T2}{2}$ = CPg

LIQUID HEAT TRANSFER COEFF. U = kcal/hr m² °C
(Read curve n° 2)

GAS HEAT TRANSFER COEFF. Ug = kcal/hr m² °C
(Read curve n° 2)

CONDENSATION HEAT TRANSFER COEFF. Uc = kcal/hr m² °C
(Read curve n° 3)

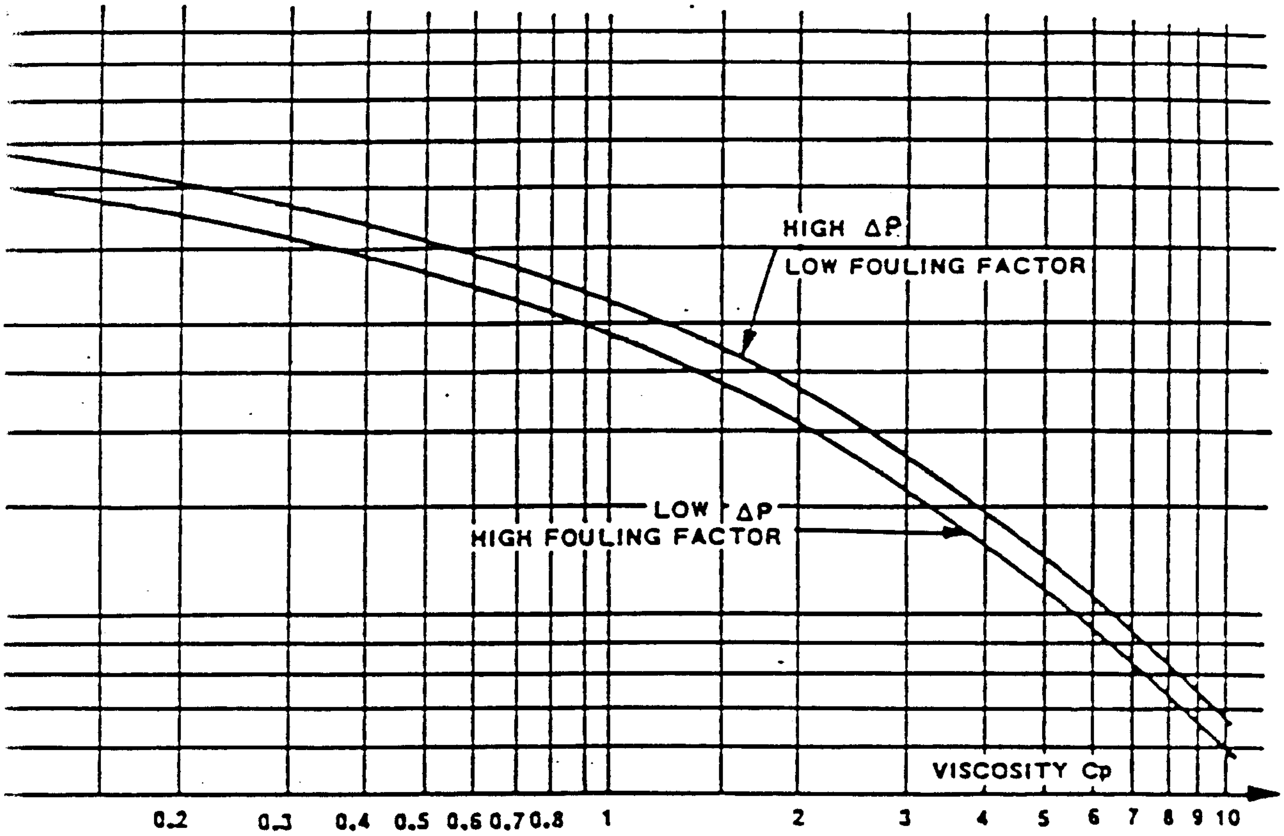
GLOBAL HEAT TRANSFER COEFF.

$$U = \frac{Q}{\frac{QL}{u_l} + \frac{QG}{u_g} + \frac{QC}{u_c}} \qquad U = \quad \text{kcal/hr m}^2\ °\text{C}$$

SELECTED GLOBAL HEAT TRANSFER COEFF. : U = kcal/hr m² °C

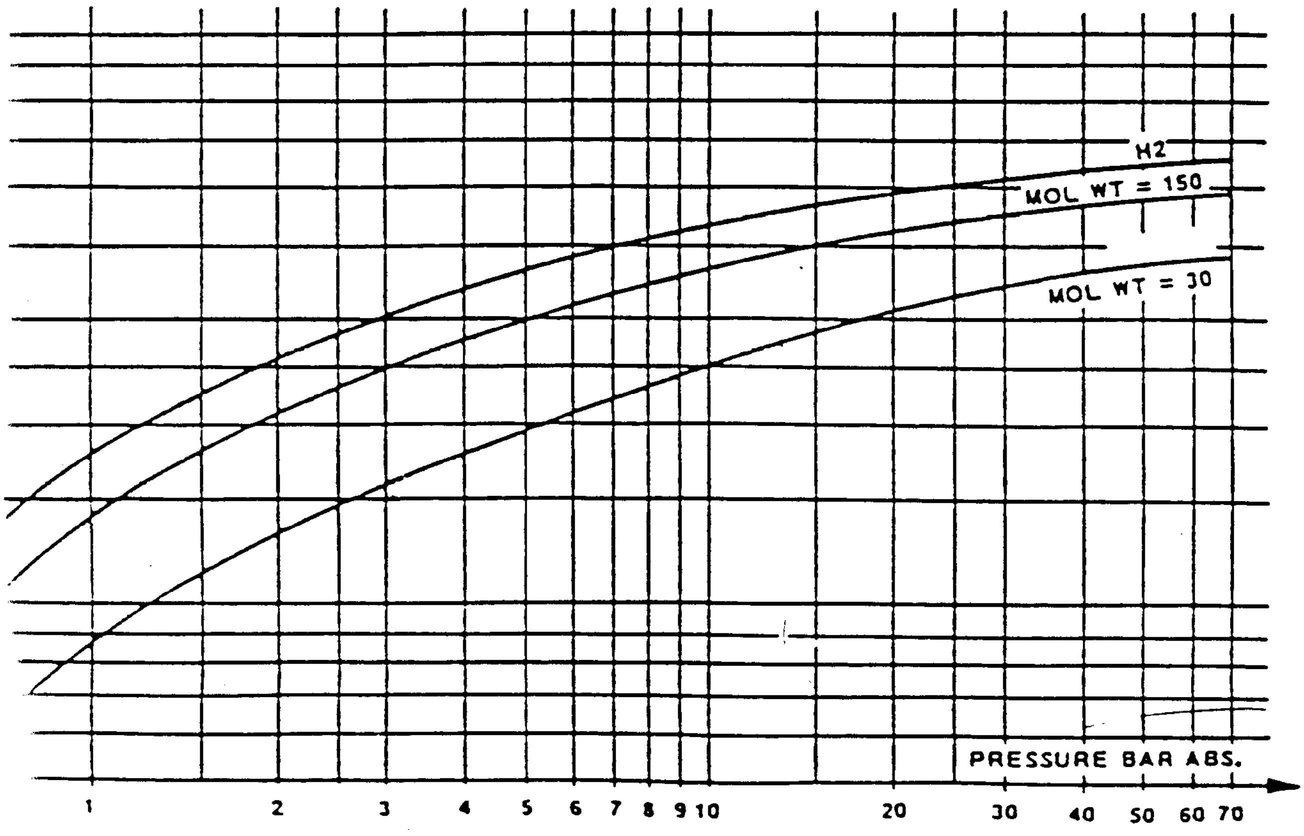
TOTAL TEP/DGP/DIP/EXP/SUR	PROCESS CALCULATION SHEET	
	AIR COOLERS HEAT TRANSFER COEFFICIENT	ITEM : NO :
BY CHK	DATE JOB TITLE :	JOB N° : REV

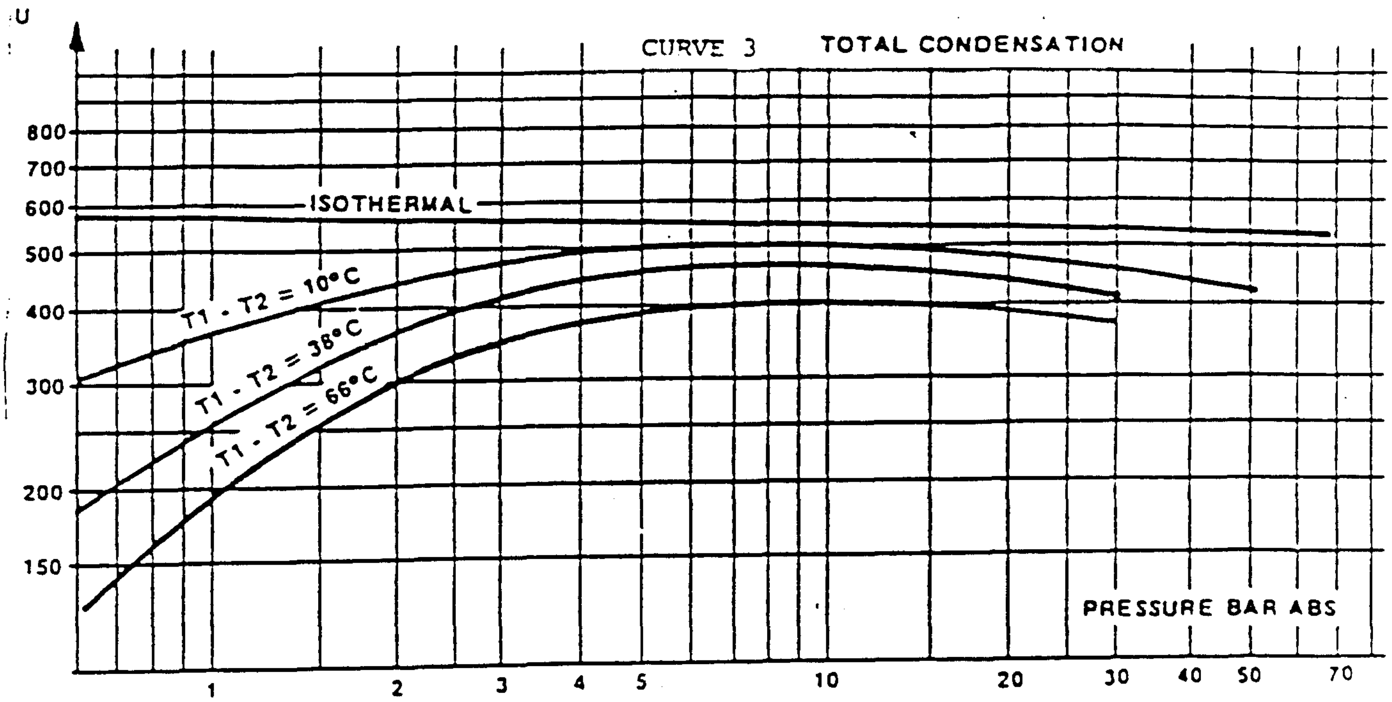
CURVE 1 - COOLING HYDROCARBON LIQUIDS



U in kcal/hr.m^2 °C

CURVE 2 - COOLING GASES





BTU/hrft^2°F x 4.885 = kcal/hrm^2

TABLE 2

Fintube data for 1-in. OD tubes

Fin height by Fins/inch	½ in. by 9		⅝ in. by 10	
APF, sq ft/ft	3.80		5.58	
AR, sq ft/ft	14.5		21.4	
Tube Pitch	2 in. Δ	2¼ in. Δ	2¼ in. Δ	2½ in. Δ
APSF (3 rows)	68.4	60.6	89.1	80.4
(4 rows)	91.2	80.8	118.8	107.2
(5 rows)	114.0	101.0	148.5	134.0
(6 rows)	136.8	121.2	178.2	160.8

Notes: APF is total external area/ft of fintube in sq ft/ft. AR is the area ratio of fintube compared to the exterior area of 1 in. OD bare tube which has 0.262 sq ft/ft. APSF is the external area in sq ft/ft of bundle face area.

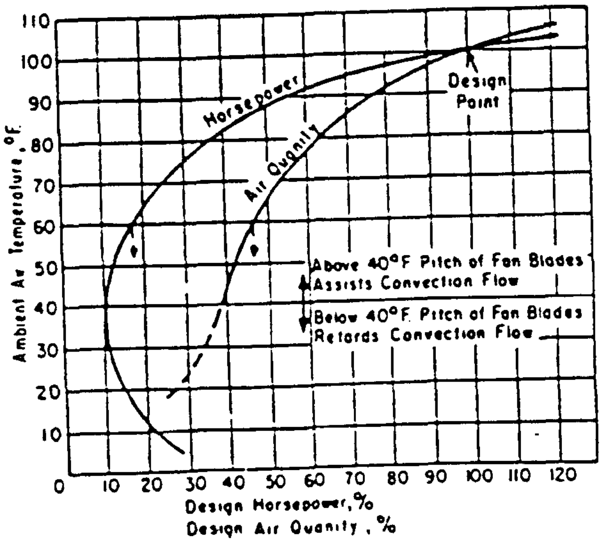


Figure 1 Temperature control and horsepower savings with automatic variable pitch fans.

TABLE 1

BTU/hrft^2°F

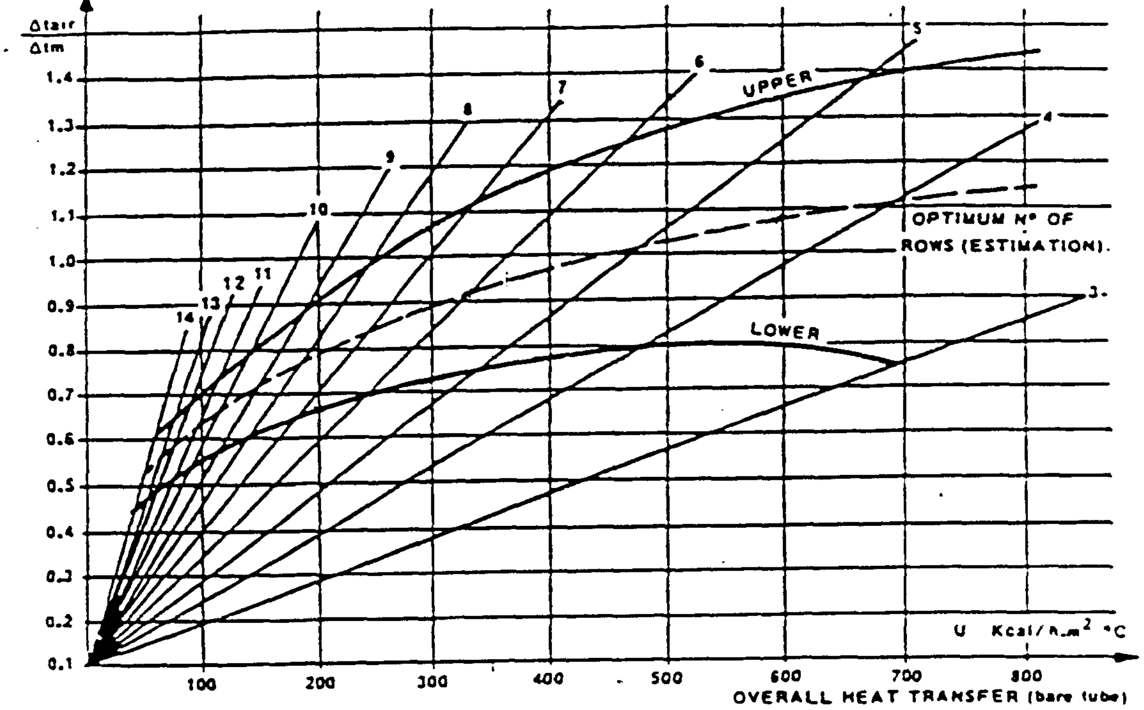
Typical overall heat-transfer coefficients for air coolers

Service	1 in. Fintubes	
	½ in. by 9	⅝ in. by 10
1. Water & water solutions	U_b U_x	U_b U_x
Engine jacket water (r_d = 0.001)	110—7.5	130—6.1
Process water (r_d = 0.002)	95—6.5	110—5.2
50–50 ethylene glycol-water (r_d = 0.001)	90—6.2	105—4.9
50–50 ethylene glycol-water (r_d = 0.002)	80—5.5	95—4.4
2. Hydrocarbon liquid coolers		
Viscosity, cp, at avg. temp	U_b U_x	U_b U_x
0.2	85—5.9	100—4.7
0.5	75—5.2	90—4.2
1.0	65—4.5	75—3.5
2.5	45—3.1	55—2.6
4.0	30—2.1	35—1.6
6.0	20—1.4	25—1.2
10.0	10—0.7	13—0.6
3. Hydrocarbon gas coolers		
Pressure, psig	U_b U_x	U_b U_x
50	30—2.1	35—1.6
100	35—2.4	40—1.9
300	45—3.1	55—2.6
500	55—3.8	65—3.0
750	65—4.5	75—3.5
1000	75—5.2	90—4.2
4. Air and flue-gas coolers		
Use one-half of value given for hydrocarbon gas coolers		
5. Steam condensers		
(Atmospheric pressure & above)	U_b U_x	U_b U_x
Pure steam (r_d = 0.0005)	125—8.6	145—6.8
Steam with non-condensables	60—4.1	70—3.3
6. HC condensers		
*Cooling Range, °F	U_b U_x	U_b U_x
0° range	85—5.9	100—4.7
10° range	80—5.5	95—4.4
25° range	75—5.2	90—4.2
60° range	65—4.5	75—3.5
100° & over range	60—4.1	70—3.3
7. Other condensers	U_b U_x	U_b U_x
Ammonia	110—7.6	130—6.1
Freon 12	65—4.5	75—3.5

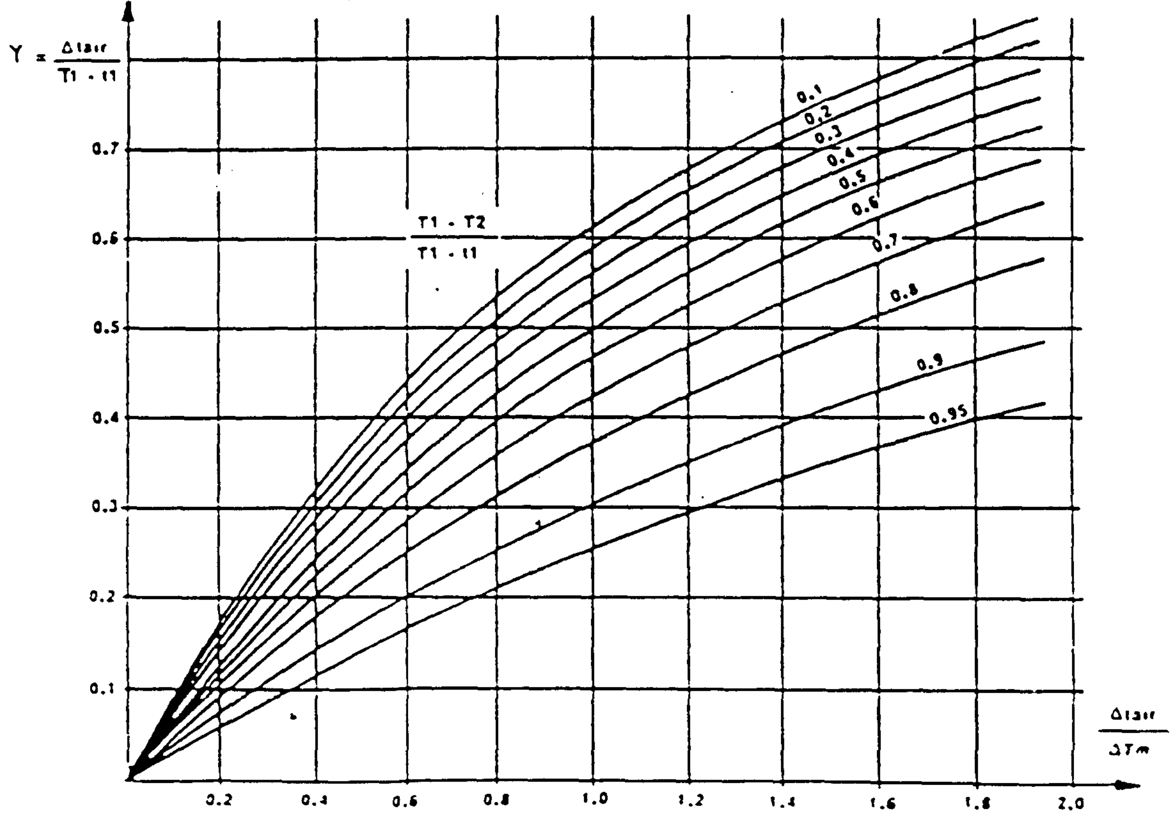
Note: U_b is overall rate based on bare tube area, and U_x is overall rate based on extended surface

*Cooling range = hydrocarbon inlet temperature minus hydrocarbon outlet temperature

CURVE 4



CURVE 5



1. APPLICABILITY

FEASIBILITY STUDY : PRE-PROJECT

Under normal circumstances, the design of plate type exchangers would be detailed by a vendor based on process data supplied by the engineer.

Two types of plate exchangers could be used :

- Plate fin exchangers ;
- Plate exchangers.

For the purpose of this design guide, only a quick description and some characteristics are given.

For plate fin exchangers, the size could be done only by a vendor.

For plate exchangers, the size could be estimated if some vendor (ALFA-LAVAL, APV, VICARB) information are available.

An estimation of the heat transfer area could be done if the heat transfer coefficient is known using the same formula as for shell and tube heat exchanger with a LMTD correction factor = 1. The heat transfer coefficient is difficult to estimate ; it depends on many factors as flow rate of different fluids, pressure drop, plate spacing, ect...

2. DESCRIPTION AND NOTES

2.1 PLATE FIN EXCHANGERS

These exchangers consist of stacked corrugated sheets (fins) separated by flat plates and an outer frame with openings for the inlet and outlet of fluids. This core is immersed in a liquid salt bath to braze all the separate parts together.

Flow in adjacent fluid passages can be cocurrent, counter current, or crossflow and several fluids can be exchanging heat at the same time.

In case of the inlet fluid is a two phases flow a drum is required to separate the two phases in order to have a good distribution. Not necessarily according to Marston Palmer. They normally put in an aluminium perforated plate to ensure good gas distribution.

These plate fin exchangers are used only with clean fluids.

Figure 1 shows the principle of construction of a platefin exchanger. A large amount of surface can be accomodated in a small volume (1,000 m2/m3).

Maximum design pressure : 54 barg (some recent models claim 80 barg)
Temperature range : - 195 °C to + 65 °C
Size max. : 1,220 mm x 6 096 mm x 1 340 mm
Temperature approach : 2 °C
Applicability : LNG, LPG recovery, ...

Pressure drop as for shell and tube heat exchangers.

2.2 PLATE EXCHANGERS

Plate exchangers are an assembly of metal plates separated by gaskets to give a small clearance between each plate. The two fluids pass in opposite directions each through every alternate plate. Refer to figure 2.

The exchanger is easily dismantled for cleaning if required. A good overall heat transfer coefficient is obtained and small temperature differences can be used.

The plates can be made from exotic materials such as titanium which are resistant to corrosion and are used for sea water coolers. They are very compact exchangers and occupy a small floor area.

Maximum pressure : 10 - 20 bars
Maximum temperature : 250 °C
(Need special gaskets)
Overall heat transfer coefficient
Water/water : 2 000 - 5 000 Kcal/hr m^2 °C
Maximum surface : about 1 500 m^2
Maximum flow : 2 500 m^3/hr

Applicability : Sea water - service water, water-TEG, TEG-TEG, ...

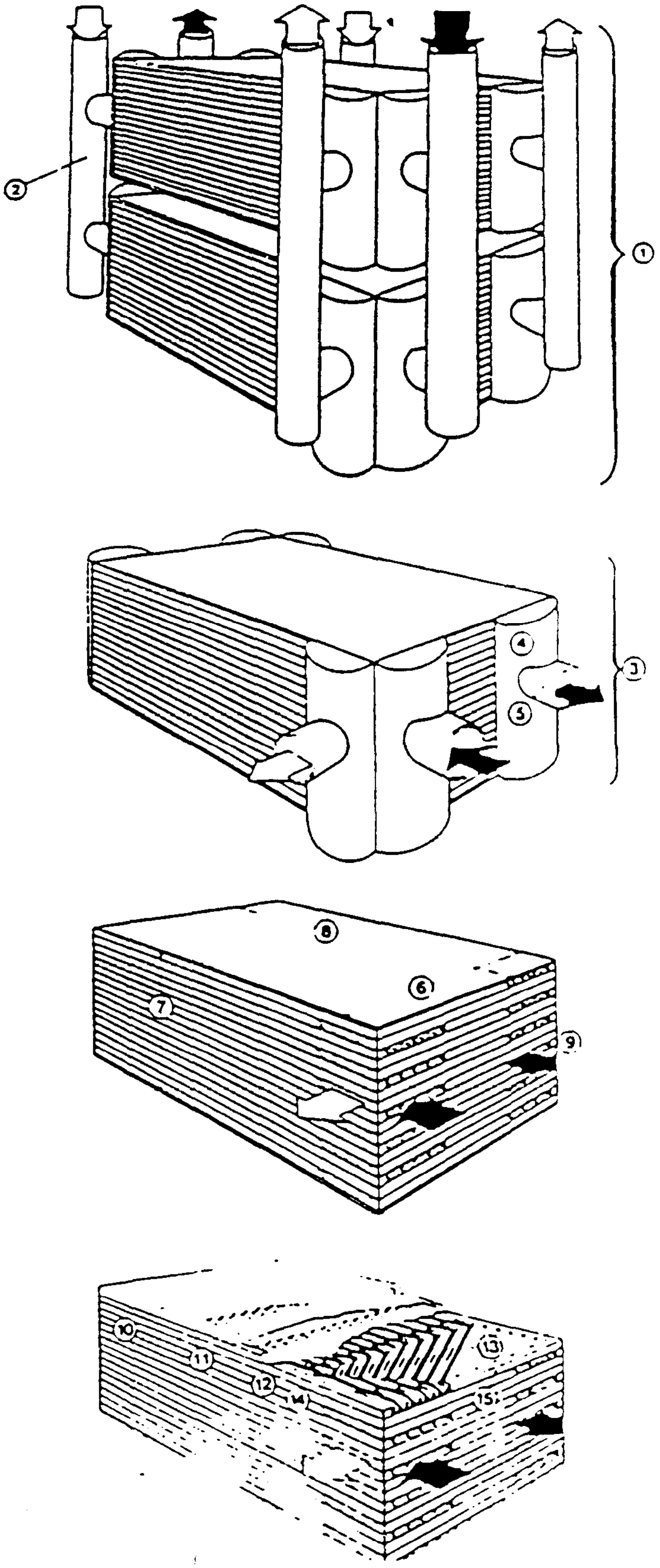
- Pressure drops : allowable pressure drops vary according to the total system pressure and the service of the fluids.
  - for sea water - service water : 0.5 to 2 bar (high ΔP increase the overall heat transfer coefficient),
  - for water-TEG or TEG-TEG the ΔP could be very low such as 10 to 20 mbar.

REFERENCES AND USEFUL LITERATURE

Vendors information.

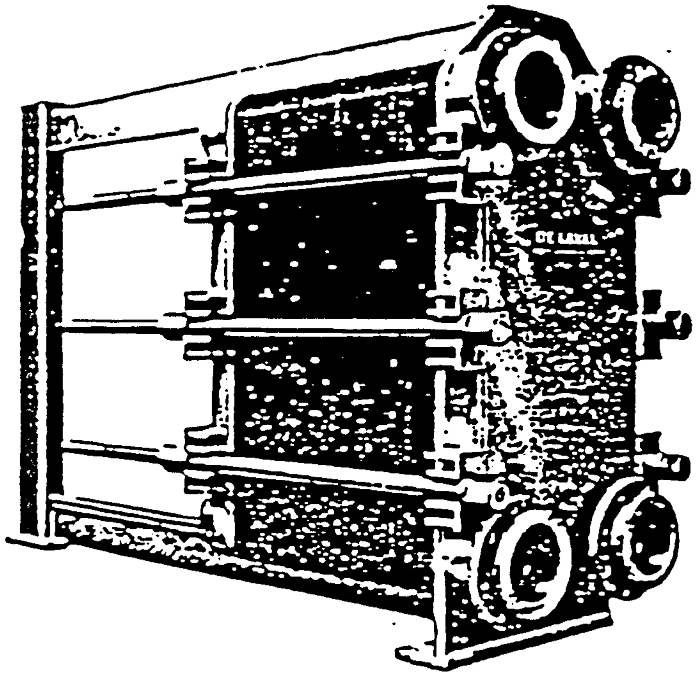
PLATE FIN EXCHANGERS

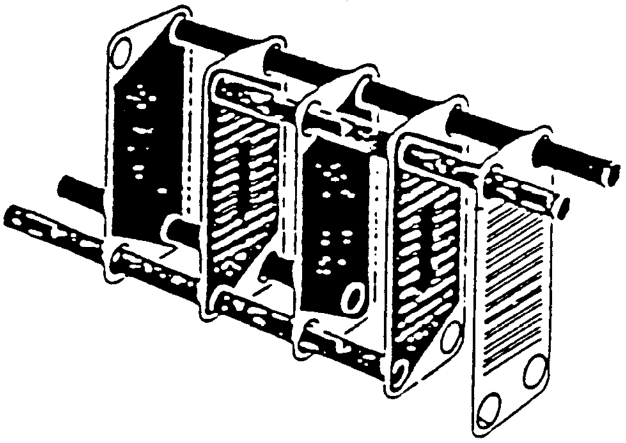
FIGURE 1

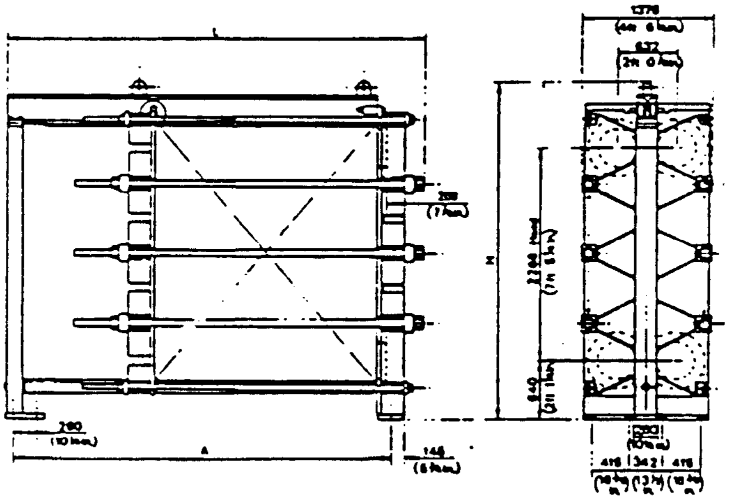


PRINCIPLE OF CONSTRUCTION

1. Assembly
2. Manifold
3. Core
4. Header
5. Nozzle
6. Width
7. Stacking height
8. Length
9. Passage outlet
10. Exterior sheet
11. Parting sheet
12. Heat transfer fin
13. Distribution fin
14. Side bar
5. End bar







DETAILS OF PLATE TYPE EXCHANGER

Frame Size	Approx. overall Length L mm	ft	in	Dim A mm	ft	in	Height H mm	ft	in	Plate capacity
1	2940	9	7¾	2525	8	3⅜	3605	11	10	100-185
2	3515	11	6⅜	3100	10	2	3605	11	10	180-265
3	4055	13	3⅝	3640	11	11¼	3605	11	10	260-340
4	4485	14	8⅝	4070	13	4¼	3605	11	10	335-400
5	6860	22	6	6445	21	2	3960	13	0	400-700

Plate materials
Stainless steel 316
Incoloy*
Commercially pure titanium
*Registered trade name
Gaskets are supplied in alternative synthetic rubbers to suit the processing requirements of a range of liquids

Heating surface per plate
2.2 m² (23.5 ft²)

Weights
Frame without plates or liquid connections

size 1	6218 kg	(6.1 tons)
size 4	6945 kg	(6.8 tons)
size 5	7620 kg	(7.5 tons)

Plate

stainless steel	13.6 kg	(30 lb)
titanium	8.2 kg	(18 lb)

Note
All dimensions given are for guidance only. Specifications may be amended at any time without prior notice.

1. APPLICABILITY

It is not expected that a hand calculation of furnaces be performed by the engineer. It is normally done by a manufacturer based on process data supplied by the engineer.

Furnaces are used to transfer heat directly to the process fluid and generally have a large duty and produce high process temperatures.

2. DESCRIPTION

2.1. A furnace consists of the following :

- A combustion chamber lined with refractory and burners
- Tubes which are located within the combustion chamber and where heat is transferred to the process fluid by radiation
- Tubes which are located externals to the combusion chamber in a convection zone which is also lined with refractory.
- Stack for disposal of flare gas.
- Air supply system by fan or induced draft.
- Instruments and controls.

2.2. TYPES OF FURNACE

2.2.1. Cabin furnace

- This is a rectangular furnace and contains tubes which can be horizontal or vertical. The burners are situated in the walls or floor, and the convection zone is located above the furnace.
- Flue gases discharge to a stack either directly or are driven by an induced draft fan.
- Burners are normally arranged in rows on two walls and are spaced so as to provide a radiation zone of constant temperature and avoid flame impingement on the tubes. An alternative arrangement is burners located in the floor of the furnace as shown in Figure 1.
- The connection bank contains rows of tubes across which the flue gas leaving the furnace is obliged to pass.
- A small negative pressure is maintained to prevent hot gas leakage.
- There is a pressure loss in the flue gas system and this has to be made up either by use of a fan discharging to a short stack or by natural bouyancy creating draft in a tall stack.

2.2.2. Cylindrical furnace (see Figure 1)

- These furnaces are vertical and contain radiation and convection zones or solely a radiation zone.
- The burners are located in the bottom and the radiation zone tubes can be vertical or helicoïdal. The convection bank is located above the radiation zone and contains rows of horizontal tubes.
- Generally the stack is vertically above the convection bank with no fan.

2.3. BURNERS

- Two types of burner are used in furnaces, induced air or natural draft burners and forced draft burners.

2.3.1. Induced air burners

These can burn gas or fuel oil simultaneously or independently. Excess air required is 15 % to 20 % for gas and 30 % to 40 % for liquids. If fuel oil is burned 0.3 kg/kg oil of steam is required for atomising. (Excess air indicates that above the stiochiometric ratio)

2.3.2. Pressure burners

The air for pressure burners is supplied by fan. It is therefore capable of control and the burner can operated with less excess air 5 to 15 %.

EXCESS AIR

- Determine the excess air recommended by the burner manufacturer and the type of burner air system proposed. See § 2.3.
- From this determine the kg of flue gas per kg of fuel fired remembering that air contains 21 % Vol of oxygen.

STACK GAS TEMPERATURE

This is controlled by 2 factors :

- The process fluid inlet temperature will determine the temperature of the gas leaving the convection bank.
- Condensation is to be avoided. If sulphur is present in the fuel the stack temperature is raised to avoid the possibility of production of corrosive sulphurous acid. This would result in a minimum exit temperature of about 120 °C.

5. EFFICIENCY

$$\eta = \frac{100 - \text{losses}}{100} \times \frac{Hf - Hc}{Hf}$$

If Hc = flue gas enthalpy at exit

Hf = enthalpy of combustion (net calorific value + sensible heat in fuel and air) + heat being by atomisation steam if required.

- Losses include radiation and unaccounted, e.g. unburned fuel (2 % is a good figure).
- For a furnace which is all radiant duty the efficiency is of the order of 50 to 55 %.
- A furnace with a convection bank will be from 75 to 85 % efficient.

6. PRESSURE LOSSES

Pressure is lost in :

. Burner air regulation	: 3 - 15 mm water	. Ducting	: variable
. Convection bank	: 5 - 15 mm water	. Stack	: variable

Pressure is gained by natural buoyancy of hot stack gas.

For a system using natural draft burners a low pressure loss is required across the burner and the furnace operates under negative pressure.

7. FLUE GASES VELOCITY

The flue gases should leave the stack at 10 - 20 m/s velocity to ensure safe dispersal.

8. CHOICE OF TYPE OF FURNACE

- Above a capacity of 60 x 10^6 Kcal/hr the cylindral furnace gives construction problems as the maximum diameter is about 10 - 11 m.
- A cabin furnace requires much more floor area than a cylindrical furnace the length can be as much as 27 m. If the tubes are horizontal then a withdrawal space for tube replacement will also be required. However for offshore applications the space requirement tends not to favour the cabin furnace.
- With a cabin furnace it is possible to obtain a uniform heat release across the radiation zone. The height can be about 15 m.
- With a cylindrical furnace it is not possible to obtain a uniform heat above release across the radiation zone. The height can be about 25 m.

9. ESTIMATION OF SIZE OF CYLINDRICAL FURNACE

The following is for a very preliminary sizing

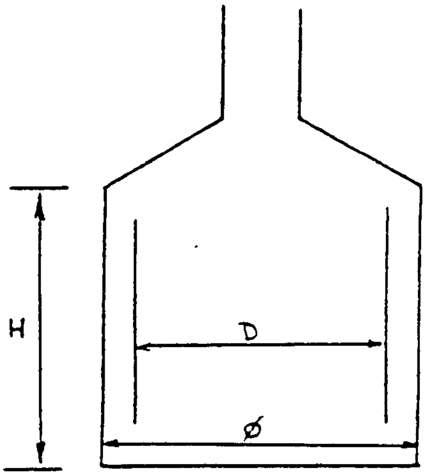
$D = \sqrt{2 \times Qa}$

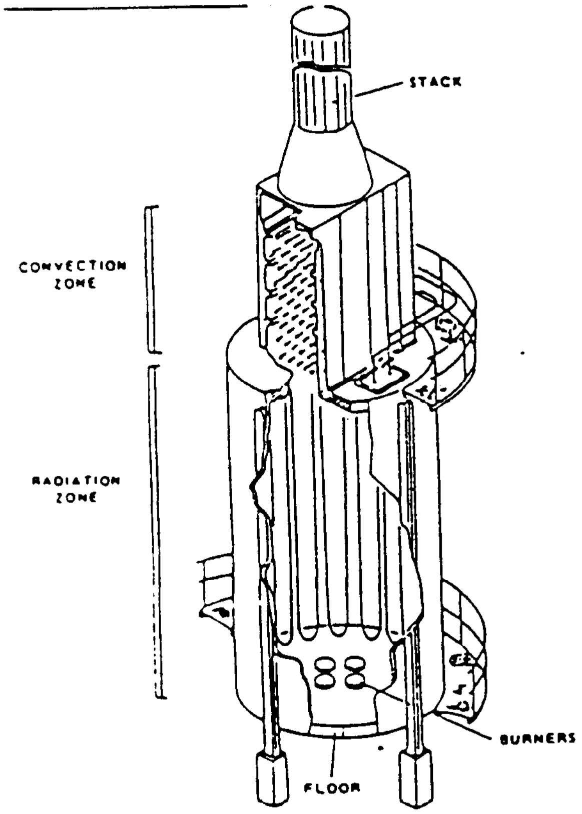
D in m

Qa = absorbed heat in 10^6 Kcal/hr

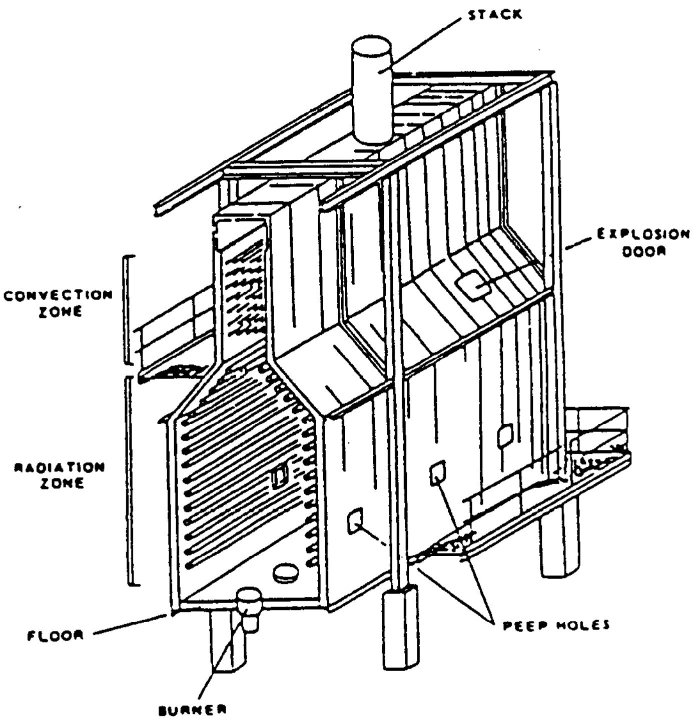
$\emptyset$ = D + 1 m

H = 2.5 D m util radiation bank





VERTICAL CYLINDRICAL FURNACE



HORIZONTAL TUBE CABIN FURNACE

5. PUMPS

1.0 APPLICABILITY

For both the feasibility study and a pre-project study the engineer will be required to evaluate a pump selection and fill in a data sheet with the basic information.

In order to provide the basis of a good cost and layout estimate it is important to understand the type and number of pumps for the service in consideration, and the associated power requirements.

2.0 DESCRIPTION AND GUIDELINE NOTES

TYPES OF PUMPS

- Generally there are three classes of pumps :

Centrifugal	Rotary	Reciprocating
1. Centrifugal	1. Cam	1. Piston
2. Propeller	2. Screw	2. Plunger
3. Mixed flow	3. Gear	3. Diaphram
4. Peripheral	4. Vane	
5. Turbine	5. Lobe	

- A pump selection chart is shown in Figure 1.

GENERAL USAGE

Centrifugal pumps (Process Pumps)

- Medium to high capacity for low to medium head requirements.
- Higher head requirements can be met by using multistage impellers.
- General service for all liquids, hydrocarbons, products, water, boiler feed.
- Simple, low cost, even flow, small floor space, quiet, easy maintainance.

Rotary pumps

- Many proprietery designs available for specific services.
- Essentially can handle clean fluids only with small suspended solids if any. Can pump liquids with dissolved gases or vapour phase.
- Can handle wide range of viscosities - upto 500 000 SSU at high pressures.
- Typical fluids pumped : mineral, vegetable, animal oils, grease, glucose, viscose, paints, molasses, alcohol, mayonaise, soap, vinegar and tomato ketchup !
- Generally specialist pumps for specific requirements.

Reciprocating pumps

- Pumps produce virtually any discharge head upto limit of driver power and strength of pistons and casings.
- Overall efficiency is higher than centrifugal pumps. Flexibility is limited.
- Piston pumps : can be single or double acting. Used for low pressure light duty or intermittent services. Less expensive than plunger design but cannot handle gritty fluids.
- Plunger pumps : high pressure, heavy duty or continuous service usage. Suitable for gritty or foreign material. Expensive.
- Diaphram pump : driven parts are sealed from fluid by plastic or rubber diaphram. No seals no leakage. Ideal for toxic or hazardous material. Can be pneumatically driven at slow speeds for delecate fluids.
- Triplex pumps : commonly used for TEG circulation.

REFERENCES AND USEFUL LITERATURE

4.1. LUDWIG VOL I CHAPTER 3

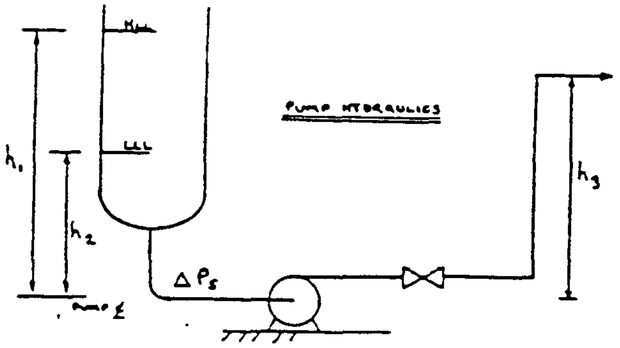
4.2. PERRY CHEM. ENG. HANDBOOK CHAPTER 6

4.3. CAMPBELL VOL II CHAPTER 14

4.4. "Centrifugal pumps and system Hydraulics"
Ugor J. Karassik Chem. Engrng Oct 4 1982

4.5. "New Program Speeds up Selection of a Pumping unit"
M. Seaman Oil and Gas J. Nov. 12 1979

4.6. "Rapid calculation of Centrifugal-pump hydraulics"
W. Blackwell Chem. Eng. Janv. 28 1980



1. FLUID CHARACTERISTICS

- Always quote at pumping temperature ie : normal suction T.

2. SUCTION PRESSURE

- Evaluate at pump suction flange

  Ps = Pop + Static head - line loss

  Pop = minimum vessel operating pressure bara.
  Static head : evaluate at LLL **always** take static head above pump centreline.
  h2 (approx. 0.6 m). head (bar) = m x specific gravity/10.197
  Line loss : evaluate ΔPline for bends, fittings, etc : for estimate use 0.1 bar/100 m.

3. NET POSITIVE SUCTION HEAD NPSHA

- NPSH, available is evaluated by the engineer. NPSHR required is stated by the vendor always try to provide 0.6 - 1 m NPSH more than vendor states.
- Vapour correction is calculated by substracting the vapour pressure of the fluid being pumped from the calculated suction pressure. Convert this to m head. For a fluid at bubble point the vapour pressure = Pop
  head (m) = bar x 10.197/SG.
  NPSHA = static head - line loss + vapour correction

4. DISCHARGE PRESSURE

- Delivery pressure - use maximum Pop of destination vessel
- Static head h3 - height of delivery point above pump or if a submerged discharge into a vessel the height of the HLL.
- ΔP discharge line - calculate based on line length, fittings etc or use minimum of 0.5 bar.
- ΔP exchangers, heaters, etc - use allowable ΔP from equipment data sheets. Estimate 0.7 - 1.0 bar if not available.

- Δ P orifices - for flow meters use 0.2 - 0.4 bar.
- Δ P control valves - use maximum valve of 0.7 bar, or 20 % of dynamic friction losses or 10 % of pump Δ P.
- TOTAL DISCHARGE PRESSURE - sum of all above Δ P values.

5. DIFFERENTIAL HEAD

- Discharge pressure - suction pressure convert to m head

6. FLOWRATE

- Normal flowrate is maximum long term operating flow
- Design flowrate is normal flowrate + design margin.
- Design margin

Use 10 % for feed pumps or transfer pumps
20 % for reflux pumps and boiler feed water pumps

7. POWER REQUIREMENTS

Note : although the term "horsepower" is still used power requirements are given in kW for metric calculations.

- Hydraulic horsepower - theoretical fluid HP = flow x head/36 KW
- Brake-horsepower (BHP) - hydraulic HP/ p pump efficiency KW
- Operating load - electrical input to electric driver at normal pump operating load = BHP/ nm motor efficiency Kw
- Connected load - electrical power to motor at rated motor size Kw
- Note pump speeds are either 1 450 rpm or 2 900 rpm

8. MAXIMUM DISCHARGE PRESSURE (shut off pressure)

- Shut off pressure = max suction pressure (calculate at HLL and ρ maxi)

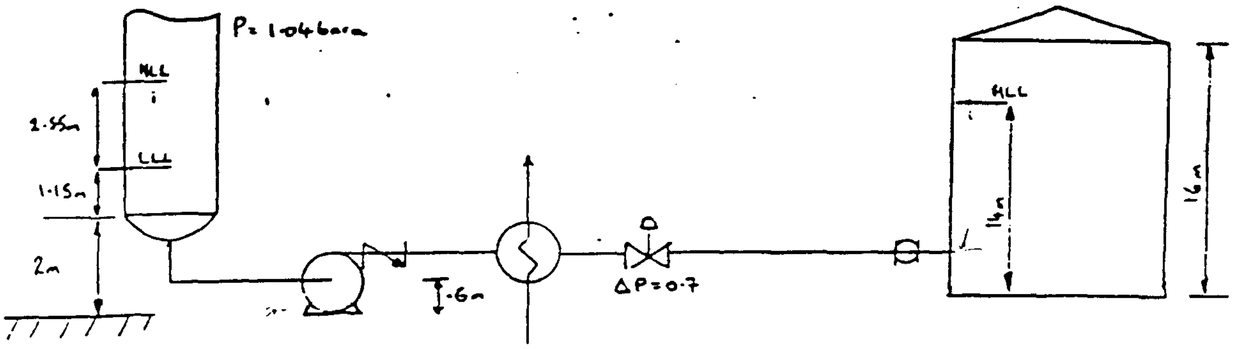
  + 120 % x normal pump Δ P

9. PUMP MINIMUM FLOW

- For an estimate use 30 % of normal flow.

10. PUMP WEIGHTS

- For an estimation purpose only Figure 4 can be used to determine the weights of a centrifugal pump package.



Indicate pressure, elevations and system sketch

FLUID PUMPED : Liquid : CRUDE

PUMP TYPE : CENTRIFUGAL SINGLE STAGE
Speed : 2900 rpm

Pumping temperature T :	34	°C
Vapor pressure at T :	1.04	bara
Density at P, T :	822	kg/m3
Specific gravity at P, T :	0.822	

Viscosity at P, T :	3.2	cP
Specific gravity 15 4 :	0.808	
Normal flow Q :	151	m3/h
Design margin :	25	%
Design flow (1) :	189	m3/h

SUCTION PRESSURE

Min. Origin Pressure =	bara	1.03 1.2
Static head @ LLL =	m	3.15
(m x sg x 0.0981)	bar	0.25
– ΔP suction line	bar	0.10
PUMP SUCTION PRESSURE		1.18

NET POSITIVE SUCTION HEAD

Static head @ LLL	m	3.15
- Line loss 0.24 cw	m	1.24
+ vapour pressure correction	m	–
TOTAL AVAILABLE NPSH	m	1.91

MAXIMUM SUCTION PRESSURE

Vessel PSV setting	bara	3.5
Static head at HLL [illegible]	bar	0.5
net bara		4.0

MAXIMUM DISCHARGE PRESSURE

Max. suction pressure	bara	4.0 51.5
Normal pump ΔP x 120 %	bar	
net bara		

Notes: 1) Sized for 20 000 BSOPD

DISCHARGE PRESSURE

Delivery pressure	bara	1.01
Static head (max)	bar	1.29
ΔP control valve(s)	bar	0.70
ΔP exchanger(s)	bar	0.70
ΔP orifice(s)	bar	–
ΔP	bar	–
ΔP line loss	bar	0.4
Other contingency	bar	0.5
TOT DISCHARGE PRESS	bara	4.6

DIFFERENTIAL PRESSURE

Suction pressure	bara	1.18
Discharge pressure	bara	4.6
Pump ΔP	bar	3.42 (2)
	m	43

POWER REQUIREMENTS

Brake Horse-power = $\frac{1 \times 2}{36\,\eta}$ (Fig 2 for η)	kW	322	3
Estimated motor size	kW	375	4
Design operating load $4/\eta m$ (Fig 3 for ηm)	kW	400	5
Estimated weight	kg	5100	5

PROCESS CALCULATION SHEET

TOTAL
TEP/DDP/DIP/EXP/SUR

PUMP

ITEM : PRODUCT PUMP
NO : P 2070 A/B

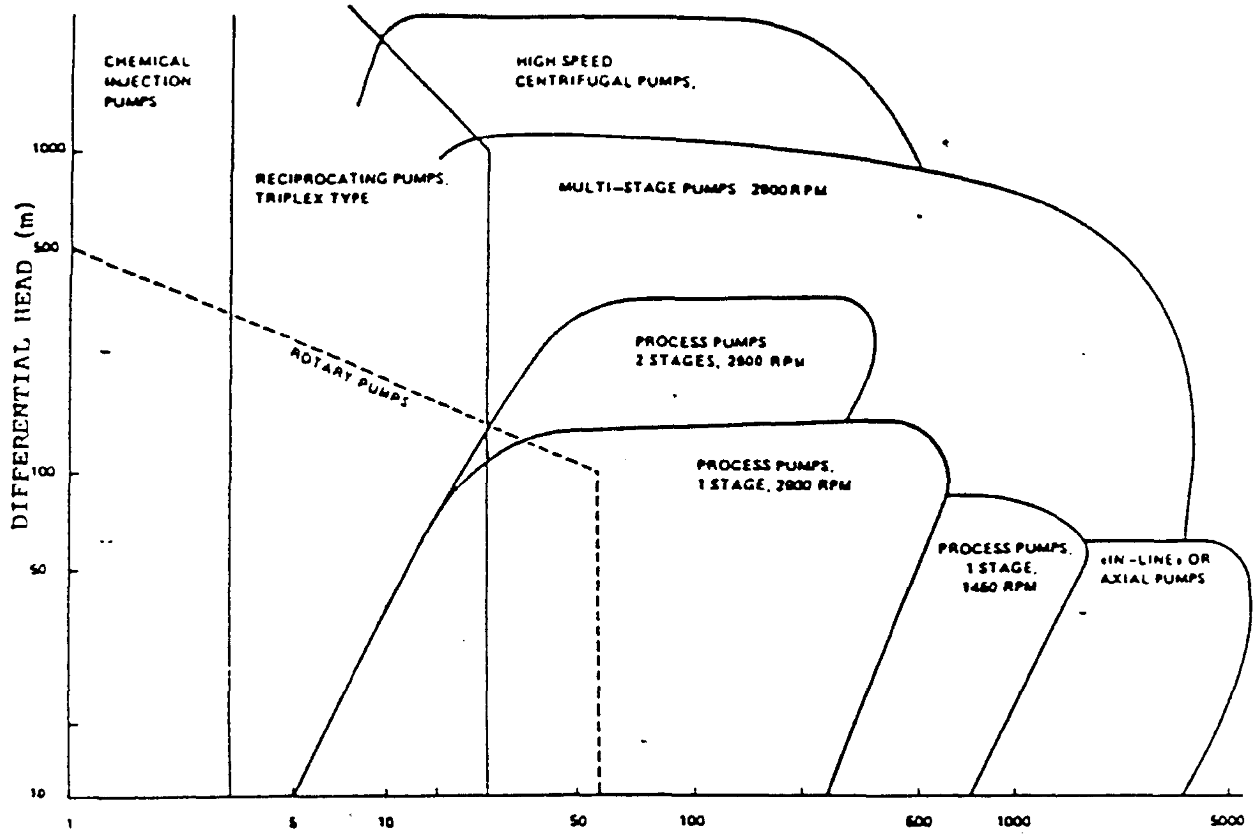


FIG. 1 GENERAL RANGES OF APPLICATION FOR DIFFERENT PUMP TYPES

FIG. 2 ESTIMATION OF CENTRIFUGAL PUMPS EFFICIENCY

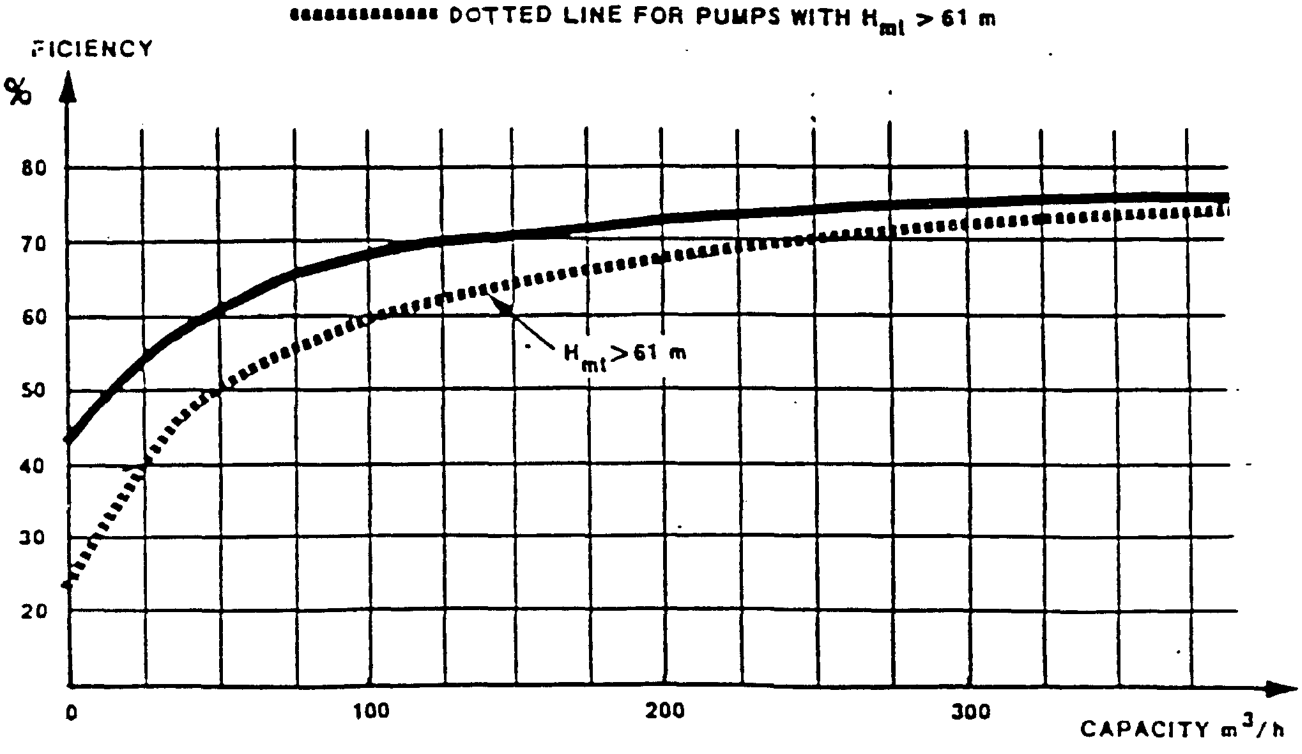


FIGURE 3

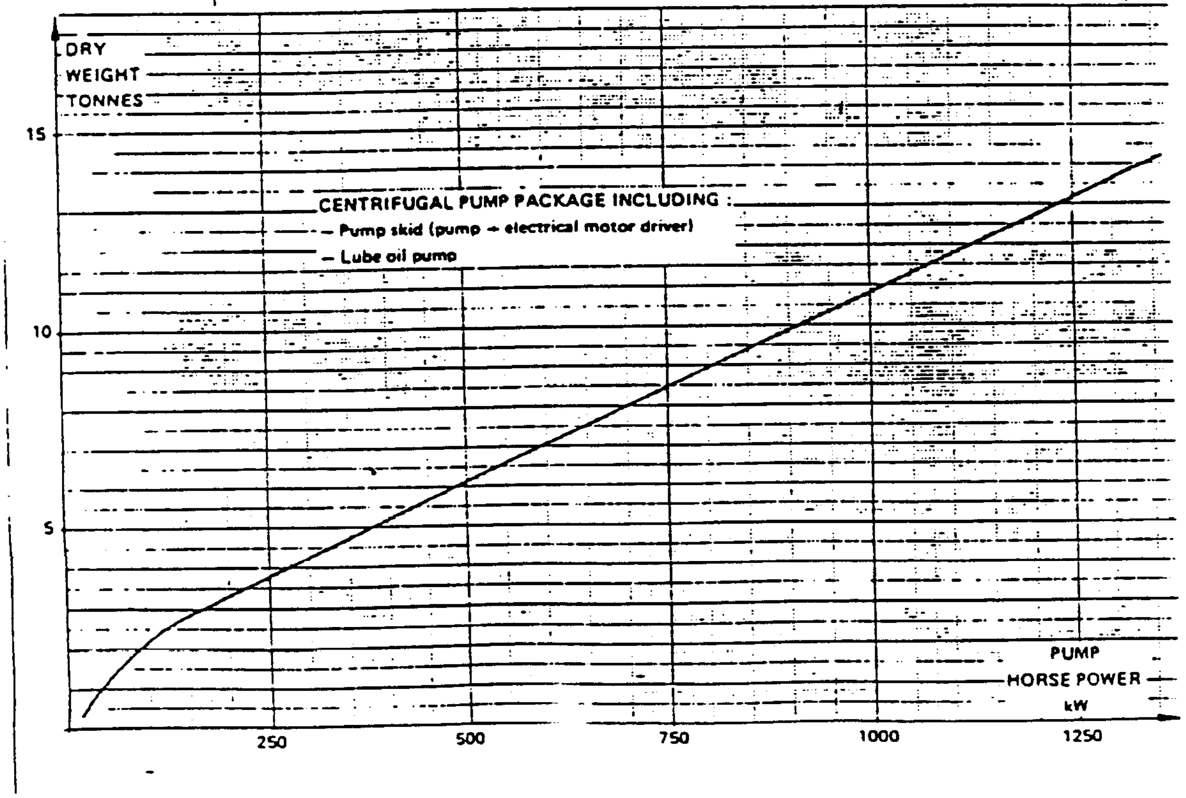
ELECTRIC MOTORS, RECOMMENDED SIZE & EFFICIENCY (1)

Kw = BHP x 0·7457

Pump Requirement At Design Conditions BHP	Probable Motor Rating HP	Motor Efficiency % of Full Load Capacity 50	75	100	Power Factor(2) % of Full Load Capacity 50	75	100
0 - 0.5	1	81	82	82.5	65	75	80
0.51 - 0.75	1-1/2	67	73	75	69	80	84
0.75 - 1.00	2	75	78	80	69	81	84
1.01 - 2.00	3	75	79	80	78	84	86
2.01 - 4.00	5	81	83	84	73	81	86
4.01 - 6.00	7-1/2	75	80	81.5	84	88	90
6.01 - 8.00	10	80	84	85	85	90	91
8.01 - 12.0	15	81	85	86.5	84	88	91
12.1 - 16.0	20	80	83	86	84	88	90
16.1 - 20.0	25	83	86.5	88	84	88	90
20.1 - 26.1	30	83	86.5	88.5	81	88	90
26.2 - 34.8	40	85	88	88.5	86	90.5	91
34.9 - 43.5	50	80	85	87.5	84	89.5	91.5
43.6 - 52.2	60	84	88	89.5	82	88.5	90
52.3 - 65.2	75	87	89.5	90.5	81	86.5	89
65.3 - 87.0	100	84	89	91	85	89	91
87.1 - 114	125	85	89.5	91.5	85	90	92
115 - 136	150	86	89	91	85	90	92
137 - 182	200	88	91	92.5	87.5	91	92
183 - 227	250	90	92.5	93.5	80	86	89
228 - 273	300	90.5	93	94	88	91	93
274 - 318	350	91	93	94	87	90	93
319 - 364	400	91	93	93	85	90	91
365 - 409	450	91	93	93	85	90	91
410 - 455	500	91.5	93	93.5	85	90	91
456 - 545	600	93	94	94.5	80	88	90

Notes: (1) Applies to totally enclosed motors only (i.e., explosion proof)
(2) To be used in determination of KVA's if desired.

FIGURE 4
DRY WEIGHT ESTIMATE FOR CENTRIFUGAL PUMP PACKAGE



7. COMPRESSORS

1. APPLICABILITY

For both feasibility and pre-project studies the engineer will be required to evaluate a compressor selection, discharge temperature, power and complete a data sheet.

To evaluate the discharge temperature and power it is more accurate to use SSI instead of the manual method presented here.

In order to estimate the basis of cost and layout it is important to understand the type of compressors for the service in consideration, and the associated power requirements.

2. DESCRIPTION AND GUIDELINE NOTES

2.1. TYPES OF COMPRESSORS

The principal types used in the oil and gas processing industries are :

- reciprocating (volumetric)
- rotary (volumetric)
- centrifugal
- axial

A compressor selection chart is shown in Figure 1..

2.2. GENERAL USE

2.2.1. Reciprocating compressors

Reciprocating compressors are widely used in the oil and gas industry for small to medium gas flows and high compression ratios. For example :

- Instrument and service air compressors
- Low capacity/high pressure gas compression for re-injection of field gas to maintain the gas lift capability.

2.2.2. Rotary compressors

The types of rotary compressors most frequently employed in the petroleum industry are as follows :

- Lobe compressors ("ROOTS" type)
- Screw compressors
- The reliability factor is generally higher than reciprocating machines.
- "Roots" type compressors are used where a high flow rate with a relatively low-pressure is required.
- Screw compressors are sometimes used in low flow gas service or for instrument and service air for installations of small to medium size.

2.2.3. Centrifugal compressors

- These Centrifugal compressors have become very popular offering more power per unit weight and essentially vibration-free. Initial costs normally are less than reciprocating compressors but efficiency is less and utility costs may be higher. Frequently used in the oil and gas process industry.

2.2.4. Axial compressors

These machines are particularly useful where a very high gas flow at moderate pressure increase is required. Such applications remain relatively rare in the industry, the exception being LNG plants.

2.3. DISCHARGE TEMPERATURE LIMITATION

- Discharge temperature is limited either for reasons of gas stability, gas condensation or compressor (or upstream equipment) mechanical resistance limit.
- For reciprocating compressor the maximum gas outlet temperature to be allowed is usually between 160 to 190 °C.
- For centrifugal compressor used in gas and oil extraction industries the discharge temperature is limited to 170/180 °C.
- Normally intercoolers are used to maintain temperatures within the above limits.

2.4. DESIGN MARGINS

If the flow is constant, no margin, but if the flow is coming from a production separator a margin of 10 % is recommended in order to take into account the possible slugs at the inlet of these production separators.

5. WEIGHT AND SIZE

For weight and size we recommend to ask the manufacturer as vendor catalogues detai only the size and weight of the compressor itself. As the compressor package also includes also the seal and lube oil console, control cabinet and sometimes the driver and gear box. The use of vendors catalogues could be misleading in estimating the installed weight.

Figure 4 could be used for a very preliminary estimation. It is established for the dry weight of a centrifugal compression package including :

- compressor skid (aeroderivative gas turbine + compressor)
- control room
- overhead tank (seal oil)

6. REFERENCES AND USEFUL LITERATURE

6.1.	LUDWIG	Volume 3	Chapter 12
6.2.	CAMPBELL	Volume 2	Chapter 14
6.3.	GPSA	Chapter 5	1979
6.4.	SSI Program		

Gas mixture components	(1) mol. frac	(2) MOL WT	(3) TC	(4) PC	(5) (1) x (2)	(6) (1) x (3)	(7) (1) x (4)	(8) Cp (note 1)	(9) (1) x (8)
		MW	K	bara				Kcal/kgmole°C	
					MW =	Tcmix =	Pc mix =		Mcp =

ote that T average = $\frac{T1 + T2}{2}$

$\gamma = \frac{Mcp}{Mcp - 1.99}$

Remark : If the composition is not known, use the curves in data Section giving Tc, Pc and γ versus molecular weight.

OPERATING CONDITIONS

SUCTION PRESSURE P1 = 6 BAR A

DISCHARGE PRESSURE P2 = 14 BAR A

PRESSURE RATIO P2/P1 = 2.33

SUCTION TEMP. T1 = 46 C
= 319 K

MW = 32.74

SUCTION FLOW W = 100000 KG/H

ACTUAL VOL FLOW V = 12400 M3/H

GAS DENSITY AT SUCTION = 7.73 KG/M3

STEP		NOTES
1. GAS PROPERTIES	PC = 44.11 BAR A TC = 291 K	
2. POLYTROPIC EFFICIENCY	η_p = 0.80	SEE FIG. 2
3. AVERAGE γ = MCP/MCP-1.99	γ = 1.15	ESTIMATE T2 (95°c)
4. DISCHARGE TEMP $= T1 * \left(\frac{P2}{P1}\right) \uparrow \left(\frac{\gamma - 1}{\gamma \eta_p}\right)$	T2 = 366 K = 93 °c T2 = 94 °C OK ✓	REPEAT STEP 3-4 IF T2 IS DIFFERENT FROM ONE USED IN STEP 3
5. DETERMINE Z AVG SUCT DISCH AVG	Z1 = 0.9651 Z2 = 0.9508 Z = 0.958	
6. CALCULATE GAS HORSEPOWER $GHP = \frac{Z * R * W * \gamma * (T2 - T1)}{MW * 3600 * (\gamma - 1)}$	GHP = 1694 KW	R = 8.314 KJ/KGMOLE
7. CALC SHAFT HORSEPOWER PS = GHP * (1 - F/100) * 1/ηm	PS = 1655 KW	F ηM GHP<800 KW 5.0 .96 800<>10 MW 7.5 .97 >10 MW 10 .98
8. ESTIMATE DRIVER POWER ELECTRIC MOTOR PS * K GAS TURBINE PS * (1.14 + K)	PO = 1903 KW PO = KW	K = 1.15 K = 0.02 TO 0.04 WITH GEARBOX
9. ESTIMATED PACKAGE WEIGHT COMPRESSOR-DRIVER-LUBE	M = 40×10^3 KG	(SEE FIG 4)

NOTES :

TOTAL TEP/DOP/DIP/EXP/SUR	PROCESS CALCULATION SHEET CENTRIFUGAL OR AXIAL COMPRESSOR	ITEM : EXAMPLE NO :
BY CHK	DATE JOB TITLE : PROCESS MANUAL	JOB N° : REV

7·5

OPERATING CONDITIONS

SUCTION PRESSURE P1 = 6

DISCHARGE PRESSURE P2 = 14

PRESSURE RATIO = $\frac{P2}{P1}$ = 2·33

SUCTION TEMP T1 = 46 °C

= 319 K

MW = 32·74

SUCTION FLOW W = 100000 kg/hr

ACTUAL VOLUMETRIC FLOW V = 12400 m³/h

DENSITY AT SUCTION CONDITIONS = 7·73 kg/m³

STEP			NOTES
1. GAS PROPERTIES	Tc = 291 Pc = 44·1	K bar a	
2. AVERAGE γ = MCp/MCp - 1.99	γ = 1·15		T2 = 90°c
3. CALCULATE DISCHARGE TEMP $= T1 \times \left(\frac{P2}{P1}\right)^{\frac{\gamma-1}{\gamma}}$	T2 = 362 = 89	°K °c	Repeat 2 - 3 if T2 differs from that used in STEP 2
4. DETERMINE OVERALL EFFICIENCY ηg	ηg = 0·8		See Fig 3
5. CALCULATE GAS HORSEPOWER $GHP = \frac{R \times W \times \gamma}{MW \times 3600 \times \gamma - 1} \times (T2 - T1)$	GHP = 2325	kW	R = 8.314 kJ/kgmole
6. CALCULATE SHAFT HORSEPOWER $PS = GHP/f \times \eta g$	PS = 3030	kW	f = 0.96 to 0.97
7. CALCULATE DRIVER POWER Electrical Motor Po = 1.15 x PS	Po = 3480	kW	
8. ESTIMATED WEIGHT	70000	kg	

NOTES :

TOTAL TEP/DDP/DIP/EXP/SUR	PROCESS CALCULATION SHEET
RECIPROCATING COMPRESSOR	ITEM : NO : EXAMPLE.

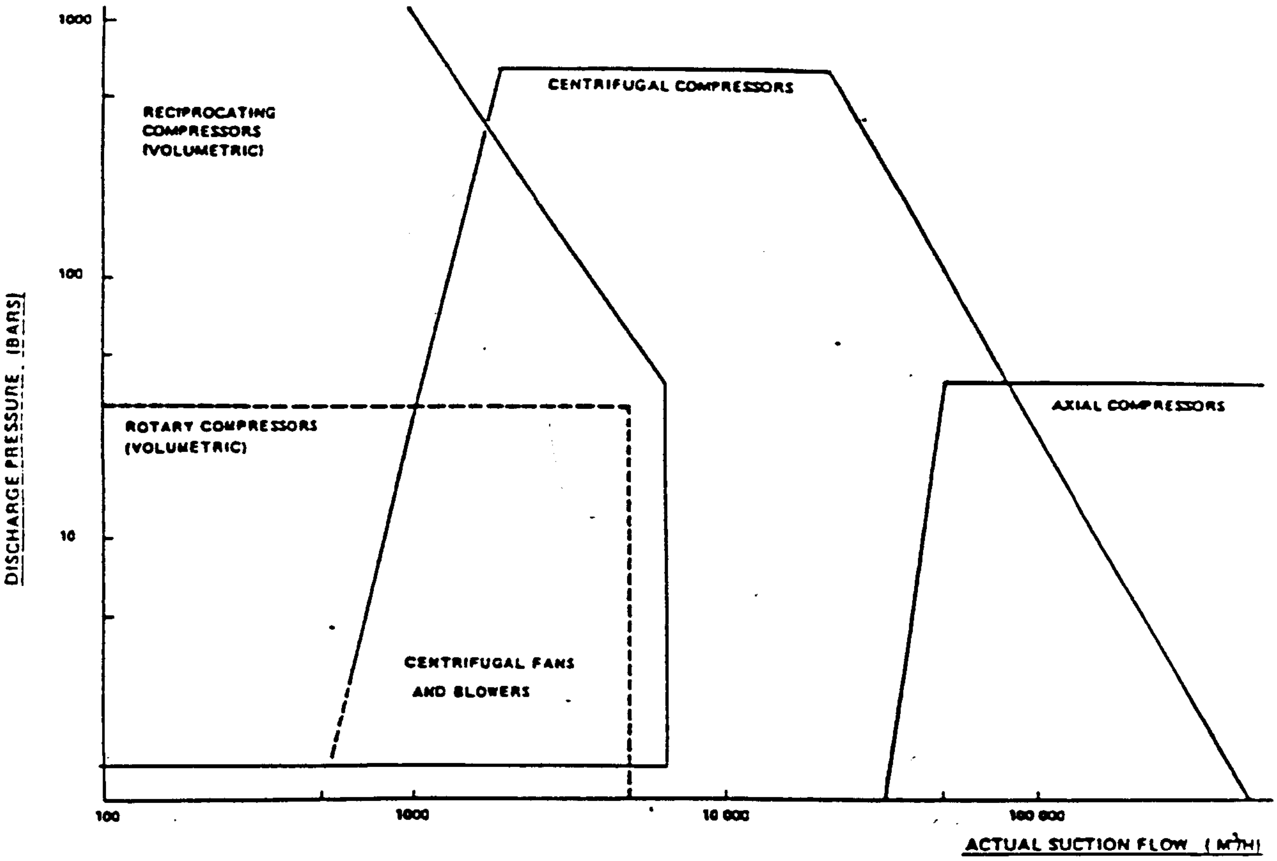


FIG. 1 GENERAL RANGES OF APPLICATION FOR DIFFERENT COMPRESSOR TYPES

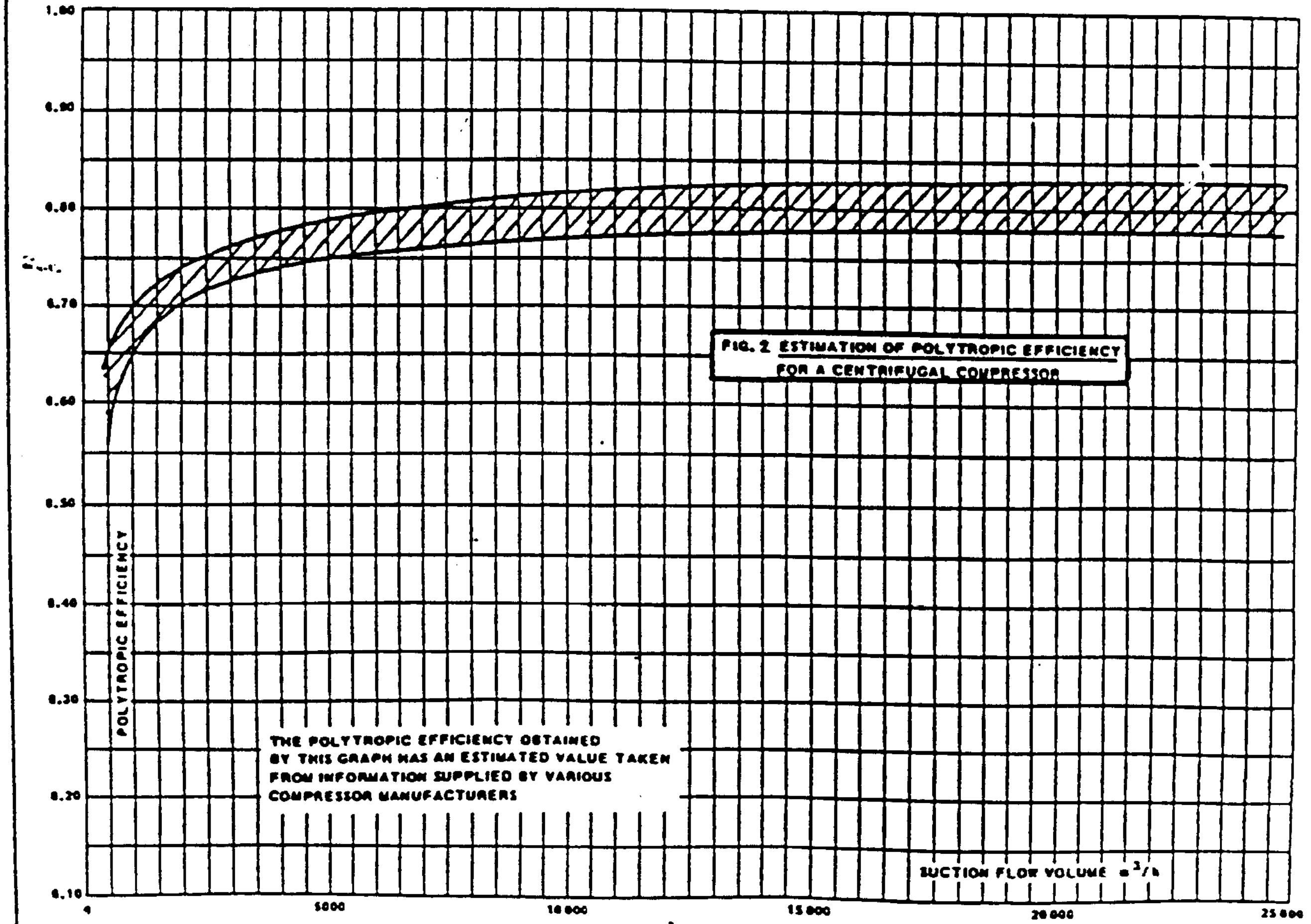


FIG. 2 ESTIMATION OF POLYTROPIC EFFICIENCY FOR A CENTRIFUGAL COMPRESSOR

THE POLYTROPIC EFFICIENCY OBTAINED BY THIS GRAPH HAS AN ESTIMATED VALUE TAKEN FROM INFORMATION SUPPLIED BY VARIOUS COMPRESSOR MANUFACTURERS

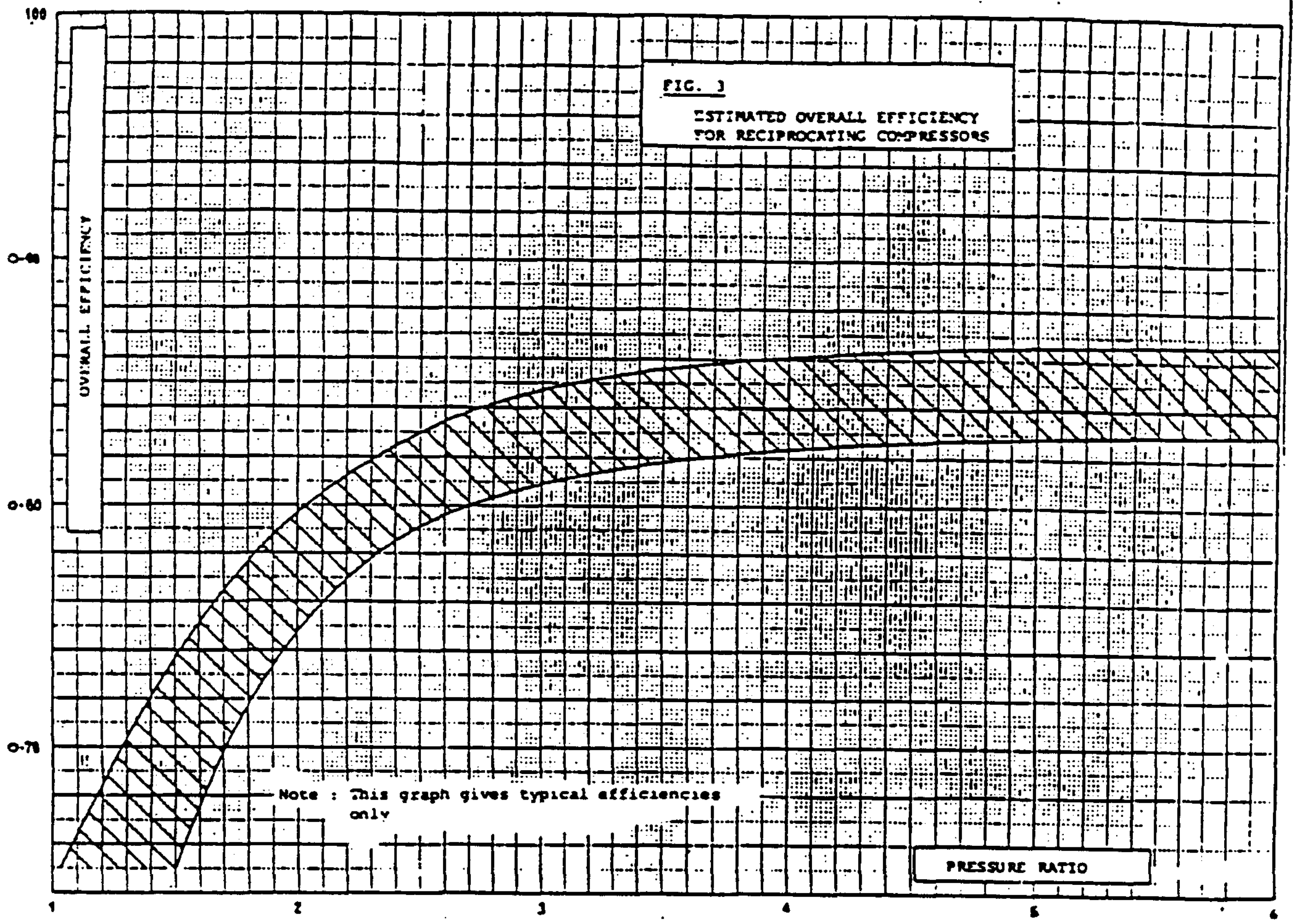


FIG. 3

ESTIMATED OVERALL EFFICIENCY FOR RECIPROCATING COMPRESSORS

Note : This graph gives typical efficiencies only

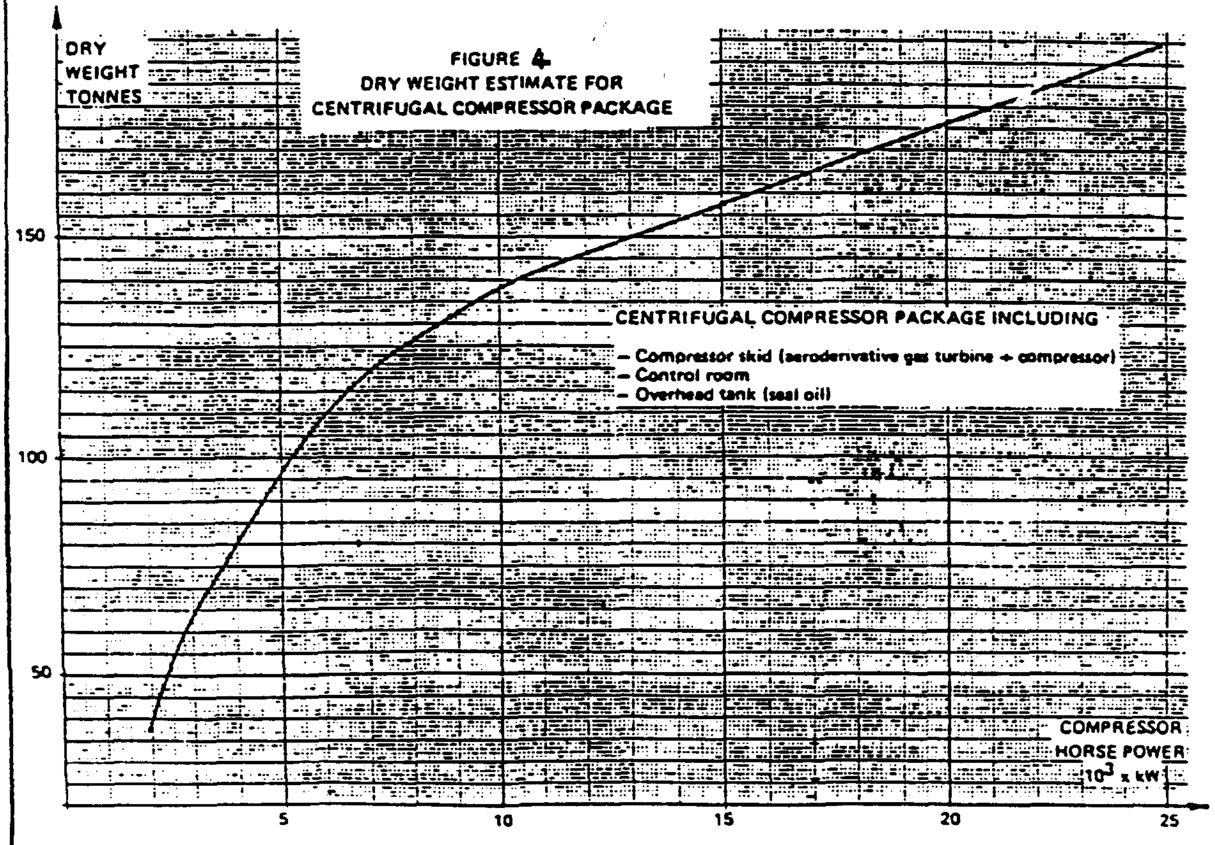


FIGURE 4
DRY WEIGHT ESTIMATE FOR CENTRIFUGAL COMPRESSOR PACKAGE

CENTRIFUGAL COMPRESSOR PACKAGE INCLUDING
- Compressor skid (aeroderivative gas turbine + compressor)
- Control room
- Overhead tank (seal oil)

8. EXPANDERS

1. APPLICABILITY

For both the feasibility study and a pre-project study the engineer will be required to fill in a process data sheet with the basic information and to estimate the expander horsepower.

Outlet conditions and horsepower estimation can be calculated accurately by computer. Hand caks for pure component systems using a MOLLIER diagram are OK.

2. DESCRIPTION AND GUIDELINE NOTES

- The turbo-expander is a mechanical device which is designed according to the laws of thermodynamics and aerodynamics. It removes energy from a process gas which results in a drop in pressure and temperature of the gas. The energy removed is converted into mechanical energy which is most often used to drive a single stage compressor.
  - Turbo-expanders could be used for :
    - cryogenic pressure let down
    - dew point control
    - C3/C4 recovery
    - C2 recovery
    - ethylene processing, etc...
- Thermodynamical principal. See Figure 1.
- Expanders efficiency

  The expander efficiency is the ratio of the actual energy removed to the maximum theoretical energy on Figure 1 :

  $$\eta = \frac{HB\ HA}{HBI\ HA}$$

  Expander efficiency depends on :

  - mass flow rate
  - inlet pressure
  - inlet temperature
  - discharge pressure
  - gas composition
  - speed

  Generally a value of 80-85 % can be used for estimation purposes. See Figure 2.
- Liquid content at the outlet of the expander varies from 10 to 30 % (weight)
- Inlet gas must be free of solid particles and <u>water</u> (ice formation is prohibited).
- Maximum horsepower of the manufactured turbo expanders is about 12 000 HP. This figure should not however be considered as a limit.
- Turbo expanders can be used in series.
- Efficiency is affected by the variation of the design flow rate See Figure 3 for an estimation.

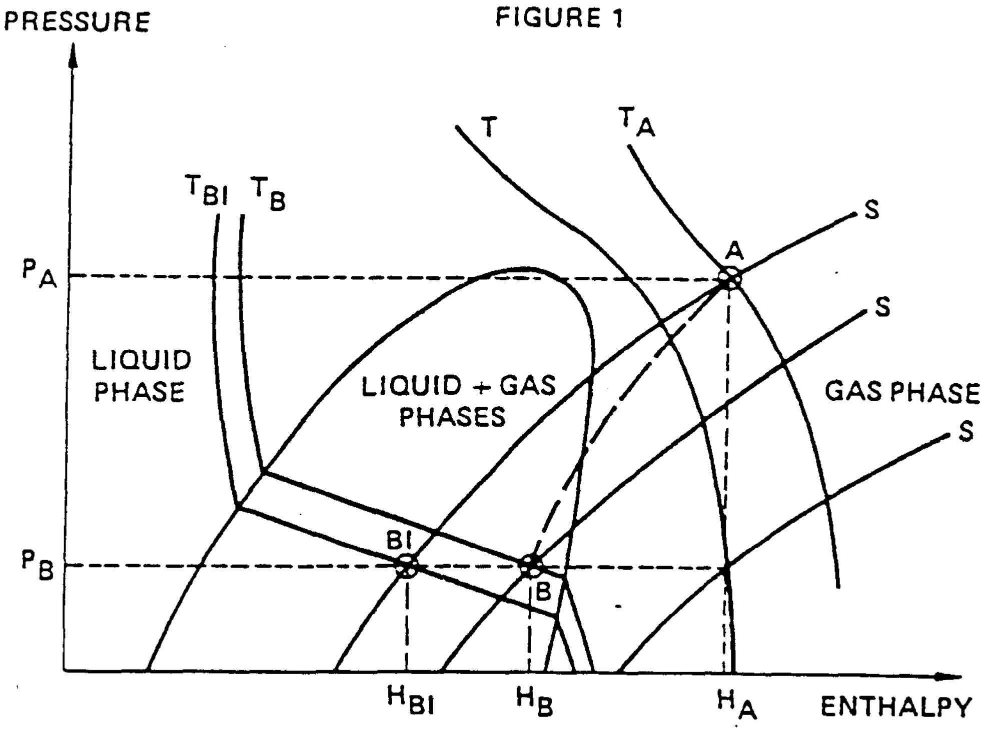
3. REFERENCES AND USEFUL LITERATURE

CAMPBELL VOLUME II

Engineer's guide to turbo expanders	HYDROCARBON PROCESSING APRIL 1970 Page 97...
Turbo expander applications in natural gas processing	JOURNAL OF PETROLEUM TECHNOLOGY May 1976 Page 611 etc...
What you need to know about gas expanders	HYDROCARBON PROCESSING February 1970 page 105...
Turbo expanders offer processors a way to conserve energy	THE OIL AND GAS JOURNAL Jan. 23, 1978 page 63...
Use expander cycles for LPG recovery	HYDROCARBON PROCESSING Dec. 1974 Page 89...

VENDOR DOCUMENTATION
i.e. : ROTOFLOW, MAFI-TRENCH..;

FIGURE 1



P_A	Inlet pressure	T_A	Inlet temperature	H_A	Inlet enthalpy
P_B	Outlet pressure	T_B	Outlet temperature	H_B	Outlet enthalpy
T_{BI}	Outlet theoretical temperature	H_{BI}	Outlet theoretical enthalpy		

FIGURE 2

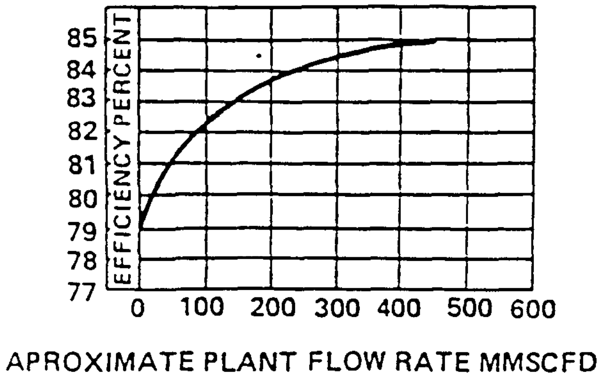
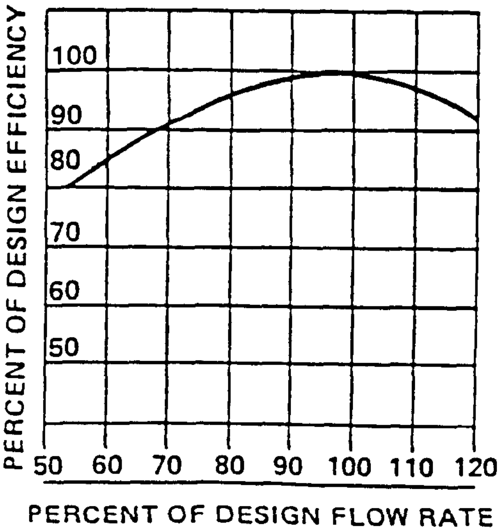


FIGURE 3

ESTIMATED PERFORMANCE AS A FUNCTION OF DESIGN FLOW RATE



9. FLARE SYSTEMS

1. APPLICABILITY

For the feasibility and preproject studies, a detailed design of the flare system is not needed. Required information for either study will include ;

- Evaluation of number and levels of flare system
- Determination of maximum relieving (and hence flare design capacity)
- Flare KO drum Design
- Estimation of height of flare stack or boom length and type of tip required
- PSV sizing (not always required, depends on project).

For further more detailed specification and design requirements consult the CFP DESIGN GUIDE ON FLARES-VENTS-RELIEF AND BLOWDOWN SYSTEMS.

2. DEFINITIONS (see section 3 in DESIGN GUIDE)

- Relief system : includes any pressure relief valve/rupture disc downstream piping and liquid separator
- Blowdown system : includes any depressuring valve, downstream piping and separator (normally the pressure relief and depressuring systems utilize common piping and separator)
- Flare system : a system which ensures the combustion of hydrocarbons
- Vent system : the release of hydrocarbons to the atmosphere without combustion
- Design pressure : the pressure used to design the vessel and calculate the wall thickness (see section 1.0.)
- Set pressure : the pressure at which a safety device is adjusted to open under service conditions. Usually equal to the Design Pressure
- Accumulation : maximum allowable increase in vessel pressure during discharge through the safety device. Normal accumulation is 10 % but 20 % is allowed for external fire due to hydrocarbon liquids. For HC gas fires an accumulation of 5 % is recommended.

3. FLARE SYSTEM ANALYSIS AND GUIDELINES

This section details how to determine the number and levels of the required flare system for a feasibility or preproject study and other guidelines.

- A system of items of equipment and piping can be protected against overpressure most economically by considering it as a single unit when calculating the relieving capacity

- Block valves should not be present in the system so as to isolate a unit from its relieving point. Special cases may warrant a car-sealed open or locked valve. However such arrangements should be avoided if possible

- Interconnecting piping should be of adequate size and not subject to plugging. The system should not be of such a size that two separate systems would be more economical

- In specifying the design pressure of the individual items and safety valve setting there are two approaches

  - Set the design pressure of each item independently. Then specify safety valve settings to protect the weakest link in the group of items

  - Study the items as a single system initially. This is preferable as it avoids having an unexpected "weak link" limit the operating conditions.

- Consideration should be given to possible abnormal conditions viz :

  - Light hydrocarbon systems can reach low temperatures during depressurization

  - Heat exchange trains may be bypassed resulting in higher than normal downstream temperatures

  - Failure of cooling medium can cause excessive downstream temperatures

  - Production separators may have a varying feed temperature, especially offshore.

- It is often required or beneficial to provide two or more separate piping systems from the items of equipment to the flare system eg : high and low temperature headers. Consideration should be given to the following

  - Relief gases below 0°C must be kept apart from warm moist gases to prevent formation of ice within the flarelines. This could cause a system plug up

  - Segregated systems may be economically desirable to minimize the extent of low temperature piping

  - By segregating the flows from high and low pressure sources into two separate flare systems greater use of the high pressure drops can be achieved without imposing severe backpressures on the low pressure systems

  - The molecular composition of some streams may warrant their segregation from other streams. e_g moist CO_2 or H_2S is corrosive. It may be cheaper to fabricate a second smaller vent system to handle these rather than fabricate the entire system in corrosion resistant material.

Determination of the flare system and level can be summarized in the following step by step analysis.

1. Does the facility contain process areas with distinct pressure levels eg : HP compression, LP compression ; atmospheric separation ?
   If so, consider two or more flare levels if sufficient limitation is imposed by the LP section
2. Does gas exist at high pressure that on depressuring will fall to below 0 °C. If so, it must be segregated from warm relief gas. If the temperature falls below - 29°C may have to consider low temperature steel headers
3. Identify any corrosive relief sources and consider if need to pipe up separately
4. Is a vent system required for tank breathers, regeneration vents etc...
5. Identify on the PFD(s) the set pressures of each PSV anticipated and consequently its maximum allowable backpressure (MABP usually 10 % of set pressure). Locate the "weak links" in the process i.e. : the low design pressure vessels. If only 1 or 2 exist within the system consider installing balanced relief valves (MABP = 40 % set) so as to incorporate them into a higher pressure flare system, or even alter the design pressure of the weak links to acheive the same. This may be more economical than specifying two flare levels.

Having determined the configuration of the flare system, it is necessary to size the main headers only and the flareline itself. For this, an idea of the maximum relief load generated will be required. For the studies a full "risk analysis" of upset conditions is not necessary neither is a listing of every relief load and conditions.

The sizing case of the flare system can usually be judged by inspection. Invariably, the largest vent flow will be a full flow relief off the first separator or compression drum or a total electrical failure. This may be supplemented by a simultaneous depressurization of a compressor or equipment loops resulting in a flare design flow higher than the normal plant throughput. Generally fire generated loads do not dictate the sizing of the flare system, but may influence the sizing of laterals and subheaders. A certain degree of experience will help in identifying the possible one or two cases that will size the flare system without having to perform a full plant risk analysis.

In some cases, the resulting flaring loads may be minimised by using ESD isolation valves or automatic controls to start back-up equipment.

4. HEADER SIZING : STACK AND TIP CHOICE

In order to estimate the main flareline and header sizes based on backpressures, 3 pieces of information are required :

- Design flowrate temperature MW
- Length of flareboom or height of stack
- Type of tip and stack to be used.

4.1. DESIGN FLOWRATE TEMPERATURE AND MW

This has already been determined from the previous section.

4.2. TYPE OF TIP + STACK TO BE USED (see section 10 in Flare Design Manual)

The choice of stack and tip type will obviously be dictated by the location of the plant under design.
For onshore plants in remote areas it is usually sufficient to use a remote vertical stack with a conventional pipeflare tip. The height of the stack will be determined by the radiation limitation on the designated sterile area round the stack. For non occupied areas, this figure could be high as 5000 BTU/h.ft^2 (15 700 W/m^2) resulting in a short stack height. For cases where high flaring loads still result in a tall stack, a further reduction in height can be achieved by using a Coanda/Indair or similar type high pressure sonic flare tip (see section 10 in FLARE DESIGN MANUAL for discussion of each tip type).
Offshore the choice is somewhat more complex in choosing between a remote vertical flare or similar, or an integral 45° mounted boom flare or even on board vertical stack. The decision between these is more often than not governed by economics, structural considerations and specifics pertainent to each platform location eg : water depth. Generally, however sonic flare tips are used where pressure levels allow (2-5 bars) at the tip entry in order to reduce stack/boom lengths, by reducing radiation levels, and associated support structure weights.

4.3. FLAREBOOM - FLARE STACK SIZING

The flareboom or stack (hereafter termed flare) length is determined by the maximum allowable thermal radiation tolerable on the platform or surrounding area. A detailed calculation of this value for vertical or inclined flares on or offshore under a variety of wind conditions and temperatures can be performed using the computer program SUPERFLARE. For feasibility and preprojects, however an estimate of radiation level can be determined using the method as detailed in AP 521. See Appendix 1.

Recommended Radiation levels are given below :

Condition	Allowable radiation Btu/h.ft^2	Exposure period
Areas where personnel may be located and expected to perform their duties continuously	1000	Infinite
Areas where personnel may be located from which escape is possible and shelter is attainable	2000	1 minute
Areas where equipment is located and personnel are not normally present during operation, but if present immediate shelter is available	3000 (Emergency flaring only)	5 seconds
Areas where personnel are not permitted during operation	5000	0
Helideck	1000	

The above figures are maximum allowable radiation intensities inclusive of solar radiation (250 BTU/hr ft^2).

It should be noted that the following recommended values of F - Fraction of heat radiated and mach numbers at the tip.

a) Pipe flare

Low MW gas F = 0.2
Ethane F = 0.25 Velocities - max at design relief = 0.5 M
Propane F = 0.3 - normal continuous = 0.2 M

b) Indair/Coanda

All gases F = 0.1 Mach 1

c) Mardair F = 0.05 . Mach 1

Having calculated the flare length based on radiation analysis and established both the design flare rates and tip type the main header can now be sized.

4.4. HEADER SIZING

The major criteria governing the sizing of the header are backpressure and gas velocity. Flare headers must be both large enough to prevent excessive backpressure on the plant safety valves and to limit gas velocity and noise to acceptable levels.

Sizing procedure

1) Identify "weak link" with respect to MABP on safety values. (this should have been done when determining the levels of relief). This is the maximum upstream pressure tolerable in the system.

2) Calculate the Δ P across the flare tip for the relief design flow. For sonic type tips the backpressure will be 2.0 to 5.0 barg depending on load.

For pipeflare tips use :	Flare tip	0.5 - 2.0	psi	(0.034 - 0.14 bar)
	Fluid seal	0.2 - 0.5	psi	(0.014 - 0.034 bar)
	Molecular seal	0.5 - 1.0	psi	(0.034 - 0.07 bar)

3) Estimate the equivalent length of piping from the tip to the flare KO drum. (Allow generous margins, flare headers are complex and rarely straight).

4) Calculate the sonic velocity of the relief gas

$$V_{sonic} = 91.19\sqrt{\frac{KT}{MW}} \text{ m/s}$$

K = CP/CV
T = K

This will give a first estimate of required pipe id based on maximum relief flow. The stack diameter should be one or two sizes less than the tip diameter. LIMIT VELOCITY IN STACK TO 0.85 M; AT DESIGN FLOW.

5) Using the estimated D calculate the Δ P from tip to flare KO drum. The Conison equation is recommended for isothermal flow :

$$P_1^2 = P_2^2 + \frac{u_2^2 \; P_2}{V_2}\left(39.4\,\frac{fL}{d} + 2\ln\frac{u_2}{u_1}\right) \times 10^{-5}$$

Where :
1 = upstream conditions
2 = downstream
p = pressure bar (a)
u = velocity m/s
v = specific vol m^3/kg
f = moody friction factor
L = equivalent length m
d = pipe id inchs

This calculation requires a degree of trial and error as $u1 = f(P1)$

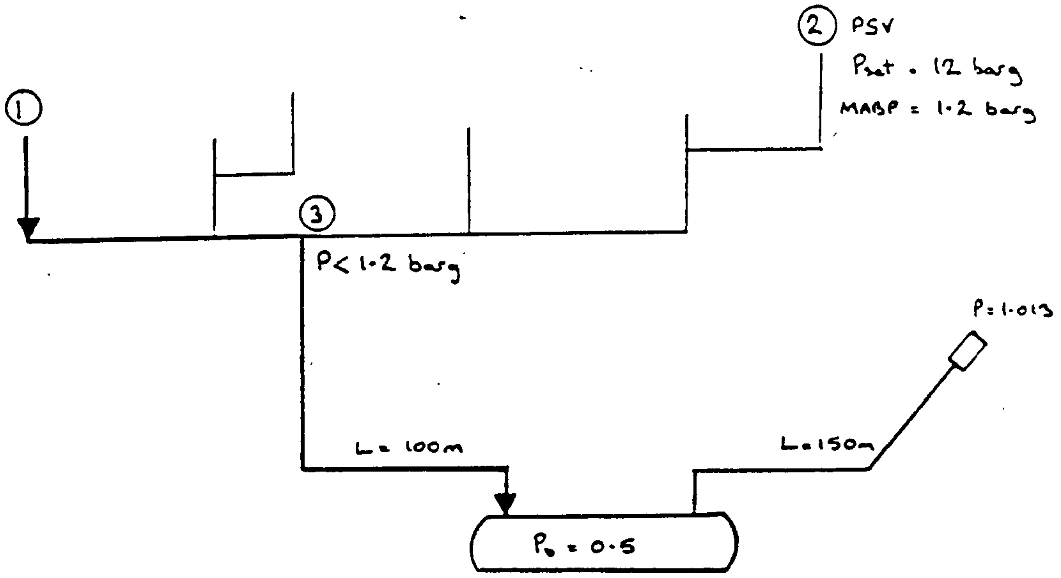
6) Examine the P1 (calc) at the relief drum and decide if the stack + header diameter is adequate ie is P1 (calc) drum approaching the maximum upstream pressure allowable at the plant ? if so increase the diameter and repeat the ΔP calc.

7) Once satisfied with the drum-tip line proceed back up the flare header and calculate the next section of line diameter.

8) Continue along the headers, adjusting flowrates as necessary if sources disappear, until the "weak link" criteria has been satisfied.

9) If the project requires sub headers and laterals can be estimated from the main line static backpressures calculated above.

EXAMPLE :



1. Flare design is based on vent flow from source (1)
2. Weak link in system is set by PSV at source (2)
3. System must be designed for a design flow from source (1) not giving a backpressure at point (3) of more than 1.2 barg.
4. Size line from tip to drum (L = 150 m) to give P_{drum} 0,5 barg (say) size line from drum to point (3) (L = 100 m) to give $P_1 < 1.2$ barg.
5. Check that source (1) can flow from (1) to (3) with pressure drop available.

NOTE : 1) Laterals ---> sub headers ---> headers must increase in diameter as the system progresses to the tip.

2) Max velocity in a line is MACH 0.7 for short duration reliefs only.

3) When calculating ΔP for flare systems isothermal flow is assumed for each section. For high source pressures with low MW a ΔT vs ΔP profile will yield more accurate results, i.e. adjust temp at specific points in the system to account for ΔP occured.

5. FLARE KO DRUM SIZING

A flare KO drum is provided to drop out and collect the liquid part of the flare vapours in order to :

- prevent liquid accumulation at the base of the flare boom or tower
- to minimize the risk of burning liquid (golden rain) emerging from the tip and falling on personnel
- to recover and reclaim valuable product materials.

5.1. DESIGN CONSIDERATIONS

- separate knock out drums are generally required for each level of flare system installed i.e. : an HP KO drum, LP KO drum, LLP drum
- cold vapour lines (i.e. < 0°C) can be introduced immediately upstream of inlet line to a "warm" drum providing the resultant temperature in the drum does not call below design. This precludes the need for two independant drums.
- FLARE KO DRUMS SHOULD BE HORIZONTAL AT ALL TIMES.
- Mist eliminators are not to be installed. Min design pressure of drum is 3.5 bar (g)
- Heating coils should be installed in flare KO drums to prevent freezing of residual liquids. Typical is to maintain a T min = 4°C
- LIQUID DROPLET SIZE (per API 521)

Recommended particle sizes are :

VERTICAL FLARE		150	(offshore)
INCLINED BOOM	> 45°	150	"
	< 45°	400	"
REMOTE FLARES		600	

5.2. DRUM SIZING

Based on the above design considerations the flare KO drum can be sized using the method outlined in section 2.0. VESSEL DESIGN.

For a flare KO drum, the normal liquid level should be kept in the lower part of the drum i.e. : utilise as much space as possible for the vapor-liquid de-entrainment. If a large diameter drum results consider using a split flow arrangement with the exist nozzles mounted on the head. This will maximise the L/D ratio and give a smaller highter drum. This is especially useful offshore where weight + space are a major concern.

An LCHH will normally be installed in the flare drum to initiate a plant shutdown wellhead shut in offshore).

6.0. RELIEF DEVICE SIZING (For more detail see API 520, 521)

6.1. GENERAL

- Safety valves are either termed balanced or conventional depending upon the backpressure limitation
- Rupture discs are less robust than an equivalent safety valve and cannot be relied on to function accurately. It is recommended that rupture discs are avoided

6.2. BACKPRESSURE

- Backpressure exists in two forms :
  - . flowing backpressure is the pressure on the discharge side of a PSV that is blowing off to the relief system
  - . superimposed backpressure, or static backpressure is the pressure on the discharge side of a PSV caused by another relief source in the system venting to flare
- For conventional valves the Maximum Allowable Backpressure (MABP) for either superimposed or flowing is 10 %. For balanced relief valves up to 40 % can be allowed for without a reduction in the valve capacity.

6.3. LIQUID RELIEF

The formula for sizing liquid-relief valves is :

$$A = \frac{\text{gpm}}{27.2\ K_p \cdot K_w \cdot K_v} \sqrt{\frac{G}{P_d}} \quad \text{ins}^2$$

Where :

A = Effective discharge area, ins^2
gpm = Flowrate, u.s. gallons/min
G = Specific gravity at flowing temperature
K_p = Capacity correction factor (from figure 6.5)
P_d = Relieving pressure minus constant back pressure
K_w = Capacity correction factor (from figure 6.4)
K_v = Viscosity correction factor (from figure 6.3.)

6.4. VAPOR RELIEF

The formula for sizing vapor relief is :

$$A = \frac{W}{C \; K \; P_1 \; K_b} \sqrt{\frac{TZ}{M}}$$

Where :

W = Relief flow, lbs/h
Z = Compressibility factor
T = Inlet vapor temperature, °R
C = Coefficient (from figure 6.1, 6.2)
K = Coefficient of discharge (0.975 unless vendor data available)
P_1 = Upstream pressure, psia. Set pressure 1.1 for blocked outlet, CV failure or 1.2 for fire plus 14.7 psia
K_b = Capacity correction factor (from figure 6.6)
M = Molecular weight of the vapour

6.5. RELIEF FOR GAS EXPANSION DUE TO FIRE

$$A = A_s \frac{F'}{\sqrt{P_1}} \qquad F' = \frac{0.1406}{C\,K} \; \frac{T^{1.25}}{T^{0.6506}}$$

A = effective discharge area of valve ins^2
A_s = exposed surface area of vessel ft^2
T = 1560 - T
T = temp. at relief pressure .°R

6.6. STEAM RELIEF

$$A = \frac{W}{50 \, P_1 \, K_{sh}} \quad ins^2$$

P_1 = Set pressure x 1.03. (ASME Power)
or 1.1. (ASME Unfired vessels)

K_{sh} - superheat correction factor table 6.1.

6.7. STANDARD RELIEF VALVE ORIFICE SIZES

The following table may be used for estimating the relief valve size based upon the effective discharge areas calculated as in paragraphs 6.3. through 6.6. :

Orifice letter	Nozzle Effective Area sq. inches	Normal size Designation
D	0.110	1D2
E	0.196	1E2
F	0.307	1 1/2 F$_2$*
G	0.503	2G3*
H	0.785	2H3
J	1.287	2J3 or 3J4*
K	1.838	3K4 or 3K6
L	2.853	3L4 or 4L6
M	3.600	4M6
N	4.340	4N6
P	6.379	4P6
Q	11.045	6Q8
R	16.000	6R8 or 6R10
T	26.000	8T10

* Avoid using 2 1/2 inch outlet flanges (F and G orifices)

** Avoid using 2 1/2 inch inlet flange (J orifice)

7.0. REFERENCES + LITERATURE

7.1. DESIGN GUIDE Flares-Vents-Relief and Blowdown systems
CFP MAY 1984 - TEP/DP/EXP

7.2. API 520 API 14C
API 521 API 14E

7.3. Det Norske Veritas : Technical Notes fixed offshore installations
Norweigen Petroleum Directorate : Guidelines for safety evaluations of platform conceptual designs.

TABLE 6-1—Superheat Correction Factors for Safety Valves in Steam Service

Set Pressure (Pounds per Square Inch Gage)	Saturation Temperature (Degrees Fahrenheit)	Correction Factor K_{sh}											
		0.99	0.98	0.97	0.96	0.95	0.94	0.93	0.92	0.91	0.90	0.89	0.88
		Total Temperature (Degrees Fahrenheit)											
10	240	269	305	335	368	400	428	460	492	520	545	570	595
20	259	286	315	343	375	405	433	463	492	518	542	565	590
40	287	310	335	357	382	410	440	467	493	515	540	561	585
60	308	330	350	370	390	422	450	472	495	515	537	560	580
80	324	345	365	385	405	432	460	478	497	515	535	556	580
100	338	360	375	395	415	440	466	485	500	515	535	555	580
120	350	370	388	405	425	450	475	490	505	520	537	557	581
140	361	—	398	415	435	455	480	497	510	525	540	560	585
160	370	—	405	425	443	463	487	502	516	530	545	565	586
180	379	—	415	432	450	470	492	508	523	535	550	570	590
200	388	—	420	440	456	475	497	513	527	540	555	575	592
220	396	—	430	445	463	480	502	517	532	546	560	577	596
240	403	—	435	452	470	485	507	522	537	550	565	583	600
260	409	—	440	460	475	490	512	526	541	555	569	586	603
280	416	—	447	465	480	495	516	531	545	558	573	590	606
300	422	—	452	470	485	500	520	535	550	562	577	593	610
350	433	—	465	480	496	512	530	545	558	572	586	602	618
400	448	—	475	492	508	523	540	553	566	580	595	610	626
500	470	—	495	513	526	543	557	568	582	597	610	625	646
600	489	—	512	530	543	556	570	585	596	610	625	638	655
800	520	—	545	558	570	585	597	610	625	635	650	665	680
1,000	546	—	567	582	595	608	620	633	645	660	675	688	705
1,250	574	—	593	605	620	630	640	655	668	681	696	710	725
1,500	597	—	—	630	642	653	664	676	688	702	715	728	744
1,750	619	—	—	647	660	670	680	692	704	717	730	743	759
2,000	637	—	—	665	675	685	696	708	719	732	745	757	773
2,500	670	—	—	690	702	712	723	733	742	755	766	780	795
3,000	697	—	—	713	723	733	742	751	762	773	785	795	812

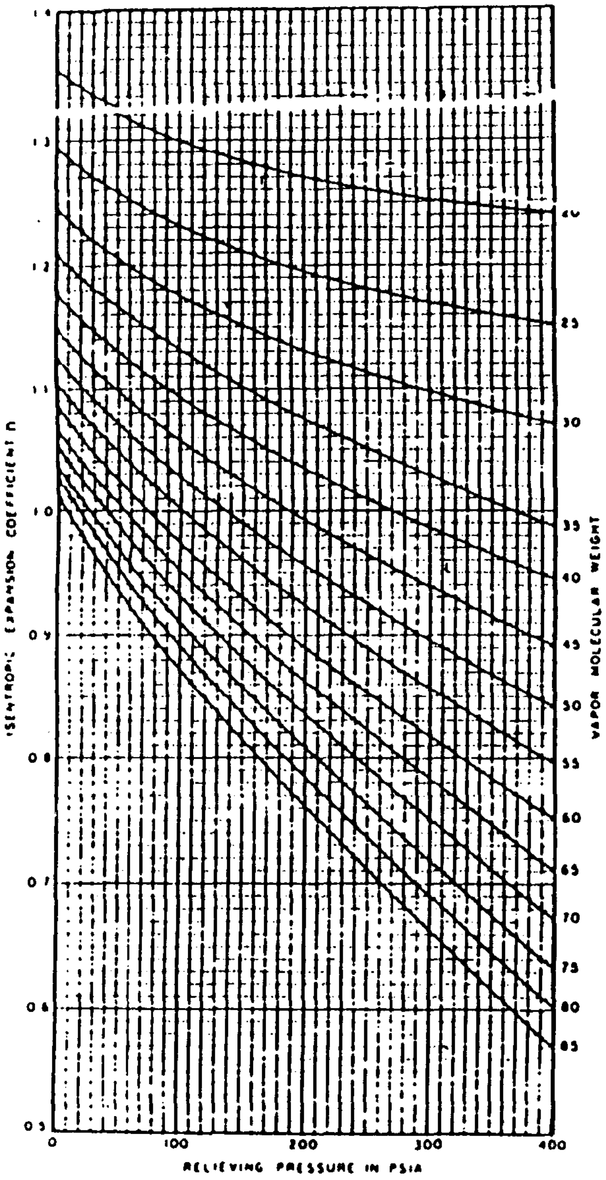


Figure 6-1 – Curves for Paraffin Hydrocarbons Expanding from Relieving Pressures At or Nea Saturation to Critical-Flow Pressures

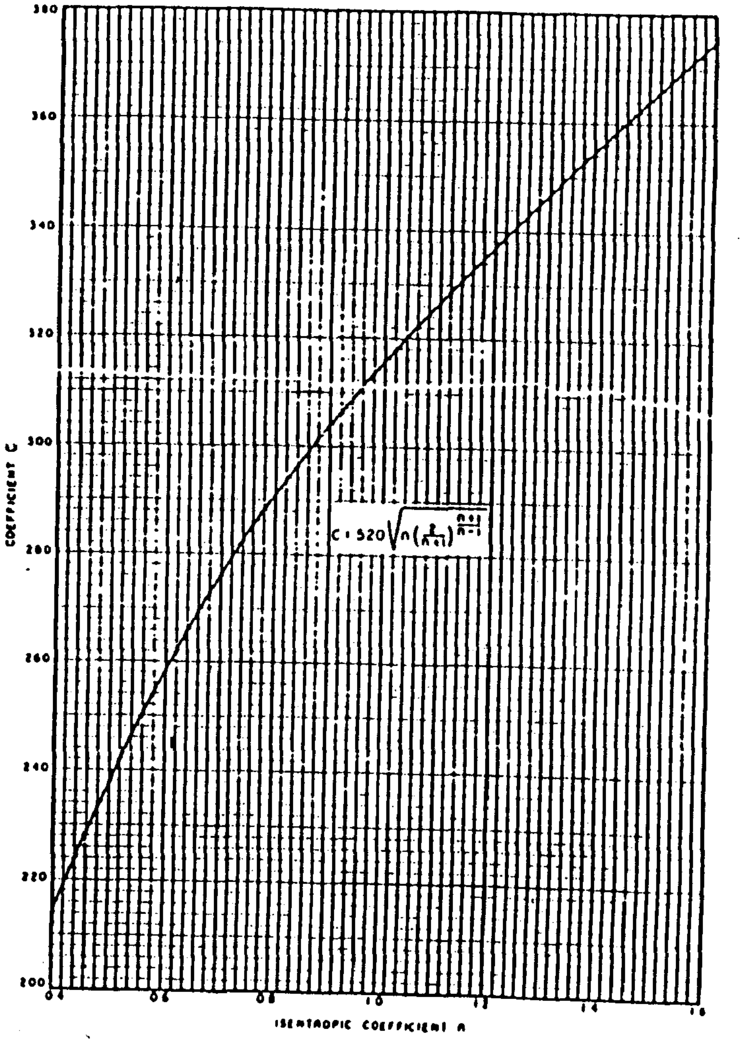


Figure G-2 – Curve for Evaluating Coefficient C in Safety Relief Valve Flow Formula from Isentropic Coefficient n

When *k* or *n* cannot be determined, it is suggested to let *C* = 315.

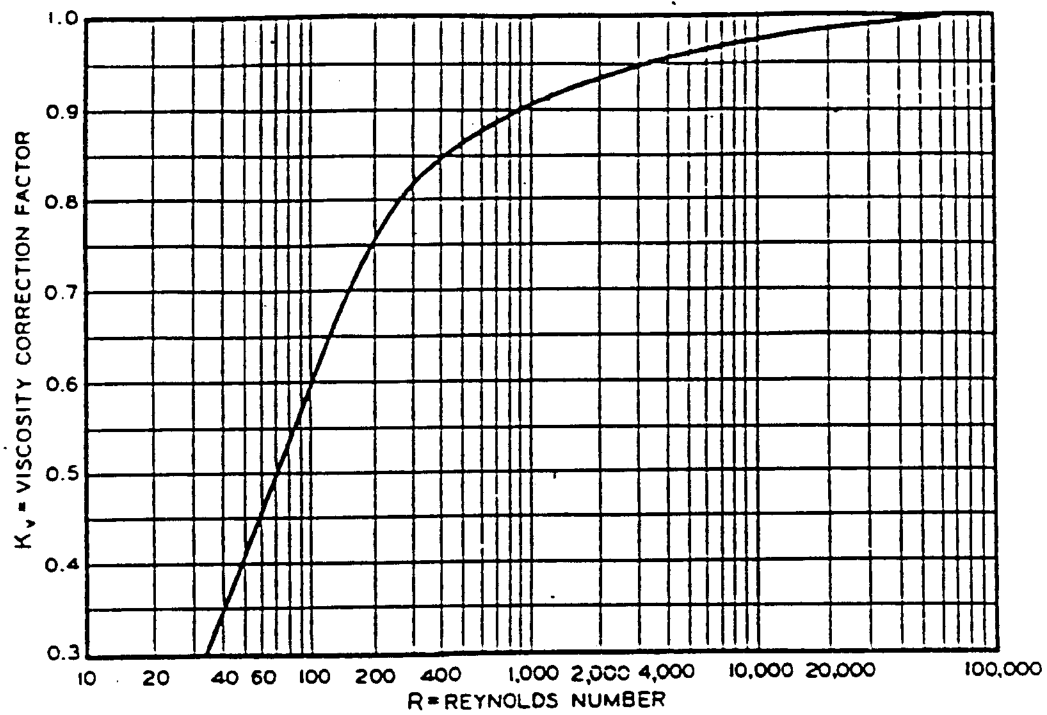
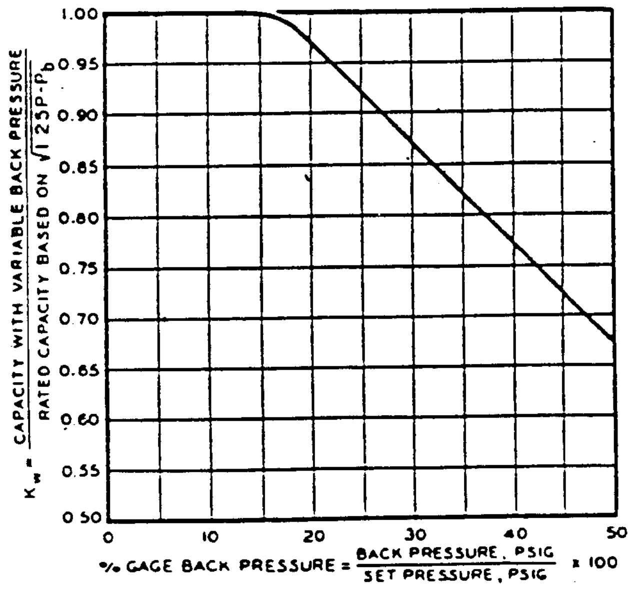
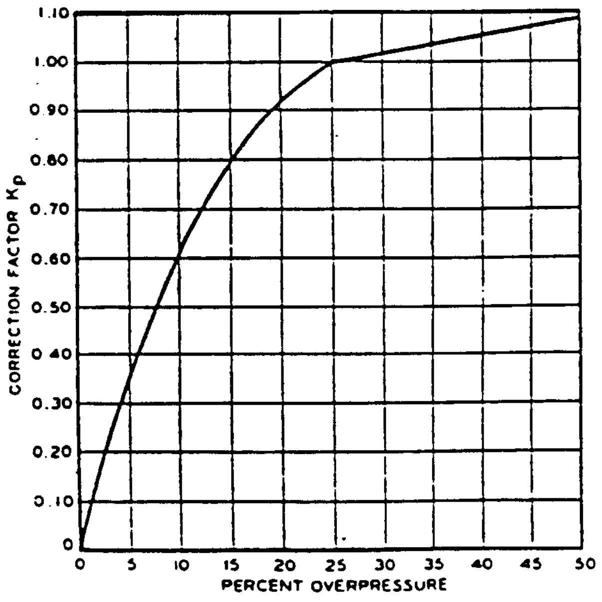


Figure 6·3 -Capacity Correction Factor Due to Viscosity K_v



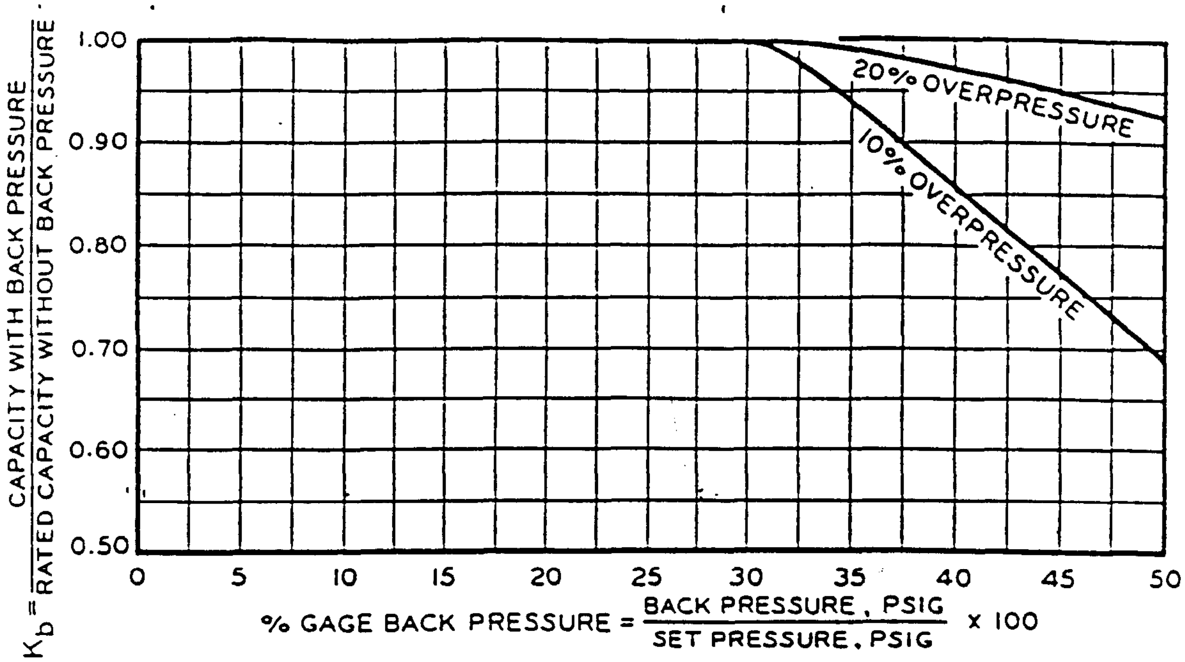
NOTE: The above curve represents a compromise of the values recommended by a number of relief-valve manufacturers. This curve may be used when the make of the valve is not known. When the make is known, the manufacturer should be consulted for the correction factor.

Figure 6·4 -Variable or Constant Back-Pressure Sizing Factor K_w for 25 Percent Overpressure on Balanced Bellows Safety-Relief Valves (Liquids Only)



NOTE: The above curve shows that up to and including 25 percent overpressure, capacity is affected by the change in lift, the change in orifice discharge coefficient, and the change in overpressure. Above 25 percent, capacity is affected only by the change in overpressure.
Valves operating at low overpressures tend to "chatter"; therefore, overpressures of less than 10 percent should be avoided.

Figure 6 5 -Capacity Correction Factors Due to Overpressure for Relief and Safety-Relief Valves in Liquid Service



NOTE: The above curves represent a compromise of the values recommended by a number of relief valve manufacturers and may be used when the make of valve or the actual critical-flow pressure point for the vapor or gas is unknown. When the make is known, the manufacturer should be consulted for the correction factor.

These curves are for set pressures of 50 pounds per square inch gage and above. They are limited to back pressure below critical-flow pressure for a given set pressure. For subcritical-flow back pressures below 50 pounds per square inch gage, the manufacturer must be consulted for the values of K_b.

Figure 6·6a–Variable or Constant Back-Pressure Sizing Factor K_b for Balanced Bellows Safety-Relief Valves (Vapors and Gases)

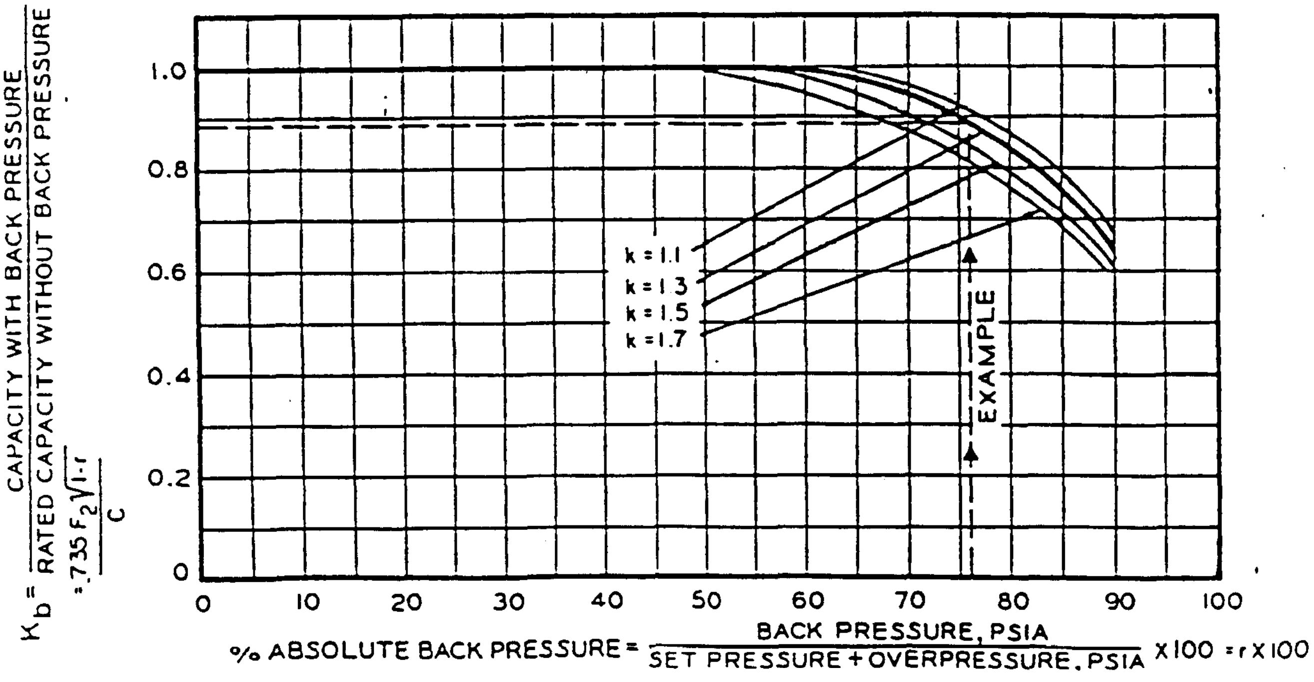


Figure 6·6b–Constant Back Pressure Sizing Factor K_b, For Conventional Safety-Relief Valves (Vapors and Gases Only)

APPENDIX A

SAMPLE CALCULATIONS FOR SIZING A FLARE STACK

A.1 General

This appendix presents examples of the two methods for sizing a flare stack based on the effects of radiation. The two methods are the "simple" approach presented in Section 4 and the more specific approach using Brzustowski and Sommer's method. Height and location should also be considered, based on gas dispersion if the flame is extinguished (see 4.4.1.4).

A.2 Example 1: Sizing a Flare Stack Using the Simple Approach

In this example, the basic data are as follows: The material flowing is hydrocarbon vapors. The flow rate, W, is 100,000 pounds per hour (12.6 kilograms per second). The average molecular weight of the vapors, M, is 46.1. The flowing temperature, T, is 760 degrees Rankine (300 F) [422 kelvins (149 C)]. The heat of combustion is 21,500 British thermal units per pound (5×10^4 kilojoules per kilogram). The ratio of the specific heats in the gas, k, is 1.1. The flowing pressure at the flare tip is 14.7 podnds per square inch absolute (101.3 kilopascals absolute). The design wind velocity is 20 miles per hour (29.3 feet per second) [32.2 kilometers per hour (approximately 8.9 meters per second)]

A.2.1 CALCULATION OF FLARE DIAMETER

The Mach number is determined as follows (see 5.4.3.1, Item 1).

$$\text{Mach} = (1.702)(10^{-5})\frac{W}{Pd^2}\sqrt{\frac{T}{kM}}$$

In metric units, this translates to:

$$\text{Mach} = (11.61)(10^{-2})\frac{W}{Pd^2}\sqrt{\frac{T}{kM}}$$

For Mach = 0.2, the flare diameter is calculated as follows:

$$0.2 = (1.702)(10^{-5})\frac{100{,}000}{14.7d^2}\sqrt{\frac{760}{(1.1)(46.1)}}$$
$$d^2 = 2.24$$
$$d = 1.5 \text{ feet (inside diameter)}$$

In metric units, this translates to:

$$0.2 = (11.61)(10^{-2})\frac{12.6}{101.3d^2}\sqrt{\frac{422}{(1.1)(46.1)}}$$
$$d^2 = 0.209$$
$$d = 0.46 \text{ meter (inside diameter)}$$

For Mach = 0.5, the flare diameter is calculated as follows:

$$d^2 = 0.897$$
$$d = 0.95 \text{ foot (inside diameter)}$$

In metric units, this translates to:

$$d^2 = 0.0833$$
$$d = 0.29 \text{ meter (inside diameter)}$$

A.2.2 CALCULATION OF FLAME LENGTH

The heat liberated, Q, in British thermal units per hour (kilowatts), is calculated as follows (see Figures 6A and 6B):

$$Q = (100{,}000)(21{,}500)$$
$$= 2.15 \times 10^9 \text{ British thermal units per hour}$$

In metric units, this translates to:

$$Q = (12.6)(50 \times 10^3)$$
$$= 6.3 \times 10^5 \text{ kilowatts}$$

From Figures 6A and 6B, the flame length, L, is 170 feet (52 meters) (see Figure A-1).

A.2.3 CALCULATION OF FLAME DISTORTION CAUSED BY WIND VELOCITY (SIMPLE CALCULATION)

The vapor flow rate is determined as follows:

$$\text{Flow} = \left(\frac{100{,}000}{3600}\right)\left(\frac{379.1}{46.1}\right)\left(\frac{760}{520}\right)$$
$$= 333.9 \text{ actual cubic feet per second}$$

In metric units, this translates to:

$$\text{Flow} = (12.6)\left(\frac{22.4}{46.1}\right)\left(\frac{422}{273}\right)$$
$$= 9.46 \text{ actual cubic meters per second}$$

NOTE: In the calculations above, the volume of a perfect gas per pound-mole under English standard conditions (60 F and 14.7 pounds per square inch absolute) is 379.1 cubic feet. The volume of a perfect gas per kilogram-mole under metric standard conditions (0 C and 101.3 kilopascals absolute) is 22.4 cubic meters.

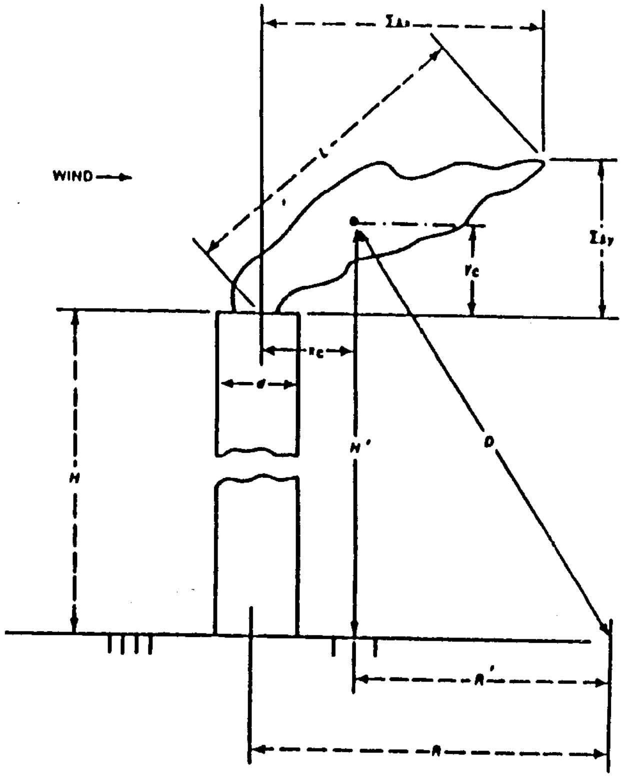


Figure A-1—Dimensional References for Sizing a Flare Stack

The flame distortion caused by wind velocity is calculated as follows (see Figure 7):

$$\frac{U_\infty}{U_j} = \frac{\text{Wind velocity}}{\text{Flare tip velocity}}$$

The flare tip exit velocity, U_j, may be determined as follows (see A.3.2 for another method of calculating U_j):

$$U_j = \frac{\text{Flow}}{\frac{\pi d^2}{4}}$$

For Mach = 0.2,

$$U_j = \frac{333.9}{\frac{\pi(1.5)^2}{4}}$$
$$= 189 \text{ feet per second}$$

In metric units, this translates to:

$$U_j = \frac{9.46}{\frac{\pi(0.46)^2}{4}}$$
$$= 56.9 \text{ meters per second}$$

For Mach = 0.5,

$$U_j = \frac{333.9}{\frac{\pi(0.95)^2}{4}}$$
$$= 471 \text{ feet per second}$$

metric units, this translates to:

$$U_i = \frac{9.46}{\frac{\pi(0.29)^2}{4}}$$
$$= 143.2 \text{ meters per second}$$

At Mach = 0.2,

$$\frac{U_\infty}{U_j} = \frac{29.3}{189} = 0.155$$
$$\sum\frac{\Delta y}{L} = 0.35$$
$$\sum\frac{\Delta x}{L} = 0.85$$
$$\sum\Delta y = (0.35)(170) = 59.5 \text{ feet}$$
$$\sum\Delta x = (0.85)(170) = 144.5 \text{ feet}$$

metric units, this translates to:

$$\frac{U_\infty}{U_j} = \frac{8.9}{56.9} = 0.156$$
$$\sum\frac{\Delta y}{L} = 0.35$$
$$\sum\frac{\Delta x}{L} = 0.85$$
$$\sum\Delta y = (0.35)(52) = 18.2 \text{ meters}$$
$$\sum\Delta x = (0.85)(52) = 44.2 \text{ meters}$$

At Mach = 0.5,

$$\frac{U_\infty}{U_j} = \frac{29.3}{471} = 0.062$$
$$\sum\frac{\Delta y}{L} = 0.53$$
$$\sum\frac{\Delta x}{L} = 0.72$$
$$\sum\Delta y = (0.53)(170) = 90.1 \text{ feet}$$
$$\sum\Delta x = (0.72)(170) = 122.4 \text{ feet}$$

metric units, this translates to:

$$\frac{U_\infty}{U_j} = \frac{8.9}{143.2} = 0.062$$
$$\sum\frac{\Delta y}{L} = 0.53$$
$$\sum\frac{\Delta x}{L} = 0.72$$
$$\sum\Delta y = (0.53)(52) = 27.6 \text{ meters}$$
$$\sum\Delta x = (0.72)(52) = 37.4 \text{ meters}$$

A.2.4 CALCULATION OF REQUIRED FLARE STACK HEIGHT

For the basis of the calculations in this section, refer to 4.4.1.3. Refer to Figure A-1 for dimensional references.

The design basis is as follows: The fraction of heat radiated, F, is 0.3. The heat liberated (see A.2.2), Q, is 2.15×10^9 British thermal units per hour (6.3×10^5 kilowatts). The maximum allowable radiation, K, at 150 feet (45.7 meters) from the flare stack is 2000 British thermal units per hour per square foot (6.3 kilowatts per square meter).

In equation (3) from Section 4, assume $\tau = 1.0$. The distance from the flame center to the grade-level boundary (that is, the object being considered), D, is then calculated as follows:

$$D = \sqrt{\frac{\tau FQ}{4\pi K}}$$
$$= \sqrt{\frac{(0.3)(2.15)(10^9)}{(4)(3.14)(2000)}}$$
$$= 160.2 \text{ feet}$$

In metric units, this translates to:

$$D = \sqrt{\frac{\tau FQ}{4\pi K}}$$
$$= \sqrt{\frac{(0.3)(6.3)(10^5)}{(4)(3.14)(6.3)}}$$
$$= 48.9 \text{ meters}$$

The physical arrangement shown in Figure A-1 is the basis of the following calculations.

At Mach = 0.2, the flare stack height, H, is calculated as follows:

$$H' = H + \tfrac{1}{2}\Delta y$$
$$R' = R - \tfrac{1}{2}\Delta x$$
$$\sum\Delta y = 59.5 \text{ feet}$$
$$\sum\Delta x = 144.5 \text{ feet}$$

(See A.2.3.)

$$R' = 150 - \tfrac{1}{2}(144.5) = 78 \text{ feet}$$
$$D^2 = R'^2 + H'^2$$
$$160^2 = 78^2 + H'^2$$
$$H'^2 = 25{,}600 - 6084 = 19{,}516$$
$$H' = 140 \text{ feet}$$
$$H = 140 - \tfrac{1}{2}(60) = 110 \text{ feet}$$

In metric units, this translates to:

$$H' = H + \tfrac{1}{2}\Delta y$$
$$R' = R - \tfrac{1}{2}\Delta x$$
$$\sum\Delta y = 18.2 \text{ meters}$$
$$\sum\Delta x = 44.2 \text{ meters}$$

(See A.2.3.)

$$R' = 45.7 - \tfrac{1}{2}(44.0) = 23.7 \text{ meters}$$
$$D^2 = R'^2 + H'^2$$
$$48.9^2 = 23.7^2 + H'^2$$
$$H'^2 = 2391.2 - 561.7 = 1829.5$$
$$H' = 42.8 \text{ meters}$$
$$H = 42.8 - \tfrac{1}{2}(18.2) = 33.7 \text{ meters}$$

At Mach = 0.5, H is calculated as follows:

$$H' = H + \tfrac{1}{2}\Delta y$$
$$R' = R - \tfrac{1}{2}\Delta x$$
$$\sum\Delta y = 90.1 \text{ feet}$$
$$\sum\Delta x = 122.4 \text{ feet}$$

(See A.2.3.)

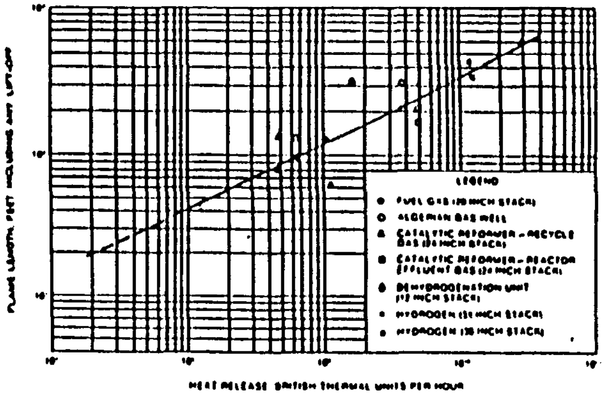
$$R' = 150 - \tfrac{1}{2}(122) = 89 \text{ feet}$$
$$D^2 = R'^2 + H'^2$$
$$160^2 = 89^2 + H'^2$$
$$H'^2 = 25{,}600 - 7921 = 17{,}679$$
$$H' = 133 \text{ feet}$$
$$H = 133 - \tfrac{1}{2}(90) = 88 \text{ feet}$$

In metric units, this translates to:

$$H' = H + \tfrac{1}{2}\Delta y$$
$$R' = R - \tfrac{1}{2}\Delta x$$
$$\sum\Delta y = 27.6 \text{ meters}$$
$$\sum\Delta x = 37.4 \text{ meters}$$

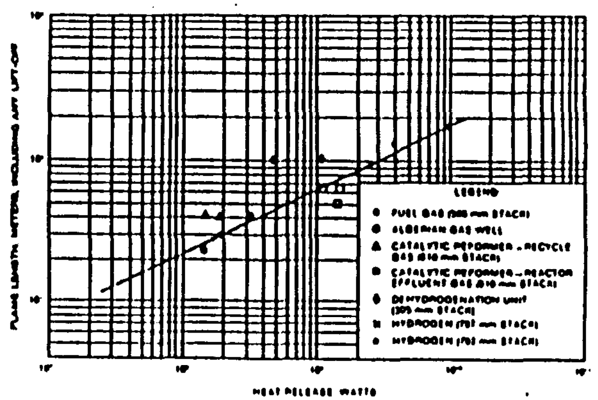
(See A.2.3.)

$$R' = 45.7 - \tfrac{1}{2}(37.4) = 27.0 \text{ meters}$$
$$D^2 = R'^2 + H'^2$$
$$48.9^2 = 27.0^2 + H'^2$$
$$H'^2 = 2391.2 - 729 = 1662.2$$
$$H' = 40.8 \text{ meters}$$
$$H = 40.8 - \tfrac{1}{2}(27.6) = 27 \text{ meters}$$



Note: Multiple points indicate separate observations or different assumptions of heat content.

Figure 8A—Flame Length Versus Heat Release: Industrial Sizes and Releases (English Units)



Note: This figure was developed from Figure 8A. Multiple points indicate separate observations or different assumptions of heat content.

Figure 8B—Flame Length Versus Heat Release: Industrial Sizes and Releases (SI Units)

Dispersion of gases from cold vent stack

Table of Contents

1 INTRODUCTION

The Welton Gathering Centre process plant is provided with an incinerator for disposal of excess associated gases and relief gases. During upsets on the incinerator e.g. fan failure or a plant initiated trip, the incinerator is isolated and the gases are disposed of through a cold vent via a bursting disc arrangement (see Fig 1).

The gases routed to the cold vent contain high levels of H2S and adequate dilution of this component with air is therefore required during dispersion for safe disposal. The option of replacing the existing incinerator with a new and larger one, means that the cold vent would still be required. Dispersion calculations were therefore carried out by Group Environmental Services (GES), London to confirm that safe disposal of the vented gases could be achieved.

Details of the atmospheric dispersion study and results are presented below. The following areas are covered :-

- basis of the study
- results of the dispersion modelling
- discussions of results
- operating experience with the cold vent
- conclusions and recommendations

2 BASIS OF THE STUDY

The dispersion calculations were performed by Group Environmental Services (GES) in London. The basis of these calculations [1] is given in the sections following.

2.1 GAS COMPOSITIONS

The composition of the gases that could be vented to the atmosphere can be wide ranging because :-

- the Welton G.C. receives crude from a number of wells at varying flows and varying H2S content. Hence the amount and composition of the gases normally produced varies;

- the relief gas composition which could superimpose on normal flows vary depending on the relief scenario. Accurate prediction of the relief flow and composition is not always possible.

Therefore for the dispersion calculations reported here, a number of vented gas data compositions were prepared from two sources (Table 1) :-

I) The Original Design composition - these were extracted from the Incinerator Work Pack 18, Vol. 1. They were based on the design wellfluid (C-site) and still represent the sourest gases that could be obtained from Gathering Plant HP/relief header. Calculations for the following cases were carried out -

a) Case 1 - Highest gas and H2S flows, Gas I;
b) Case 2 - Lowest H2S concentration, Gas A;
c) Case 3 - Lowest H2S flows, Gas E and
d) Case 4 - Lowest total flow, Gas B.

II) The preliminary revised design cases - these are based on the Welton Upgrading design material balance and represent conditions in the plant when production from C site is diluted by production from the less sour wells (A and B sites). The combined HP, LP and acid gas stream compositions were used for the dispersion calculations. The following cases were considered :-

a) Case 1 - based on the total of normal flows in the gas headers;
b) Case 2 - startup flow based on 300 BOPD;
c) Case 3 - similar to Case 2 but with twice the H2S concentration. This can be considered as starting up

with production from C-site.

d) Case 4 - based on normal flows superimposed with fire relief flow in the gas headers.

2.2 COLD VENT

The cold vent is a 16 inch pipe erected vertically with a 6 inch top section acting as a nozzle. Detail dimensions of the cold vent as used in this dispersion study are shown in Fig 2.

Process gas is normally isolated from the cold vent by means of four bursting discs - two in use in series and two spare, as shown in Fig 1. Should these rupture, a common alarm signal is produced in the control room and the incinerator is tripped which in turn causes a general plant shutdown. Thus the duration and quantities of the emissions is minimised.

2.3 H2S CONCENTRATION MONITORING

The following locations were selected for monitoring the H2S concentration during dispersion of gases from the cold vent :-

a) At positions in the plant above grade which could be manned, of which the following were selected :-

Crude stripper tower	- 14.9 m
Amine contactor	- 13.6 m
Crude tanks	- 9.0 m
Incinerator	- 12.5 m

b) At grade level at various locations within the Welton Gathering Centre perimeter fence, car park and workshop areas.

c) At farm houses in the near vicinity and in particular, the Barfields farm.

The co-ordinates for these receptors are shown in Table 1.

3 RESULTS

Details of the results are given in a GES memo, Ref. [2] ; an extract of the results is given in this section. The results of H2S concentrations at various locations are quoted in mg/m3. To convert these figures to ppmv then the quoted figures should be multiplied by a factor of 0.7121.

Tables 2 - 9 show the maximum ground level concentrations for a range of weather conditions and the cases specified. The wind has been arbitrarily set to SW, so the location of the point of the maximum ground level concentration is of no significance, but the distance from source is of use.

Table 10 - 17 shows concentration at the seven sites specified for the same range of weather conditions and the same eight cases. However, in these tables, the wind direction has been deliberately chosen to place the specified site directly downwind of the vent. Thus each concentration represents the worst possible condition at each site.

The result presented in the Tables 2 - 17 are the 3 minute average concentrations at the receptors. To extrapolate the 3 minute average concentration to longer time average concentrations, the following can be applied :-

Cx = Cp * (3 / Tx) **0.2

where Cx is concentration average for time Tx
Cp is the 3 minute average concentration
Tx is the new time for average concentrations.

Thus, for a 15 minute average concentrations, the 3 minute average concentrations have to be multiplied by a factor of 0.72. Similarly, for an 8 hour average concentration, the factor is 0.36.

For time average concentrations of less than 3 minute, the above correlation does not apply. However as a guide, a similar approximation to that made for odour nuisance investigations (where the 5 second average concentrations are relevant) can be used. Thus, to convert the 3 minute averages to 5 second averages, the results in weather category A should be multiplied by 10, and the remaining weather categories should be multiplied by 5.

4 DISCUSSIONS

This section discusses the interpretation of the results given in the above section. It must be stressed that the interpretation of the results and inferences made here concerning the H2S level with respect to safety, occupational health and nuisances are mainly those of the author. Advice from Group Safety and Occupational Health was obtained verbally and is incorporated.

The following observations are made on the results :-

a) The odour threshold for H2S is 0.00066 mg/m3 for 5 sec. average time. Tables 2 - 9 shows that the 3 minute average ground level concentrations (1/5th of 5 second averages for most weather categories) exceed the odour threshold. The use of the cold vent would thus result in odorous emission which would be perceived at fairly remote locations e.g. the Barfields farm, when the wind is blowing in that direction.

It is noted that there are no evidence that odorous emissions due to H2S at low level are a health hazard, but complaints relating the two may still be received.

b) The cold vent facility is provided with two bursting disc installed back to back in the duty line with a parallel spare set provided. Should these rupture, an alarm signal is produced and the incinerator is shutdown. The latter also causes a general plant shutdown thereby minimising the amounts of emissions. The release of H2S containing gases through the cold vent is therefore restricted to the HP/Relief header depressuring or the depressuring of the separators in the HP gas blowthrough scenario. It is not possible to quantify these periods.

c) Tables 10 - 17 show the 3 min average concentrations at the seven particular receptors specified; in all cases the odour threshold is exceeded.

If the concentrations are adjusted to give 8 hr. time interval averages by multiplying by a factor of 0.36 then in some cases the Long Term Exposure Limit of 14 mg/m3 is exceeded. This occurs mainly at elevated receptors and in particular the crude stripper and the amine contactor/regenerator.

If the 3 min average concentration is converted to give the 15 min average value by multiplying by 0.72, then the Short Term Exposure Limit of 21 mg/m3 is also exceeded at the same receptors.

It is concluded that if cold venting were proposed as a normal operation then it would not be acceptable to BP or to environmental authorities. However, it is noted that the original design intentions were that the discharge would be of limited duration and of low probability.

d) In assessing the safety implications of the discharge, the instantaneous level of H2S perceived is relevant. It is noted that the closest approximation of this is the 3 min average concentrations which shows that at elevated levels, and in particular the top of the amine contactor/regenerator towers, concentrations in excess of 42 mg/m3 are predicted. At these levels, the H2S would cause eye and respiratory tract irritation. This is unpleasant and would be a signal for evacuating the area.

It is noted that if the 5 sec. average concentration (5 times the 3 min average concentrations in most cases) are considered, then levels in excess of 140 mg/m3 are predicted. These could cause a loss of the sense of smell and result in a loss of signal for evacuating from the affected areas. However, the levels are below 700 mg/m3 which would cause a loss of consciousness within 15 mins of exposure. It should also be noted that such duration for exposures are unlikely as explained in b) above.

It must be noted that the effects of H2S depends on a number of variables and above limits are only for guidance. It is noted that people who are regularly exposed to even very low concentrations eventually become unable to detect the gas by smell.

e) The models used for the dispersion calculations lose their validity at distances less than 10 metres of the source. In this area, it is also noted that the mechanism for dispersion is different and low flow emissions have greater impact on the resultant ground level concentrations especially if the gaseous emissions are denser than air as in this particular case. Within this 10 meter area of the Gathering Centre is the location of the pig receivers and wax traps which may well be manned in event of the cold vent operating.

f) Analysis of the results given in Tables 2 - 10 shows that the dilutions resulting from cold vent dispersion increase as the flow of the emission decrease (see Fig 3) but is generally less affected by the concentration of the pollutant in the emissions. Low flows would result during plant startup when plant upsets generally arise and reduction of H2S content in the gas header by starting up using less sour wells would reduce the impact of cold venting during this period.

g) The concentrations of the hydrocarbons at the various receptors after dispersion has not been calculated; this can be roughly estimated from the dilutions imparted on the pollutant during dispersion. Estimated hydrocarbon concentration for two receptors has been calculated for Original Design Case 1 and shown below.

i) Amine contactor, maximum H2S concentration is 50.7 mg/m3 which corresponds to a dilution of 249. Therefore, max concentration of hydrocarbon is 0.40 %.

ii) Incinerator, maximum H2S concentration is 1.9 mg/m3 which corresponds to a dilution of 6636. Therefore, max concentration of hydrocarbon is 0.02 %.

It will be seen that in both cases the concentrations predicted are below the Lower Flammability Level assumed 2 % (methane) for this purpose.

5 OPERATING EXPERIENCE

During commissioning of the Welton Gathering Centre plant, problems were experienced with the use of graphite bursting discs which ruptured at low bursting pressure. These may have been due to fatigue caused by pulsation of gases in the HP/relief header. This was resolved by reducing pulsation from the burner and the installation of a bursting disc of composite material which has proved reliable.

After commissioning, the cold vent has operated several times mainly as a result of instrument failure on the incinerator. However the production through the plant was below 50 % of the full capacity. With the recent modifications to the incinerator, it has been possible to increase the flow through the Gathering Centre. This increase has resulted in the header pressures being close to the bursting disc pressure. This could increase the frequency of operation of the cold vent resulting from surges through the plant.

If the existing incinerator is replaced by a new incinerator in the next Phase of the Welton Upgrading then the design of the incinerator and the setting of the bursting disc should be such that frequency of rupture due to surges is eliminated.

6 CONCLUSIONS AND RECOMMENDATIONS

a) The study has concentrated on the dispersion of H_2S based on a number of the Original Design and the Welton Upgrading gas compositions. It has been predicted that emissions would be odorous at ground level at short and long distances from the cold vent.

b) At elevated areas, for example, the top of the amine contactor/regenerator towers, the predicted levels of H_2S are higher than the Short Term Exposure limit. It is therefore recommended that access to the elevated areas close to the cold vent be restricted. Adequate warnings should be displayed at access points of the elevated areas and BA sets must be immediately available.

c) The cold vent is located close to the incinerator and the pig receiving areas. These areas are likely to be manned during operation of the cold vent. An audible alarm which is activated from a bursting disc failure has therefore been installed to provide warning to personnel. Regular testing should establish that this alarm, located near incinerator control panel, provides adequate alarm near the pig receiving area.

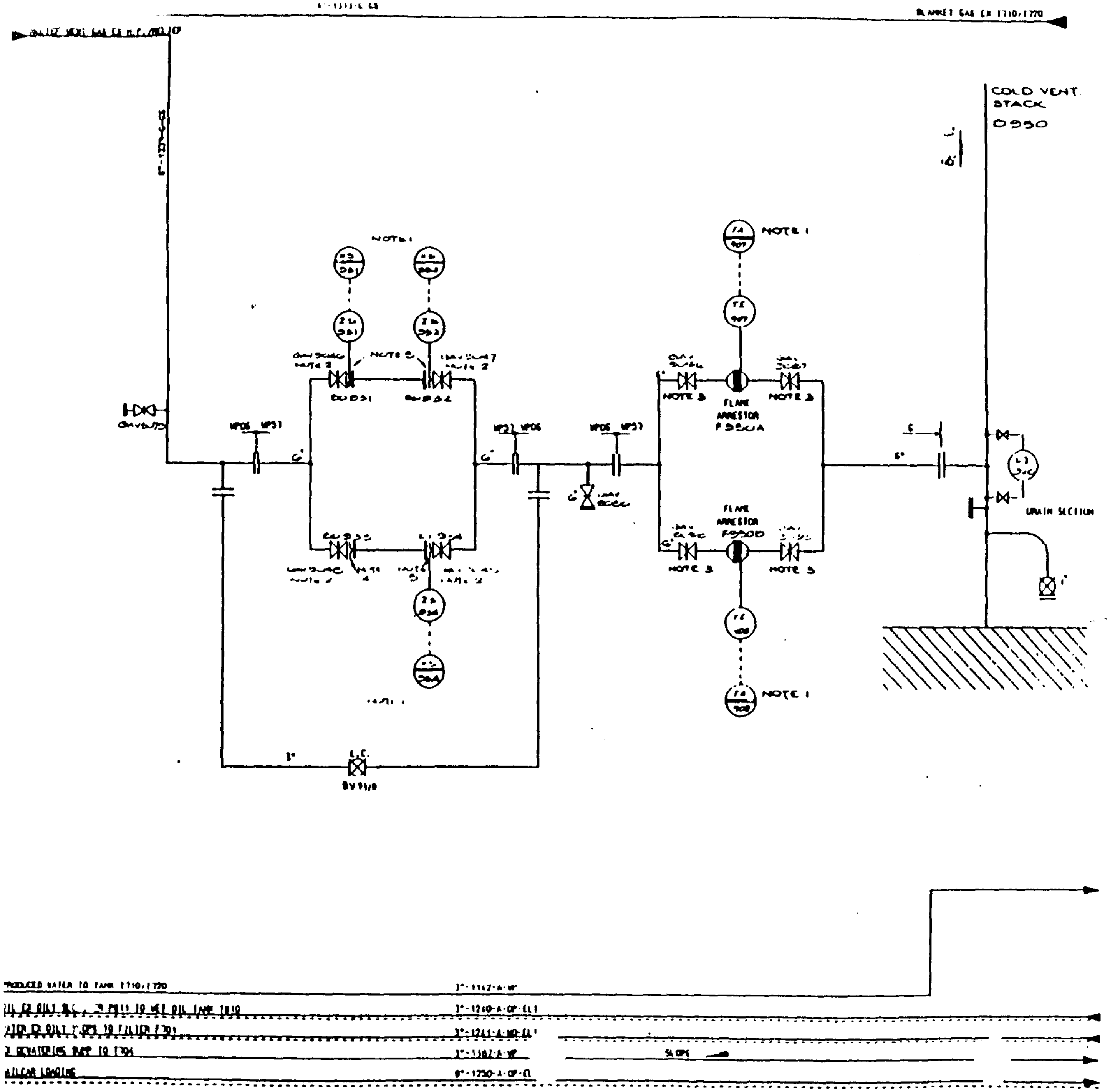
d) The plant licence was based on utilising the cold vent on failure of incinerator. The results of this study show that the emissions would be odorous and high concentrations of H_2S are predicted at elevated areas near the cold vent. It is therefore important that if operations are changed such that the design intention of utilising the cold vent is changed, then appropriate BP authorities should be consulted.

e) Due to the higher GOR's and plant surges, it is possible that increased pressures close to the bursting disc pressures are experienced as a result of the higher gas flows. This would increase the frequency of operation of the cold vent and represents a further constraint to the maximum allowable plant throughput.

f) For the Welton Upgrading work, a number of options for provision of additional gas disposal capacity are being investigated. Due to the potential hazardous nature of the cold vent emissions, it is recommended that preference should be given to solutions which remove the need to use a cold vent. Such a route is provided if a ground flare is selected to burn gases on incinerator failure.

g) In view of the complex nature of the safety and health hazard of H2S, it is recommended that an interpretation of the GES dispersion results reported here should be obtained from HTH, Dyce. Any changes perceived from the original intentions of the using the cold vent should be similarly addressed.

c.t. J. Caven-Atack
A.A. Croll
M. Broadribb
J.A. Lewis
E.A. Mullin/R.W. Bridger
1058/97



ALARM IS INDICATED IN THE CONTROL ROOM

2. VALVES TO BE INTERLOCKED SUCH THAT ONE BURSTING DISC IS ALWAYS IN SERVICE
3. VALVES TO BE INTERLOCKED SUCH THAT ONE FLAME ARRESTOR IS ALWAYS IN SERVICE
4. COMPOSITE DISC ([illegible] 2.5 [illegible])
5. [illegible] DISC ([illegible] 2.05 [illegible])

FIGURE 1

CONTINUED ON DRG. No. 75-3E-0025

APPROVED BY

	NAME	DATE	SIGNATURE
CHECKED FOR CONTROL			
CHECKED FOR PROCESS			
CHECKED JOB ENGINEER			
APPROVED BY DESIGN COORDINATOR			

REV	DESCRIPTION	DATE	BY
C1	AS BUILT UPDATED	17-06-87	CM
C	AS BUILT	[illegible]	[illegible]
B1	[illegible]	[illegible]	[illegible]
B	ISSUED FOR CONSTRUCTION	22/8/83	ACR
A	REVISED AS PER [illegible] & FINAL ISSUE FOR APPROVAL	[illegible]	ACR
0	PRELIMINARY FOR COMMENT	25-1-83	CEG

BP PETROLEUM DEVELOPMENT LIMITED
(A SUBSIDIARY OF BP EXPLORATION COMPANY LIMITED)
[illegible] INDUSTRIAL ESTATE, DYCE, ABERDEEN AB2 0PB

PROJECT
[illegible] DEVELOPMENT
PROJECT 7260

SCALE -
DRAWN CEG

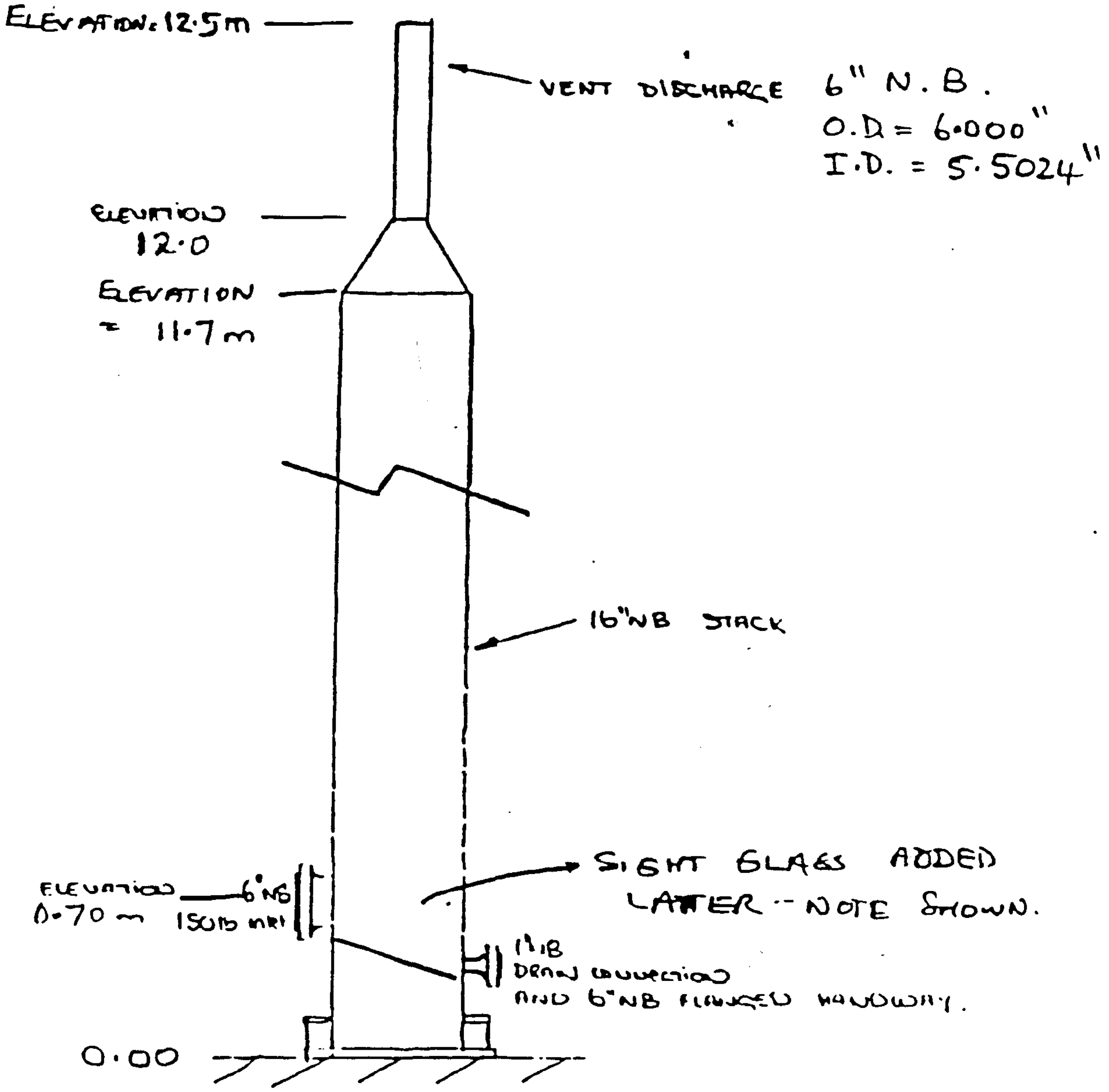
TITLE
ENGINEERING LINE DIAGRAM
COLD VENT

AS BUILT

This information is confidential and the property of Kaldair Limited and is disclosed on condition that none of the information shall be disclosed to any third party or reproduced in whole or part without the consent in writing of Kaldair Limited.

DRG. No. E

FIGURE 2



NOTES

① MATERIAL : CARBON STEEL ,

② DESIGN CODES · CP3, BS 4076

③ STACK PAINT FINISH: BP STD 141

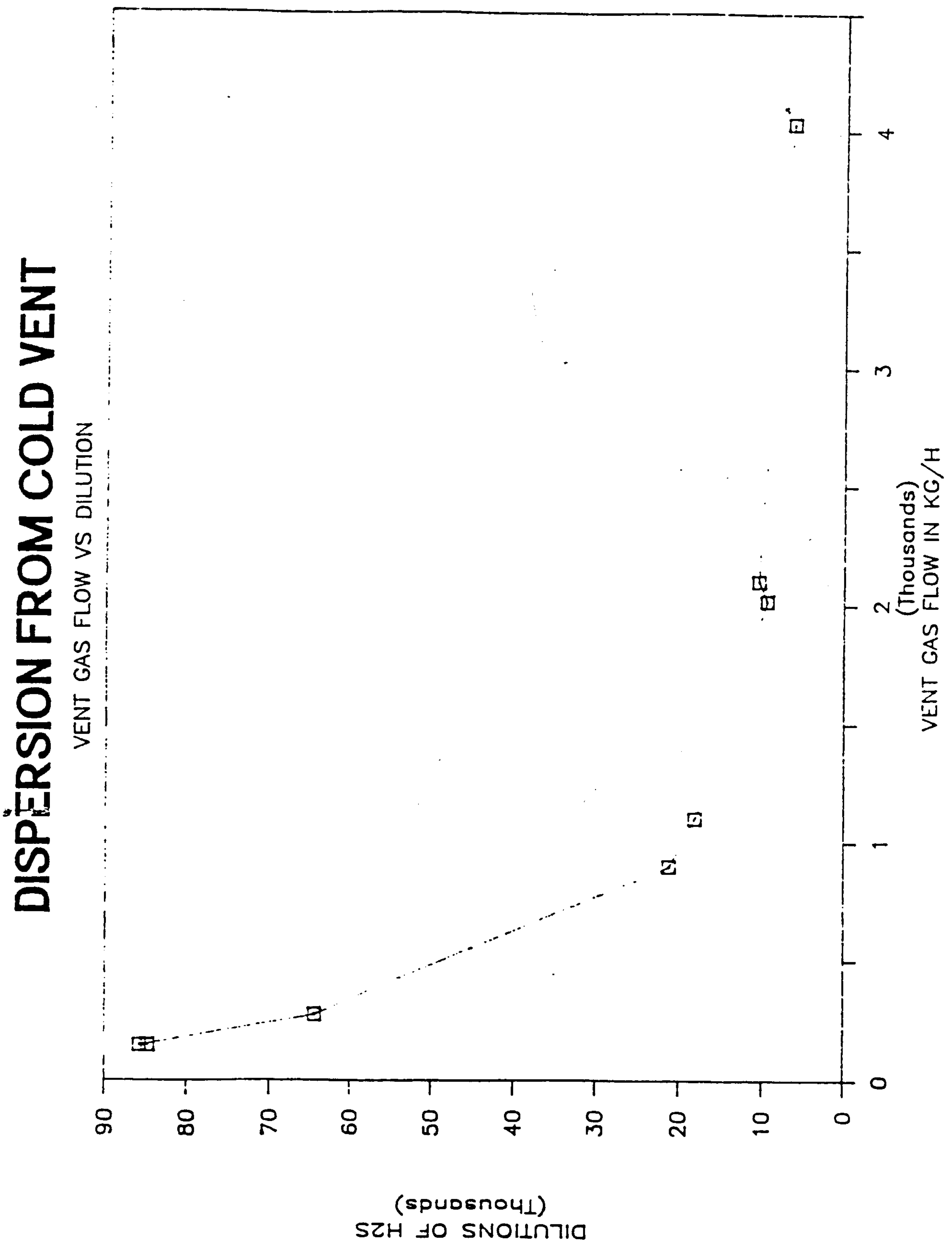
1	10.12.84	PROPOSED VENT STACK	R			R
ISSUE	DATE	DESCRIPTION	DRN.	CKD.	ENG.	PROJ ENG.

PROJECT 796	TITLE BP WELTON VENT STACK PROPOSAL (WPS7)
SCALE NTS	

Kaldair Limited DRG. No. E —

DRG. No. E

Figure 3


DISPERSION FROM COLD VENT
VENT GAS FLOW VS DILUTION
DILUTIONS OF H2S
(Thousands)
0
10
20
30
40
50
60
70
80
90
0
1
2
3
4
(Thousands)
VENT GAS FLOW IN KG/H

10. PIPES VALVES + FITTINGS

1. APPLICABILITY

- For a feasibility study a quick estimate of the line size will be required.
- For a pre-project study a better estimate of the line size will be required.
- The purpose of this guide is to size only the lines in the process unit.
- For the both the feasibility and pre-project studies abaques AFTP can be used :
  - "Pour le calcul des pertes de charges des liquides dans les conduites"
  - "Pour le calcul des pertes de charges des gaz dans les conduites"
- The line sizing depends on the service :
  - Flare lines, pipeline and riser sizing are not included on this chapter.

2. LIQUID LINES SIZING CRITERIA

See Table 1.

3. VAPOR AND STEAM LINES SIZING CRITERIA

See Table 2.

4. TWO PHASE FLOW LINE SIZING CRITERIA

The ρV^2 criteria as stated for vapor lines to be followed with :

$$\rho = \rho m = \frac{W}{\frac{Wl}{\rho l} + \frac{Wv}{\rho v}} \quad \text{in kg/m}^3$$

W = Wl + Wv = total flow rate in kg/hr

Wl = liquid flow rate in kg/hr

Wv = vapor flow rate in kg/hr

ρl = liquid density in kg/m^3

ρv = vapor density in kg/m^3

$$\text{and } V = Vm = \frac{W}{\rho m . \frac{\pi . \varnothing^2}{4} . 3600} \quad \text{m/s}$$

Øi = internal diameter of the line in m.

om and Vm are respectively the apparent density and velocity of the fluid.

- The flow regime to be checked on the figure 1 for horizontal lines and on the figure 2 for vertical lines.
- For horizontal lines slug and plug flow regimes should be avoided.
- For vertical lines slug flow regime should be avoided.

Remark : Flow chart Fig. 1 and 2 are based on author's experimental results.

5. PRESSURE DROP CALCULATIONS

5.1. MONOPHASIS FLUID (GAS OR LIQUID)

5.1.1. "ABAQUES AFTP" could be used with the correction of the line diameter such as indicated on these ABAQUES.

5.1.2. Method using MOODY or "regular" Fanning friction factors.

a. Calculate Reynolds number

$$Re = \frac{\text{Ø}i \rho V}{\mu e} =$$

Øi = line internal diameter in mm

ρ = fluid density in kg/m^3

V = velocity in m/s

μe = fluid dynamic viscosity in Cpo

Re is a dimensionless number

b. Determine the relative roughness : See Figure 3 → $\frac{\varepsilon}{D}$ =

c. Determine f = friction factor : See Figure 4 → f =

d. $\Delta P = f \times \frac{100}{\text{Ø}i} \times \frac{\rho V^2}{2g \times 10.2} = \frac{f \rho V^2}{2\text{Ø}i}$ bar/100 m

5.2. TWO PHASE FLUID

Many correlations exist to calculate the pressure drop for two phases flow, depending of the vertical or horizontal line, ratio of vapor/liquid and pressure and temperature conditions. That is out of scope of this guide and we mention only some authors :

	POETTMAN/CARPENTER	FLANIGAN
	EATON	BEGGS/BRILL
	LOCKHART/MARTINELLE	TAITEL/DUCKLER

quick methods for an estimation are as follows :

5.2.1. "ABAQUE AFTP" for gas could be used with the correction of the line diameter. Takin as defined in § 4 and the liquid viscosity as the fluid viscosity.

5.2.2. Method using MOODY or "regular" fanning friction factors.

It is the same method as on § 5.1.2. with $\rho = \rho m$ and V = Vm as defined on § 4 and the fluid viscosity taken as the liquid viscosity.

5.2.3. A more detailed method using the Lockhart Martinelli method is given in section 11.0 PIPELINES.

6. NOTES

- Tubes dimensions are standard and an example is given on Table 3.
- With "ABAQUE AFTP" the correction for the internal diameter must be done and an estimation of the line thickness could be done with the following formula used mainly for high pressure.

$$e = \frac{P\,\varnothing e}{2(SE + PY)} + c$$

e = thickness mm　　Y = coefficient having values for ferritic steels
P = Design pressure bar g　　Øe = external diameter inch
S = allowable stress bar　　C = corrosion allowance mm
E = longitudial weld joint factor

S, E and Y are not always available so the following formular could be used for an estimation.

$$e = \frac{P\,\varnothing e}{K} + c$$

e = thickness in mm　　c = corrosion allowance in mm
P = design pressure in bar g　　K = 43 for carbon steel and low temperature carbon steel
Øe = external diameter in inch　　54 for 3.5 % Ni and stainless steel

For small diameters upto about 10" use the thickness given by the schedule on Table 3.

- For Δ P do not forget to take into account the change in elevation for liquid and two phase flow.

TABLE I

LIQUID LINE TYPE	Δ P bar/km		MAXIMUM VELOCITY m/s			
	NORMAL	MAXI	Ø < 2"	3" < Ø < 6"	8" < Ø < 18"	Ø > 20"
Pumps suction :						
. Liquid at bubble point or with dissolved gas	0.6	1.0	0.6	1.0	1.4	1.8
. Non boiling liquid	2.3	3.5	0.9	1.2	1.8	2.4
Unit lines :						
. Liquid at bubble point or with disolved gas	0.6	1.0	0.6	1.0	1.4	1.8
. Non boiling liquid	2.3	3.5	0.9	1.2	1.8	2.4
Pump discharge :						
. Discharge pressure < 50 barg	3.5	4.6		Velocity 1.5 to 4.5 m/s		
. Discharge pressure > 50 barg	7.0	9.0		Velocity 1.5 to 4.5 m/s		
Water lines :						
. Cooling water and service Note 1	2.3	3.5	1.3	2.5	3.5	4
Boiler feed :						
. Pressure < 50 barg	3.5	4.6		Velocity 1.5 to 4.5 m/s		
. Pressure > 50 barg	7.0	9.0		Velocity 1.5 to 4.5 m/s		
Steam condensate return				Velocity 1 to 1.5 m/s		

Note 1 : For sea water with line in titanium the maximum velocity could increase by 20 to 30 %.

TABLE 2

VAPOR AND STEAM LINES	MAXIMUM ρV^2 ρ = gas density kg/m3 V = gas velocity m/s	MAXIMUM VELOCITY m/s	ΔP bar/km NORMAL	ΔP bar/km MAXI
VAPOR LINES . P < 20 bar g	6 000		ΔP must be considered and be compatible with the corresponding service	
. 20 < P < 50 bar g	7 500			
. 50 < P < 80 bar g	10 000			
. P > 80 bar g	15 000			
- Discontinuous operation eg: compressor anti-surge : . P < 50 bar g	10 000			
. 50 < P < 80 bar g	15 000			
. P > 80 bar g	25 000			
. compressor suction	To be compatible with above		0.25	0.7
. compressor discharge			0.5	1.2
STEAM LINES				
- P < 10 bar g . Short line L < 200 m			0.5	1.0
. Long line L > 200 m			0.15	0.25
- 10 < P < 30 barg . Short line L < 200 m		42	1.2	2.3
. Long line L > 200 m		42	0.25	1.0
- P > 30 bar g . Short line L < 200 m		30	1.2	2.3
. Long line L > 200 m		30	0.35	1.0

7. REFERENCES AND USEFUL LITERATURE

- LUDWIG
- Flow of fluids CRANE
- "Gas liquid flow in pipelines I Research results" by A.E. DUKLER May 1969 Pupl by A.G.A., API and Union of HOUSTON
- "Gas liquid flow in pipeline II - Design manual" by O. BAKER, H.W. BRAINERD, C.O. COLDREN, FLANIGAN and J.K. WELCHEN, October 1970, published by A.G.A. and API
- "Proposed correlation of data for isothermal two phase, two component flow in pipelines" LOCKHART, R.W. and R.C. MARTINELLI (1949)
- BEGGS , H.D., and BRILL, J.P. Manual for "Two phase flow in pipes" 1975 university of TULSA
- ABAQUES AFTP :
  - "Pour le calcul des pertes de charges des liquides dans les conduites"
  - "Pour le calcul des pertes de charges des gaz dans les conduites"
- PEPITE PROGRAM
- "CHEMICAL PROCESS DESIGN ON A PROGRAMMABLE CALCULATOR" W. WAYNE BLACKWEIL, B.S. Page 22
- "Two phase pressure drop computed" - Mafik Soliman, Hyd. Processing April 84

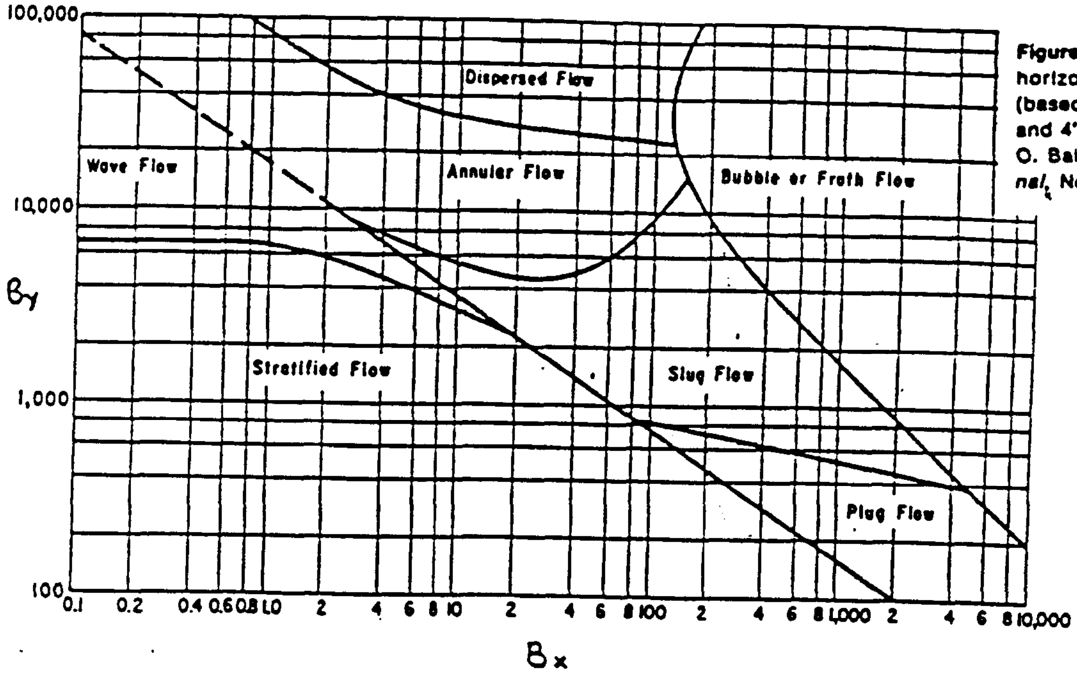


Figure 1 . Flow patterns for horizontal two-phase flow (based on data from 1", 2" and 4" pipe). O. Baker, *Oil and Gas Journal*, Nov. 10, 1958, p. 156.)

OSHINOWO – CHARLES TWO PHASE FLOW MAP FOR VERTICAL UPWARD FLOW

FIGURE 2

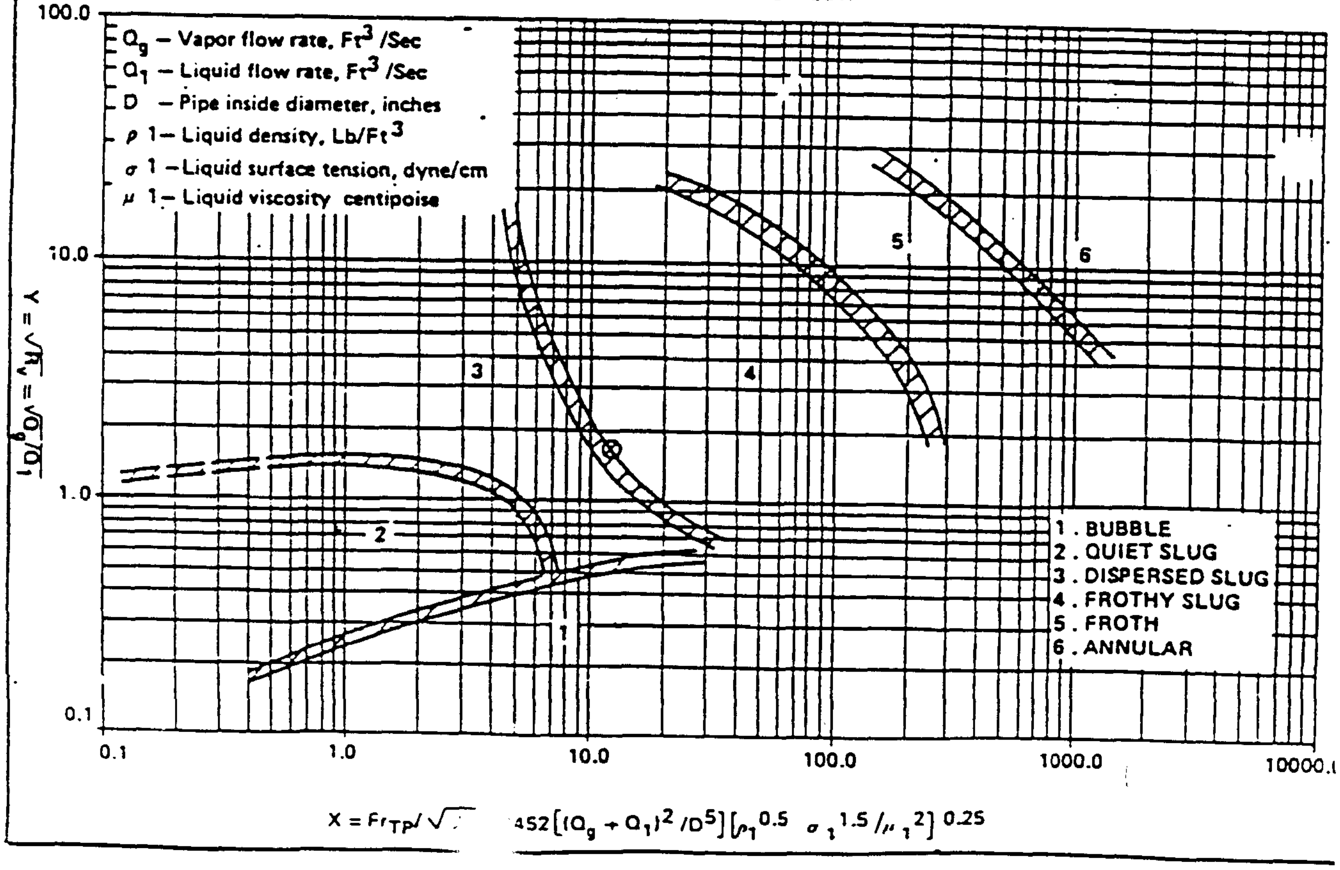
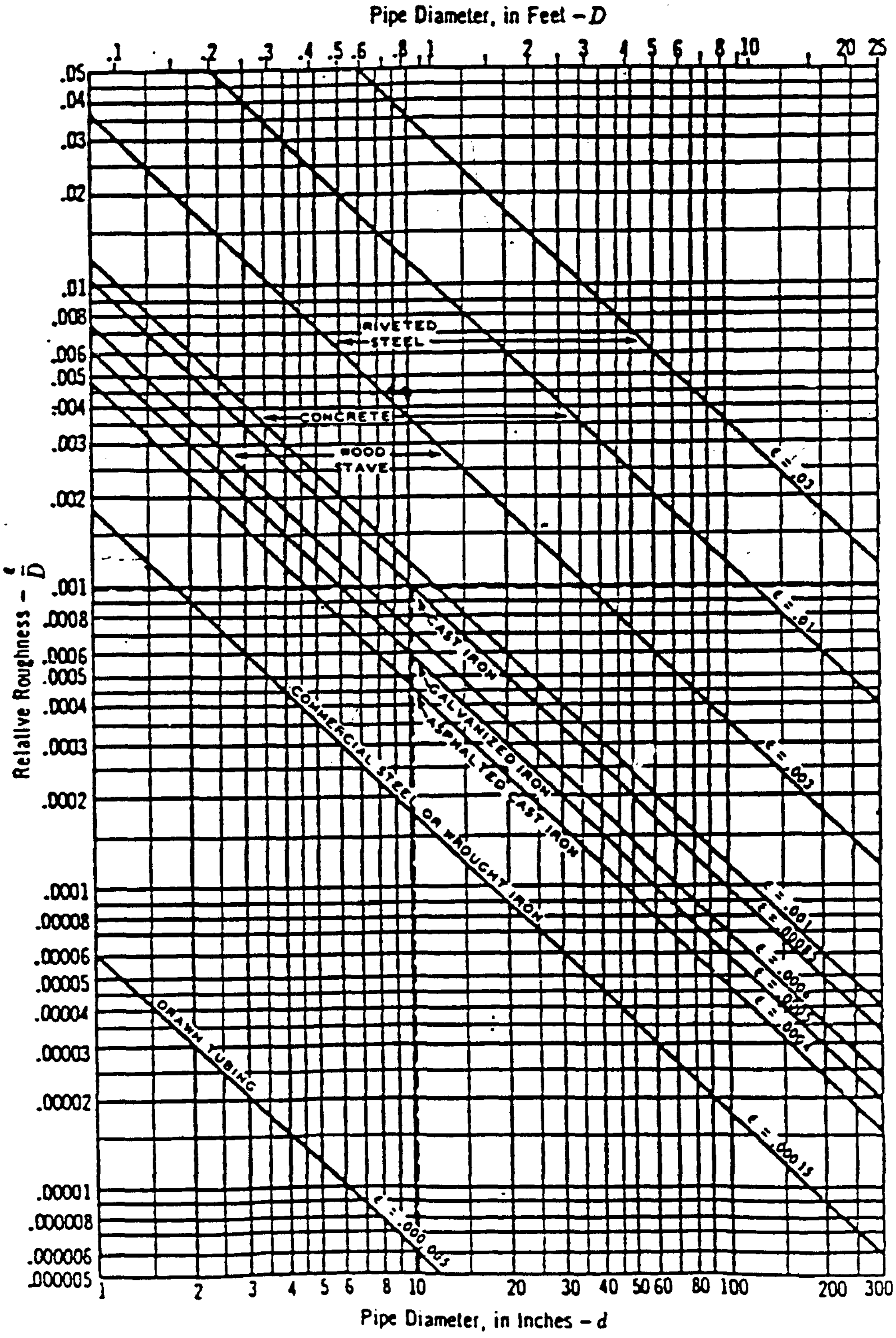


Figure 3 - Relative roughness of pipe



FRICTION FACTORS FOR CYLINDRICAL PIPE

FIG. 4

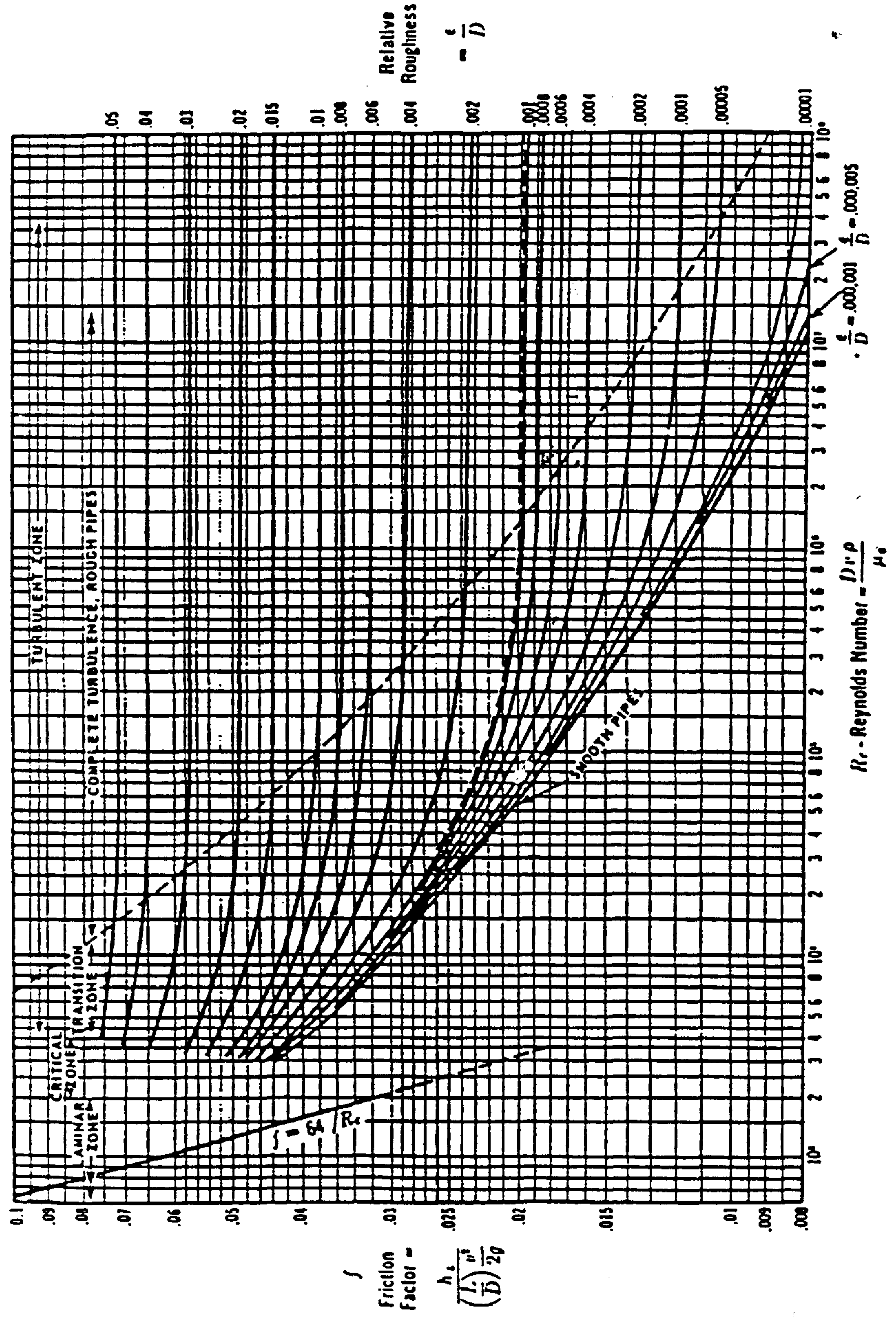


TABLE N°3

TUBE DIMENSIONS IN MM

		NORMAL THICKNESS WITH TOLERANCE -12,5 %										A.P.I. NORMALISATION		
DIAM INS	DIAM	"SCHEDULE NUMBER"										STANDARD SERIES 30 OU 40	EXTRA STRONG SERIES 60 OU 80	DOUBLE EXTRA STRONG SERIES 160 OU 200
		SCH 10	SCH 20	SCH 30	SCH 40	SCH 60	SCH 80	SCH 100	SCH 120	SCH 140	SCH 160			
1/8"	10,29	1,24			1,73		2,41					1,73	2,41	
1/4"	13,72	1,65			2,23		3,02					2,24	3,07	
3/8"	17,16	1,65			2,31		3,20					2,31	3,20	
1/2"	21,34	2,11			2,77		3,73				4,75	2,77	3,33	7,47
3/4"	26,67	3,21			2,87		3,91				5,56	2,87	3,91	7,82
1"	33,38	2,77			3,38		4,55				6,35	3,38	4,55	9,09
1 1/4"	42,16	2,77			3,56		4,85				6,35	3,56	4,83	9,70
1 1/2"	46,26	2,77			3,68		5,08				7,54	3,68	5,08	10,10
2"	50,32	2,77			3,91		5,54				8,71	3,97	5,54	11,07
2 1/2"	73,02	3,06			5,16		7,01				9,52	5,16	7,01	14,02
3"	88,90	3,06			5,40		7,62				11,10	5,49	7,02	15,24
3 1/2"	101,60	3,06			5,74		8,08					5,74	8,09	18,15
4"	114,30	3,06			6,02		8,56		11,10		13,40	6,02	8,50	17,12
5"	141,30	3,40			6,55		9,52		12,70		15,87	9,55	9,52	19,05
6"	168,27	3,40			7,11		10,97		14,27		18,24	7,11	10,97	21,95
8"	219,07	3,78	6,35	7,04	8,18	10,31	12,70	15,08	18,24	20,62	23,01	7,04 8,18	12,70	22,22
10"	273,04	4,18	6,35	7,80	9,27	12,70	15,08	18,24	21,41	25,40	28,57	7,07 7,80 9,27	12,70	
12"	323,24	4,57	6,35	8,38	10,31	14,27	17,45	21,41	25,40	28,57	33,32	8,38 9,52	12,70	
14"OD	355,59	6,35	7,93	9,52	11,10	15,06	19,05	23,80	26,97	31,75	35,71	9,52	12,70	
16"OD	406,39	6,35	7,93	9,52	12,70	16,66	21,41	26,19	30,94	36,50	37,67	9,52	12,70	
18"OD	457,10	6,35	7,93	11,10	14,27	18,24	23,00	27,38	34,11	39,07	44,45	9,52	12,70	
20"OD	507,99	6,35	9,52	12,70	15,09	20,52	26,19	31,75	38,10	44,45	49,20	9,52	12,70	
24"OD	609,59	6,35	9,52	14,27	17,45	23,80	30,94	38,10	44,45	52,37	58,72	9,52	12,70	
30"OD	761,99	7,93	12,70	15,37										

Notes :
1) For schedule 10 Ø < 14" are not used normally
2) The following diameters are not common : 1/8", 3/8", 1 1/4", 2 1/2", 3 1/2", 5"
3) For OD > 30" line diameters increase in 2" increments

1. APPLICABILITY

The purpose of this chapter is to determine the piping class used as shown on a PID line when the piping material class document does not exist. This is generally the case for feasibility and pre-project studies.

2. CLASS NUMBERING PRINCIPLES (From DD-SP-TCS-112 "PIPING MATERIALS CLASSES")

2.1. GENERATION OF NUMBER

The class number shall consist of a capital letter representing the ANSI series and a two-digit number representing the main material entering into the composition of material used for the valve bodies, tubes, fittings and flanges of the network in question.

Example :

B 01

Series 150........... ....Carbon steel

The tables below give the letters and numbers to be used for numbering piping classes.

2.2. LETTERS representing the series of the class

Series	125	150	250	300	400	600	900	1500	2500	Tracing
Symbol	A	B	C	D	E	F	G	H	J	Y

2.3. NUMBERS representing the main material of the class

01 to 20	:	Carbon steels (ordinary, galvanized, normalized, etc...)
21 to 45	:	Alloy steels
46 to 70	:	Stainless steels
71 to 85	:	Special alloys (Monel, Hastelloy, etc...)
86 to 99	:	Other materials (Cast-iron, copper, copper alloy, etc...) Glass Plastic, cement-asbestos fiber, etc...

3. PRESSURE TEMPERATURE RATINGS

The following ANNEX G is extracted from ANSI B 16-5 1977 (AMERICAN NATIONAL STANDARD STEEL PIPE FLANGES AND FLANGED FITTINGS).

For pressure temperature ratings higher than series 2500 the following is used 5000 PSI, 10000 PSI, ... (used mainly for well tubing and wellhead).

FIG. G

Pressure-temperature ratings for steel flanges and flanged fittings from ANSI B16.5-1977

Temp. °C	Temp. °F	CLASS 150	300	400	600	900	1500	2500	See Notes
					Norm. Carbon steel	A105, A181-II, A216-WCB, A515-70 A516-70 *350-LF2, A537-C1.1			(a) (h) (a) (g) (d)
					Pressures are in pounds per square inch, gage (psig)				
-28.9 to 37.8	-20 to 100	285	740	990	1480	2220	3705	6170	
93.3	200	260	675	900	1350	2025	3375	5625	
141.5	300	230	655	875	1315	1970	3280	5470	
204	400	200	635	845	1270	1900	3170	5280	
260	500	170	600	800	1200	1795	2995	4990	
316	600	140	550	730	1095	1640	2735	4560	
343	650	125	535	715	1075	1610	2685	4475	
371	700	110	535	710	1065	1600	2665	4440	
	750	95	505	670	1010	1510	2520	4200	
427	800	80	410	550	825	1235	2060	3430	
	850	65	270	355	535	805	1340	2230	
482	900	50	170	230	345	515	860	1430	
	950	35	105	140	205	310	515	860	
538	1000	20	50	70	105	155	260	430	
					Type 304	A182-F304, A182-F304H A240-304, A351-CF8 A351-CF3	not to be used over 800°F		
-28.9 to 37.8	-20 to 100	275	720	960	1440	2160	3600	6000	
93.3	200	235	600	800	1200	1800	3000	5000	
141.5	300	205	530	705	1055	1585	2640	4400	
204	400	180	470	630	940	1410	2350	3920	
260	500	170	435	585	875	1310	2185	3640	
316	600	140	415	555	830	1245	2075	3460	
343	650	125	410	545	815	1225	2040	3400	
371	700	110	405	540	805	1210	2015	3360	
	750	95	400	530	795	1195	1990	3320	
427	800	80	395	525	790	1180	1970	3280	
	850	65	390	520	780	1165	1945	3240	
482	900	50	385	510	770	1150	1920	3200	
	950	35	375	500	750	1125	1870	3120	
538	1000	20	325	430	645	965	1610	2685	
	1050		310	410	620	925	1545	2570	
593	1100		260	345	515	770	1285	2145	
	1150		195	260	390	585	980	1630	
649	1200		155	205	310	465	770	1285	
	1250		110	145	220	330	550	915	
704	1300		85	110	165	245	410	685	
	1350		60	85	125	185	310	515	
760	1400		50	65	90	145	240	400	
	1450		35	45	70	105	170	285	
851	1500		25	30	50	70	120	200	

NOTES:

(a) permissible but not recommended for prolonged use above 800°F

(d) not to be used over 650°F

(g) not to be used over 850°F

(h) not to be used over 1 000°F

Conversion: kPa (ga) = 6.895 × psig

SI

1. APPLICABILITY

The purpose of this chapter is to determine the types of valves used for designation on the PID.

- valves are used for two mains functions, isolation and control.

The following is only a guide line for selection of types of valves which must follow the piping material class document when it exists.

2. BLOCK VALVES

The main types are :

- ball
- gate
- plug
- butterfly

2.1. BALL VALVE

Ball valves can be full bore or reduced bore.

2.1.1. Full bore uses

- flare system : upstream and downstream of PSV, rupture disc, flare line if required.
- downstream pig launcher and upstream pig receiver.
- vents and drains on hydrocarbon equipments.
- piping valves on instruments for hydrocarbon.
- for block valves an hydrocarbon lines if the pressure drop is critical.
- utility except water for diameter larger than 2".

2.1.2. Reduced bore uses

- Block on hydrocarbon service without solid particles.

2.2. PLUG VALVE USES

Plug valves have the same use as reduced bore ball valves when used for high pressure (600 #). Plug valves can be assimilated to reduced ball valves, generally, plug valves are smaller and lighter of the two.

2.3. GATE VALVE USES

- Gate valves can be used as ball valves except for downstream of pig launcher and upstream of pig receivers. The vertical physical space required by a gate valve is greater than a ball valve.
- Tight shut off for ball or plug valves is superior to that of a gate.
- For hydrocarbon service with solid particles presentor as wing valves on well heads.
- For quick closure purposes.
- On utility lines for low diameters < 2"

2.4. BUTTERFLY VALVE USES

- On water lines for service, utility or sea water, generally for diameters larger than 2".

3. CONTROL VALVES

The main types are :
- globe
- butterfly
- special

3.1. GLOBE VALVE USED

- Control valve used in most of cases except at very high P as defined by instrument group, or on water networks, and compressor suction lines for throttling purposes.

3.2. BUTTERFLY VALVE USED

- On water networks
- Throttling at compressor suction

3.3. SPECIAL VALVES USED

Special valves are defined by instrument group :

- for very high ΔP the angle valve could be used
- for compressor anti-surge cage valves could be used.

1. APPLICABILITY

The purpose of this chapter is to calculate precisely the pressure drop in a piping network. This may be required for either study phase for situations where Δ P is a critica consideration. For most projects however calculation of process line Δ Ps wil not be required.

The pressure drop calculations are based on a summation K method.

2. Δ P THROUGH VALVES

2.1. VALVES OPEN

TYPE	GATE VALVE	GLOBE VALVE	PLUG COCK NALINE	CHECK VALVE
K	0.15	5	0.1	2.4

$$\Delta P = K \frac{\rho V^2}{1.962 \times 10^5}$$

ΔP in kg/cm^2 1 bar = 1.02 Kg/cm2

ρ : fluid density in kg/m^3 V : fluid velocity in m/s

2.2. BALL VALVE

Δ P through ball valve with reduced bore : This Δ P depends on the valve vendor. An extract of CAMERON BALL VALVE PRODUCTS is given as an example.

3. Δ P THROUGH FITTINGS

The example below gives some values of the pressure drop coefficient K for fittings encountered in cylindrical conduits. For further information, refer to "MEMENTO DES PERTES DE CHARGE by I.E. IDEL'CIK, EYROLLES édition, PARIS".

ΔP in kg/cm^2

ρ : fluid density in kg/m^3

V : fluid velocity in m/s

$$\Delta P = K \frac{\rho V^2}{1.962 \times 10^5}$$

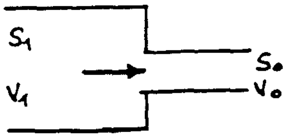
3.1. ELBOWS

K values for elbows.

R/D	1.5	3	5
90 °	0.17 + 2.36 f	0.12 + 4.72 f	0.09 + 7.87 f
45 °	0.11 + 1.18 f	0.08 + 2.36 f	0.06 + 3.94 f

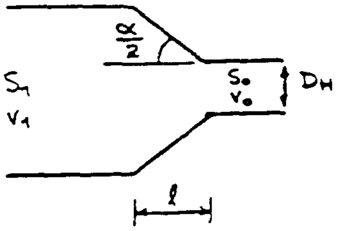
f = friction factor see chapter PROCESS/UTILITY -LINE SIZING § 5

3.2 INLET OF A CONDUIT



$$K = 0.5\left(1-\frac{S_o}{S_1}\right)$$

S = area

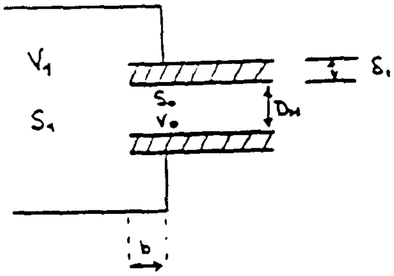


$$K = K'\left(1-\frac{S_o}{S_1}\right) + \frac{f}{8\sin\alpha/2}\left[1-\left(\frac{S_o}{S_1}\right)^2\right]$$

f = Friction factor see § 3.1

VALUES OF K'

l/D_H	$\alpha°$								
	0	10	20	30	40	60	100	140	180
0,025	0,50	0,47	0,45	0,43	0,41	0,40	0,42	0,45	0,50
0,050	0,50	0,45	0,41	0,36	0,33	0,30	0,35	0,42	0,50
0,075	0,50	0,42	0,35	0,30	0,26	0,23	0,30	0,40	0,50
0,10	0,50	0,39	0,32	0,25	0,22	0,18	0,27	0,38	0,50
0,15	0,50	0,37	0,27	0,20	0,16	0,15	0,25	0,37	0,50
0,60	0,50	0,27	0,18	0,13	0,11	0,12	0,23	0,36	0,50

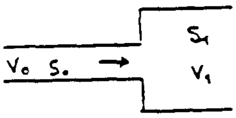


$$K = K'\left(1-\frac{S_o}{S_1}\right)$$

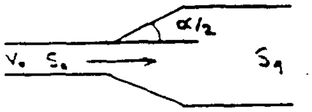
K'

δ_1/D_h	b/D_h										
	0	0,002	0,005	0,010	0,020	0,050	0,100	0,200	0,300	0,500	∞
0	0,50	0,57	0,63	0,68	0,73	0,80	0,86	0,92	0,97	1,00	1,00
0,004	0,50	0,54	0,58	0,63	0,67	0,74	0,80	0,84	0,90	0,94	0,94
0,008	0,50	0,53	0,55	0,58	0,62	0,68	0,74	0,81	0,85	0,88	0,88
0,012	0,50	0,52	0,53	0,55	0,58	0,63	0,68	0,73	0,79	0,83	0,83
0,016	0,50	0,51	0,51	0,53	0,55	0,58	0,64	0,70	0,74	0,77	0,77
0,020	0,50	0,51	0,51	0,52	0,53	0,55	0,60	0,66	0,69	0,72	0,72
0,024	0,50	0,50	0,50	0,51	0,52	0,53	0,58	0,62	0,65	0,68	0,68
0,030	0,50	0,50	0,50	0,51	0,52	0,52	0,54	0,57	0,59	0,61	0,61
0,040	0,50	0,50	0,50	0,51	0,51	0,51	0,51	0,52	0,52	0,54	0,54
0,050	0,50	0,50	0,50	0,50	0,50	0,50	0,50	0,50	0,50	0,50	0,50
∞	0,50	0,50	0,50	0,50	0,50	0,50	0,50	0,50	0,50	0,50	0,50

3.3 OUTLET OF A CONDUIT



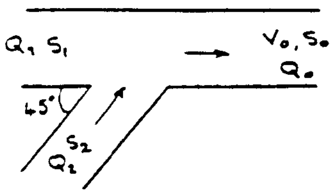
$$K = \left(1-\frac{S_o}{S_1}\right)^2$$



$$K = 3.2\tan\frac{\alpha}{2}\sqrt[4]{\tan\frac{\alpha}{2}}\left(1-\frac{S_o}{S_1}\right)^2 + \frac{f}{8\sin\alpha/2}\left[1-\left(\frac{S_o}{S_1}\right)^2\right]$$

3.4 TEE

3.4.1



$S_1 + S_2 > S_0$ α = 45°

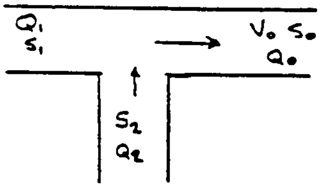
$S_0 = S_1$

Q = Volumetric flow

a) Lateral branch $$K = 1 + \left(\frac{Q_2}{Q_0} \times \frac{S_0}{S_2}\right)^2 - 2\left(1 - \frac{Q_2}{Q_0}\right) - 1.41\,\frac{S_0}{S_2}\left(\frac{Q_2}{Q_0}\right)^2$$

b) Straight branch $$K = 1 - \left(1 - \frac{Q_2}{Q_0}\right)^2 - 1.41\,\frac{S_0}{S_2}\left(\frac{Q_2}{Q_0}\right)^2$$

3.4.2



$S_1 + S_2 > S_0$

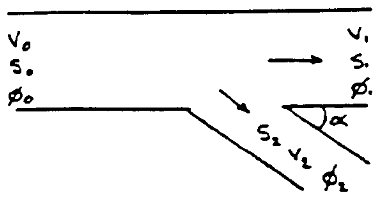
$S_0 = S_1$

a) Lateral branch $$K = A\left[1 + \left(\frac{Q_2}{Q_0} \times \frac{S_0}{S_2}\right)^2 - 2\left(1 - \frac{Q_2}{Q_0}\right)\right]$$

S_2/S_0	0 - 0.2	0.3 - 0.4	0.6	0.8	1.0
A	1.0	0.75	0.7	0.65	0.6

b) Straight Branch $$K = 1.55 \times \frac{Q_2}{Q_0} - \left(\frac{Q_2}{Q_0}\right)^2$$

3.4.3



$S_2 + S_1 > S_0$

$S_0 = S_1$

$0 < \alpha < 60°$

$\alpha = 90°$

a) Lateral Branch

1) $0 < \alpha < 60°$ and $\alpha = 90°$ for $\frac{Ø_2}{Ø_0} \leqslant 2/3$

$$\underline{K = A' \left[1 + \left(\frac{V_2}{V_0}\right)^2 - 2 \times \frac{V_2}{V_0} . \cos \alpha \right]}$$

2) $\alpha = 90°$ $\frac{Ø_2}{Ø_0} = 1$

$$\underline{K = A' \left[0.34 + \left(\frac{V_2}{V_0}\right)^2 \right]}$$

$A' = 1$ for $\frac{V_2}{V_0} \leqslant 0.8$

$A' = 0.9$ for $\frac{V_2}{V_0} > 0.8$

b) Straight branch

$$\underline{K = 0.4 \left(1 - \frac{V_1}{V_0}\right)^2}$$

Resistance Coefficient, K

The resistance coefficient is calculated by the formula:

$$K = f\frac{L}{D}$$

Valves of the friction factor, f, for various pipe sizes are listed in table 1-17.

Values for L/D and C for fully opened valves were calculated from theoretical considerations. Valves of C for partially open valves were extrapolated from test results for representative sizes of ball valves.

Chart 1 · 18 provides graphic representation of valve position versus the percent of full open area.

Table 1-17
Friction Factor (f)

Nominal Pipe Size	Friction Factor (f)	Nominal Pipe Size	Friction Factor (f)
2"	0190	24"	0115
3"	0175	26"	0113
4"	0165	28"	0112
6"	.0150	30"	.0110
8"	.0140	34"	0107
10"	0135	36"	0105
12"	.0130	40"	0104
14"	0125	42"	0103
16"	0123	48"	0102
18"	0120	—	—
20"	0117	—	—
22"	0116	—	—

Table 1-3
Calculated Values of L/D for Full Opening Cameron Ball Valves in Full Open Position

Valve Size	150 lb. ANSI		300 lb. ANSI		400 lb. ANSI		600 lb. ANSI	
	Flange	Welded	Flange	Welded	Flange	Welded	Flange	Welded
2	3.40	4.00	4.12	4.12	–	–	5.60	5.60
3	2.56	3.56	3.56	3.56	–	–	4.48	4.48
4	2.22	2.95	2.95	2.95	3.94	3.94	4.18	4.18
6	2.58	2.65	2.65	2.65	3.25	3.25	3.66	3.66
8	2.25	2.56	2.06	2.56	2.94	2.56	3.25	2.56
10	2.10	2.20	2.24	2.20	2.65	2.33	3.10	2.33
12	2.00	2.08	2.12	2.08	2.50	2.16	2.75	2.16
14	2.03	1.97	2.26	1.97	2.45	2.10	2.64	2.09
16	1.96	1.77	2.16	1.77	2.33	1.95	2.56	1.95
18	1.97	1.74	2.08	1.74	2.24	1.96	2.50	1.96
20	1.87	1.66	2.02	1.66	2.16	1.97	2.44	1.97
22	1.88	1.62	2.02	1.62	2.12	1.90	2.40	1.91
24	1.81	1.60	1.93	1.60	2.08	1.89	2.36	1.89
26	1.80	1.78	1.96	1.78	2.06	1.78	2.28	1.78
28	1.81	1.74	1.96	1.76	2.04	1.74	2.26	1.74
30	1.76	1.69	1.90	1.62	2.07	1.69	2.24	1.69
32	1.76	1.69	1.95	1.69	2.11	1.69	2.28	1.69
34	1.77	1.66	1.95	1.66	2.14	1.66	2.32	1.66
36	1.74	1.64	1.97	1.64	2.14	1.66	2.38	1.64
40	1.77	1.69	1.95	1.69	2.13	1.69	2.39	1.69
42	1.89	1.65	1.89	1.65	2.09	1.65	2.34	1.65
48	1.76	1.81	1.94	1.81	2.11	1.81	2.15	1.81
54	1.77	1.78	1.85	1.78	2.00	1.78	2.07	1.78
60	1.74	1.85	1.88	1.85	2.05	1.85	2.12	1.85
64	–	1.82	–	1.82	–	1.82	–	1.82

Valve Size	900 lb. ANSI		1500 lb. ANSI		2500 lb. ANSI	
	Flange	Welded	Flange	Welded	Flange	Welded
2	7.00	7.03	7.00	7.00	8.61	8.61
3	4.80	4.80	5.97	5.97	7.78	7.78
4	4.42	4.42	5.29	5.29	6.52	6.52
6	4.00	4.00	4.62	4.62	6.00	6.00
8	3.62	2.69	4.10	2.69	5.04	3.50
10	3.30	2.65	3.90	2.65	5.00	3.30
12	3.16	2.54	3.71	2.54	4.66	3.05
14	3.06	2.44	3.74	2.44		
16	2.97	2.44	3.58	2.44		

Table 1·5
Calculated Values of L/D for Reduced Opening Cameron Ball Valves in Full Open Position

Valve Size	150 lb ANSI		300 lb ANSI		400 lb ANSI		600 lb ANSI	
	Flange	Welded	Flange	Welded	Flange	Welded	Flange	Welded
3x2	130	98.6	117	98.6	—	—	179	206
4x3	45.1	35.5	46.0	35.5	38.4	39.2	50.1	40.1
6x4	110	119	121	119	175	89.6	127	92.4
8x6	62.5	61.8	61.8	61.8	48.6	47.2	49.2	47.2
10x8	37.8	37.0	37.6	37.0	38.1	29.8	34.9	29.8
12x10	26.3	25.8	25.6	25.9	26.6	21.2	24.4	21.3
14x12	11.5	11.5	11.2	11.5	11.0	8.54	10.2	8.56
16x14	17.8	18.1	17.1	18.1	18.1	14.9	18.3	15.0
18x16	15.0	15.4	14.9	15.4	14.4	13.1	13.1	13.1
20x18	13.6	13.5	13.6	13.5	12.7	11.9	11.8	11.9
22x20	12.0	12.2	12.2	12.2	11.6	11.0	10.9	10.9
24x22	10.7	10.7	11.0	10.7	9.65	9.90	8.44	9.91
26x24	9.45	9.43	9.61	9.43	9.71	9.43	9.92	9.43
28x26	9.95	9.87	10.10	9.87	10.17	9.87	10.39	9.87
30x28	9.41	9.35	9.58	9.35	9.72	9.35	9.89	9.35
32x30	8.45	8.38	8.64	8.38	8.81	8.38	8.97	8.38
34x32	8.87	8.77	9.06	8.77	9.25	8.77	8.93	8.77
36x34	8.08	7.98	8.37	7.98	8.49	7.98	8.72	7.98
40x36	14.93	14.86	15.12	14.86	15.30	14.86	15.56	14.86
42x40	7.75	7.51	7.75	7.51	7.95	7.51	8.70	7.51
48x42	22.47	22.52	22.65	22.52	22.81	22.52	22.86	22.52
56x48	30.55	30.60	30.67	30.60	30.82	30.60	30.89	30.60
60x56	9.81	9.88	9.92	9.88	10.09	9.88	10.16	9.88
64x60	—	10.95	—	10.95	—	10.95	—	10.95

Valve Size	900 lb ANSI		1500 lb. ANSI		2500 lb. ANSI	
	Flange	Welded*	Flange	Welded	Flange	Welded
3x2	127	108	150	118	89	130
4x3	41.9	41.1	53.2	44.7	41.9	49.2
6x4	123	94.5	132	96	112	107
8x6	56.8	47.6	47.6	47.6	—	51.7
10x8	30.0	30.4	30.6	30.4	—	—
12x10	21.8	27.9	16.1	27.9	—	—
14x12	13.4	12.1	12.5	—	—	—
16x14	18.5	—	18.7	—	—	—

Table 1·6
Calculated Values of L/D for Venturi Opening Cameron Ball Valves in Full Open Position

Valve Size	150 lb ANSI		300 lb. ANSI		400 lb. ANSI		600 lb. ANSI	
	Flange	Welded	Flange	Welded	Flange	Welded	Flange	Welded
10x6	282	—	—	—	—	—	—	280
12x8	152	153	141	153	153	147	147	147
14x10	61.9	61.9	67.9	61.9	63.5	56.7	56.8	56.8
16x12	45.0	45.0	45.0	45.0	—	44.0	44.1	43.7
18x14	58.7	58.7	58.5	58.5	59.0	56.7	57.9	56.7
20x16	45.2	45.2	45.7	45.2	45.2	44.3	44.7	40.8
22x18	38.0	38.0	38.0	38.0	36.9	34.6	34.6	34.6
24x20	31.4	31.4	31.4	31.4	30.5	28.8	28.9	28.9
26x22	27.0	27.0	26.4	27.0	26.4	24.8	25.1	24.9
30x24	46.3	46.7	46.3	46.5	44.8	44.8	41.9	44.9
36x30	31.6	31.6	30.6	31.6	29.1	25.3	25.5	25.3

Valve Size	900 lb ANSI		1500 lb. ANSI		2500 lb ANSI	
	Flange	Welded	Flange	Welded	Flange	Welded
10x6	—	—	—	—	—	—
12x8	149	135	138	135	—	—
14x10	—	64.1	—	—	—	—
16x12	46.8	46.1	44.4	44.1	—	—

1. APPLICABILITY

The purpose of this chapter is to give some formulae to estimate the size and the number o control valves installed for one given service, and to estimate the capability of the contro valves in case of revamping. The final sizing should be done by instrument people.

2. CONTROL VALVES CHARACTERISTICS

These are determined principally by the design of the valve trim. The three fundamental characteristics available are quick opening, linear, and equal percentage.

2.1. QUICK OPENING

As the name implies, this type provides a large opening as the plug is first lifted from the seat with lesser flow increase as the stem opens further. The most common application if for simple on-off control with no throttling of flow required.

2.2. LINEAR

Linear trim provides equal increases in stem travel. Thus the flow rate is linear with plug position throughout its travel.

2.3. EQUAL PERCENTAGE

Provides equal percentage increases in rate of flow for equal increments of stem travel. The characteristics provide a very small opening for plug travel near the seat and very large increase toward the fully open position. As a result, a wide rangeability of flow rate is achieved.

3. CONTROL VALVE RANGEABILITY

- For an estimation only it is common practice to select a valve in which the valve opening at maximum flow is smaller than or equal to 95 per cent.
- For normal flow the valve opening should be at least 60 per cent while for minimum flow, if applicable, the opening should be larger than 10 per cent. If the minimum fl is close to or smaller than 10 per cent, a smaller valve should be installed in parallel with the main valve.
- For a flow rate the valve opening depends on the valve characteristics and it is given by vendor in their catalogue.

4. FORMULAE

The valve area is characterized by the coefficient Cv (except for FISHER which use Cg for the gas (see hereafter).

The Cv coefficient is the number of U.S. gallons of water flowing during one minute through a restriction and the pressure drop through this restriction equal 1 PSI.

The following formulae are simplified and to be used only for an estimation of the Cv. Some corrections may be necessary for the installation of reducers around the control valve. If so, the formulae given by manufacturers in their catalogues will be used for a better Cv calculation.

4.1. LIQUID

A - Sub critical flow	B - Critical flow
$P_v < P_2$ and $P_1 - P_2 < C_f^2 \Delta P_s$	$P_1 - P_2 > C_f^2 \Delta P_s$
$C_v = 1.16\,Q \sqrt{\dfrac{sg}{P_1 - P_2}}$	$C_v = \dfrac{1.16\,Q}{C_f} \sqrt{\dfrac{sg}{\Delta P_s}}$

Cf = critical flow coefficient (given by manufacturers and depends on the type of valve and the action of valve by increase of variable) cf < 1

Pv = fluid vapor pressure in bar

P1 = upstream pressure in bar

P2 = downstream pressure in bar

$$\Delta P_s = P_1 - \left(0.96 - 0.28 \sqrt{\frac{P_1}{P_c}}\right) P_v$$

or to simplify, if $P_v < 0.5\,P_1$, $\Delta P_s = P_1 - P_v$

Pc = fluid critical pressure in bar

Q = flow rate in m^3/hr at upstream conditions

sg = specific gravity at flowing temp. (water = 1 at 15°C)

4.2. GAS AND STEAM

A - Sub critical flow	B - Critical flow
$P_1 - P_2 < 0.5\,C_f^2\,P_1$	$P_1 - P_2 \geqslant 0.5\,C_f^2\,P_1$

GAS

A - Sub critical flow	B - Critical flow
$C_v = \dfrac{Q}{295} \sqrt{\dfrac{G.T.Z}{(P_1 - P_2)(P_1 + P_2)}}$	$C_v = \dfrac{Q \sqrt{G.T.Z}}{257\,C_f\,P_1}$

SATURATED STEAM

$$Cv = \frac{72.4\ W}{\sqrt{(P1 - P2)\ (P1 + P2)}}$$

$$Cv = \frac{83.7\ W}{Cf\ P1}$$

SUPER HEATED STEAM

$$Cv = \frac{72.4\ (1 + 0.00126\ Tos)\ W}{\sqrt{(P1 - P2)\ (P1 + P2)}}$$

$$Cv = \frac{83.7\ (1 + 0.00126\ Tos)\ W}{Cf\ P1}$$

Cg, P1, P2, Q same definition and unit as § 4.1.

G = relative density (air = 1.0)
T = upstream gas temperature OK = 273 + °C
Z = upstream compressibility factor
W = steam weight in t/hr
Tos = steam superheat in °C

4.3. TWO PHASE FLOW

For sizing, maximum $\Delta P = P1 - P2 = 0.5\ Cf^2\ P1$

A - Without liquid vaporization	B - With liquid vaporization
$Cv = \frac{51.8\ W}{\sqrt{P\ (d1 + d2)}}$	$Cv = \frac{36.6\ W}{\sqrt{\Delta P\ d1}}$

Cg, P1, P2 same definition and unit as § 4.1.

W = total fluid flow in t/hr
d1 = upstream mixture density in kg/m^3

$$d = \frac{W \times 10^3}{\frac{Wll}{dll} + \frac{Wlv}{dlv}}$$

Wll = upstream liquid flow in kg/hr
dll = upstream liquid density in kg/m^3
Wlv = upstream vapor flow in kg/hr
dlv = upstream vapor density in kg/m^3

d2 = downstream mixture density in kg/m^3

$$d2 = \frac{W \times 10^3}{\frac{W2l}{d2l} + \frac{W2v}{d2v}}$$

W2l = downstream liquid flow in kg/hr
d2l = downstream liquid density in kg/m^3
W2v = downstream vapor flow in kg/hr
d2v = downstream vapor density in kg/m^3

4.4. FISHER FORMULAE

For gas "FISHER" use Cg instead of Cv

$$Cv = \frac{Cg}{C1}$$

C1 = valve coefficient (given by catalogue)

$$Cg = \frac{W}{0.4583 \quad d\ P1 \sin \left[\frac{3417}{C1}\sqrt{\frac{P1 - P2}{P1}}\right]} \quad \text{deg.}$$

W = gas flow rate in kg/hr
d = gas density at upstream conditions in kg/m^3
P1 = upstream pressure in bar also
P2 = downstream pressure in bar also

5.0. REFERENCES AND USEFUL LITERATURE

- Vendors documentations
- GPSA chapter 2

11. PIPELINES

1. APPLICABILITY

For both feasibility and preproject studies, long pipeline ΔP and ΔT calculations will normally be performed using PETITE or RESEAU. It may be necessary, however to make an estimate by hand. Details are given below on how to proceed on this.

2. PIPELINE PRESSURE DROP FORMULAE

2.1. GAS TRANSMISSION

There exist many methods of calculating ΔP for gas transmission lines. Some of these are :

American Gas Association Formula	Weymouth
Panhandle 'A' and 'B'	Darcy
Colebrook	

Below is given the Panhandle 'A' for use :

$$P_1^2 - P_2^2 = \left[\frac{q \times G^{0.4604}}{\left(\frac{Ts}{Ps}\right)^{1.07881} \times d^{2.6182} \times E}\right]^{1.8539} \times T \times L_m \times Z \times 0.3015$$

Where	P_1	=	Upstream pressure	bar (a)	
	P_2	=	Downstream pressure	bar (a)	
	G	=	Specific gravity of gas		
	T_s	=	Base temperature	K	(273 K or 298 K)
	P_s	=	Base pressure	bara (1.01325 bar)	
	T	=	Gas flowing temp	K	
	Lm	=	Pipeline length	km	
	q	=	Flowrate at Ts, Ps base	m^3/d	(at Ts, Ps)
	d	=	PIPELINE DIAMETER	cm	
	Z	=	Average gas compressibility		
	E	=	Efficiency (0.92 for a clean line)		

The formula does not take into account the pipeline profile which, if significant, can be added to the Δ P calculated if required.

2.2. LIQUID FLOW IN PIPELINES

Use Darcy equation :

$$P = 6.254 \frac{F \quad M^2}{\rho \quad D^5} \text{ bar/km}$$

M	=	Mass flow	kg/h	F	=	Moody friction factor	
ρ	=	Density	kg/m^3	E	=	Absolute roughness	cm
D	=	line id	cm			(see page 10.8 and 10.9)	
ΔP	=	pressure drop	bar/km		=	viscosity	cp

$$Re = \frac{35.368 \times M}{x\ D} \qquad F = 64/Re \quad \text{for } Re < 2000$$

$$F = \left[(8/Re)^{12} + 1/(A+B)^{3/2}\right]^{1/12} \quad \text{for } Re > 2000$$

$$\text{with : } A = \left[2.457\ Ln \frac{1}{(7/Re)^{.9} + (0.27\ E/D)}\right]^{16}$$

$$B = (37530/Re)^{16}$$

* BE CAREFUL when using friction factor charts as confusion arises between MOODY F and FANNING F' : F' = 1/4 F

2.3. TWO PHASE HORIZONTAL

Estimating 2-phase flow ΔP by hand for long pipelines is not recommended, as the flow characteristics and equilibrium will alter along its length. However an estimate of ΔP can be hand calculated providing the phase regime is fairly stable.

Given below is a calculation method based on LOCKHEART-MARTINELLI-BAKER method. This method can be used for both longpipelines (stable regime) or process lines.

METHOD

$$\Delta P_{2\ PHASE} = \Delta P_{HORIZ} + \Delta P_{VERT}$$

1. Evaluate flow regime and adjust Pipeline Ø if required
2. Calculate ΔP_G
3. Calculate ΔP_L
4. Calculate $(\Delta P_L / \Delta P_G)^{1/2}$
5. Calculate $\Delta P_{2\ PHASE}$ factor.... P: H
6. Calculate ΔP_{vert} factor (vertical section of pipe)

For convenience pipe ids are in cm

viscosity is in cp.

OPERATING DATA

GAS				LIQUID			
FLOWRATE	Wg	kg/h	= 100 000	FLOWRATE	Wl	kg/h	= 50 000
DENSITY	Dg	kg/m^3	= 18	DENSITY	Dl	kg/m^3	= 750
VISCOSITY	Vg	cp	= 0.0124	VISCOSITY	Vl	cp	= 2.4
				SURF TEN	St	dynes/cm	= 5.0
FLOWING TEMP		°C	= 34				
PIPELINE DIAMETER	D	cm	= 40.64	PIPELINE LENGTH	L	m	= 1000
INTERNAL AREA	A	m^2	= 0.13	Vertical change	Δ h	m	= + 200

STEP		NOTES
1. DETERMINE FLOW REGIME $Bx = 210.3 \dfrac{Wl}{Wg} \times \dfrac{\sqrt{Dl \times Dg}}{Dl^{2/3}} \times \dfrac{Vl^{1/3}}{St}$ $BY = \dfrac{7.087 \times Wg}{Ax(Dl x Dg)^{\frac{1}{2}}}$ REGIME FROM BAKER CHART	= 39.63 = 46719 DISPERSED	see page 10.7
2. CALCULATE ΔP_{gas} $Re = \dfrac{35.368 \times Wg}{Vg \times D}$ Friction factor (Moody) $\Delta PG = \dfrac{6.254 \times f \times Wg^2}{Dg \times D^5}$	$Re = 7.02\ 10^6$ F = 0.012 ΔPG = 0.376 bar/km	see page 10.1
3. CALCULATE ΔP_{LIQ} $Re = \dfrac{35.368 \times Wl}{Vl \times D}$ $\Delta P_L = \dfrac{6.254 \times f \times Wl^2}{Dl \times D^5}$	Re = 18130 f = 0.026 ΔP_L = .005 bar/km	

TOTAL TEP/DOP/DIP/EXP/SUR	PROCESS CALCULATION SHEET Sheet 1 of 2	
	TWO PHASE PIPELINES Δ P CALCULATION	ITEM : No.
BY / CHK	DATE / JOB TITLE Example	JOB No. / REV

4. AVERAGE VELOCITY

$$V_s = \frac{3.537}{D^2}\left(\frac{Wg}{Dg} + \frac{Wl}{Dl}\right)$$

V_s = 12.00 m/s

V_s = average velocity

5. CALCULATE X RATIO

$$X = \left(\frac{\Delta PL}{\Delta PG}\right)^{\frac{1}{2}}$$

X = 0.115

6. CALCULATE LOADING FACTOR WS

$$WS = \frac{Wl \times 0.205}{A}$$

WS = 78850

7. CALCULATE PH FACTOR FOR HORIZONTAL FLOW

TYPE OF FLOW	PH
DISPERSED	$(\exp\left[1.4659 + 0.49138 \ln X + 0.04887(\ln X)^2 - 0.000349(\ln X)^3\right])$
ANNULAR	(aX^b) $a = 4.8 - 0.123\, d$ $b = 0.343 - 0.021\, d$ use d = 25 max. -0.0085
BUBBLE	$\frac{(14.2)\, X^{0.75}}{WS^{0.1}}$
STRATIFIED	$\frac{15400\, X}{WS^{0.8}}$
SLUG	$\frac{1190\, X^{0.815}}{WS^{0.5}}$
PLUG	$\frac{27.315\, X^{0.855}}{WS^{0.17}}$
WAVE	$H_x = \frac{WL}{WG} \cdot \frac{\mu L}{\mu G}$: $\ln FH = 0.2111 \ln Hx - 3.993$ $\Delta P_{2h} = 6.254 \; FH\, WG^2 / \rho_G D^5$ bar/km

FLOW TYPE = DISPERSED

PH = 1.888

FLOW TYPE = WAVE

ΔP_{2H} =

bar/km

8. CALCULATE PH FACTOR FOR VERTICAL SECTION

VERTICAL SECTION	$FRN = 10.2\; V^2/D$ V in m/s, D in cm $X_D = 0.19\,(X)\,(FRN)^{0.185}$ use $X = X_D$ in Dispersed flow equ. to get PH vert.

ΔPH_v = 1.510

9. CALCULATE TOTAL TWO PHASE ΔP

Horizontal : PH = 1.888 $\Delta P_{2H} = \Delta P_G \times PH^2$ = 1.340 bar/km

Vertical : PH_v = 1.510 $\Delta P_{2v} = \Delta P_G \times PH_v^2$ = 0.257 bar/km

TOTAL $\Delta P = (\Delta P_{2H} \times L + \Delta P_{2v} \times h)/1000$ = 1.511 bar

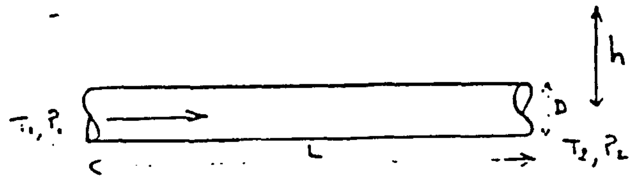
TOTAL TEP/DOP/DIP/EXP/SUR	PROCESS CALCULATION SHEET			Sheet 2 of 2		
	TWO PHASE PIPELINES P CALCULATION		ITEM No.			
BY	CHK	DATE	JOB TITLE EXAMPLE	JOB No	REV	

3. TEMPERATURE PROFILE

For detailed and accurate ΔT and ΔP calculations in 2 phase lines buried, subsea or in ai the program PEPITE should be used. The hand calculation method presented on pages 11.6, 11.7 is accurate to within 10 % for both gas and liquid lines. The procedure is easily adapted to a small programmable calculator and increases in reliability the greater the number of segments used.

The following should be remembered when designing pipelines,

- For long pipelines assuming isothermal flow can result in overdesign in pipeline siz ΔP.
- If the pipeline is constant with regard to material, insulation and burial depth along its route a fixed thermal conductivity (k) can be assumed.
- For gas pipelines the internal film resistivity is negligible - ignore it.
- For all steel pipelines the resistivity of the metal is also negligible.
- Small pipelines (< 20") have a large heat flow compared to the specific heat of the flowing medium. Consequently the gas will reach ground/sea temp in a relatively short length. For large pipelines the converse is true and a long distance is required to reach ambient.
- For oil and small gas pipelines the asymptotic temperature Ta is that of the surrounding medium. For large diameter gas lines, Ta depends largely on the Joule-Thompson effect.
- The attached calculation sheet can be used for hot lines in cold surroundings or vice versa.
- For subsea pipelines, epoxy wrapped, concrete coated resting on the bed an overall heat transfer coeff of $U = 10 - 15$ kcal/hm^2°C is a good estimate for calculation purposes.



Covering Medium :

Tg	Temperature	°C	= 10
k	Therm. cond.	kcal/hm°C	= 1.49

DATA

	Total pipeline length	m	= 20000
	No of segments		= 2
L	Length per segment	m	= 10000
Δy	Total elevation change	± m	= +100
D	Pipeline diameter	ins	= 30
	Pipeline diameter	m	= 0.762
h	Burial depth to centre	m	= 1.27
P1	Inlet pressure	bara	= 40
P2	Exit pressure	bara	= 30
ΔP	Total pipeline	bar	= 10
T1	Initial Temperature	°C	= 48

LIQUID FLOW

Volumetric flow	m^3/h	=
Density (av)	kg/m^3	=
M Mass flow	kg/h	=
Cp Specific heat	kcal/kg °C	=

GAS FLOW

Volumetric flow	m^3/d (std)	= 20000
Molecular mass		= 19
M Mass flowrate	kg/h	= 70634.5
Cp Specific heat	kcal/kg°C	= 0.6

J FLUID JOULE THOMSON COEFFICIENT = 56 °F/1000 psi (x 0.00805) = 0.45 °C/bar
(see fig. 1, page 11.8)

STEP	VALUE	NOTES
1. Calculate heat transfer factor s $x = 2h/D$ $s = 2k\pi / \ln\left[x + (x^2 - 1)^{\frac{1}{2}}\right]$	$x = 3.33$ $s = 5$ kcal/hm°C	Covering — k kcal/hmC Soil 1.49 Air 0.022 Water 0.508 Sand dry 0.30 Sand wet 1.49
2. Calculate heat flow ratio per unit length a $a = s/MCp$ (liquid or gas)	$a = 1139\ 10^{-6}\ m^{-1}$	
3. Calculate Asymptotic temperature Ta $Ta = Tg - (J\Delta P + \Delta y/jCp)/aL$	$Ta = -11$ °C	L is segment length $j = 426.5\ \frac{kg.m}{kcal}$
4. Calculate downstream temp T2 $T2 = (T1 - Ta)e^{-aL} + Ta$	$T2 = 41.4$°C	
Repeat steps 3 + 4 for each segment See sheet 2 for stepwise spreadsheet		

Btu/hrft°F
× 0.472

TOTAL
TEP/DDP/DIP/EXP/SUR

PROCESS CALCULATION SHEET
Sheet 1 of 2

BURIED PIPELINE ΔT CALCULATION

ITEM
NO.

BY | CHK | DATE | JOB TITLE EXAMPLE | JOB NO | REV

ITERATIVE CALCULATION LOG FOR A BURIED PIPELINE ΔT.

SEGMENT N°	LENGTH	ELEVATION	P1	T1	Ta	T2	P2	
	m	± m	bar a	°C	°C	°C	bar a	
1	10000	+ 100	40	48	-11	41.4	35	
2	10000	0	35	41.4	-11	35.6	30	
3								
4								
5								
6								
7								
8								
9								
10								

NOTE : 1) For more accurate calculations, correct Cp and J for each new temperature. Also recheck ΔP based on average flowing temperature of each section.

TOTAL TEP/DOP/DIP/EXP/SUR	PROCESS CALCULATION SHEET	Sheet 2 of 2
	BURIED PIPELINE ΔT CALCULATION	ITEM : No. :
BY CHK	DATE JOB TITLE : EXAMPLE	JOB No. : REV

4. LITERATURE AND USEFUL INFORMATIONS

4.1. LUDWIG VOL I chapter 2

4.2. CAMPBEL VOL I chapter 12

4.3. KATZ, HANDBOOK OF GAS ENGINEERING chapter 7

4.4. CRANE MANUAL

4.5. "Equations predict buried pipeline temperatures" G.King OGJ March 16, 1981

4.6. "Two phase ΔP computed" R. Soliman Hydrocarbon Processing April 1984

k kcal/h.m. °C x 0·672 → Btu/ft.hr.°F

Soil	(0.80) 1.19	concrete	0.65	- 1.19
Wet soil	1.49	sand (dry)	0.30	
Ground —> air	2.98	sand (wet)	1.49	
Ground —> water	29.8	Air	0.022	
Steel	38.7 (26.0)	Water	0.510	
Epoxy coating	0.67 (0.45)			
Coal tar	0.22			

Fig. 1

Joule-Thomson coefficient*

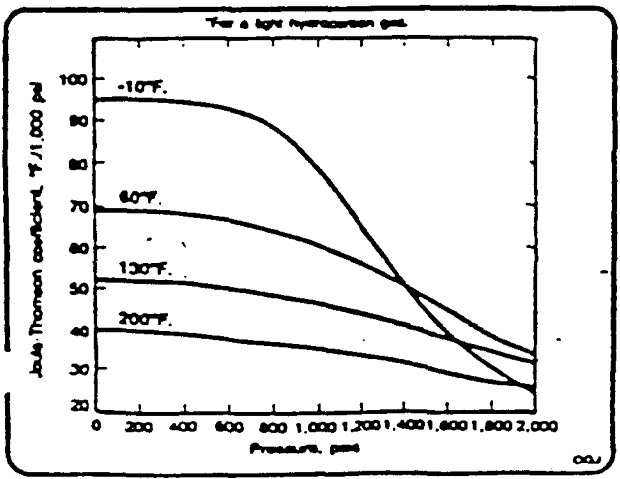
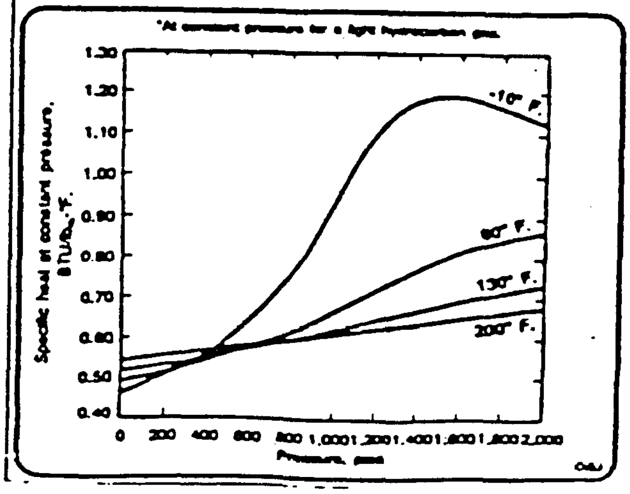


Fig. 2

Specific heats*



```
         WRITTEN BY M.A. KHAKOO, SAG/ETS 30/4/86
   (WITH ADDITION OF PRESSURE DROP AND VELOCITY BY H.TAYLOR)

:   MILLER GAS TO NSHEW - BASE CASE 346 MMSCFD, 2 inch INSULATION


 LENGTH =        18000 M       MASS FLOW    =       477265 KG/H     1/U PIPE      =      0.001008
 SEG    =           30         MOL WT       =       27.697          1/U INSULAT'N=       5.747022
TION    =          100 M       HEAT CAPACIT=        0.3307 KCAL/KGC U SOIL        =      1.642585
NE DIAM=            26 IN OD   J-T CONSTANT=         0.800 C/BAR    U OVERALL     =      0.157311
HICKNE =         0.500 IN      STD TEMP     =          273 K
       =          38.9 KCAL/HMC STD PRESS   =      1.01325 BARA
T THICK=             2 IN      GAS GRAVITY =     0.955069
T K    =         0.025 KCAL/HMC VOL FLOW    =      9269277 NM3/D
 DEPTH =           1.2 M       IN PRESS     =         31.0 BARA
/SEG   =           600         IN TEMP      =         38.0 C
       =          1.49 KCAL/HMC SOIL TEMP   =          0.0 C


   IN PRES  OUT PRES IN TEMP  OUT TEMP HEAT LOSSASYM TEMP        DEL P(PSI) PSI/100'

 0    31.00    30.79    38.00    37.74 3.13E-06 -100.786           3.004     0.153
10    30.79    30.58    37.74    37.48 3.13E-06 -101.309           3.021     0.154
10    30.58    30.37    37.48    37.22 3.13E-06 -101.843           3.040     0.154
10    30.37    30.16    37.22    36.95 3.13E-06 -102.387           3.058     0.155
 0    30.16    29.95    36.95    36.69 3.13E-06 -102.943           3.077     0.156
 0    29.95    29.74    36.69    36.43 3.13E-06 -103.510           3.096     0.157
 0    29.74    29.52    36.43    36.17 3.13E-06 -104.090           3.116     0.158
 0    29.52    29.31    36.17    35.90 3.13E-06 -104.682           3.136     0.159
 0    29.31    29.09    35.90    35.64 3.13E-06 -105.287           3.157     0.160
 0    29.09    28.87    35.64    35.37 3.13E-06 -105.906           3.178     0.161
 0    28.87    28.65    35.37    35.10 3.13E-06 -106.539           3.200     0.163
 )    28.65    28.43    35.10    34.84 3.13E-06 -107.186           3.222     0.164
 3    28.43    28.20    34.84    34.57 3.13E-06 -107.848           3.244     0.165
 )    28.20    27.98    34.57    34.30 3.13E-06 -108.526           3.267     0.166
 )    27.98    27.75    34.30    34.03 3.13E-06 -109.220           3.291     0.167
 ,    27.75    27.52    34.03    33.76 3.13E-06 -109.932           3.315     0.168
      27.52    27.29    33.76    33.49 3.13E-06 -110.660           3.340     0.170
      27.29    27.06    33.49    33.22 3.13E-06 -111.407           3.365     0.171
      27.06    26.83    33.22    32.95 3.13E-06 -112.174           3.391     0.172
      26.83    26.59    32.95    32.67 3.13E-06 -112.960           3.418     0.174
      26.59    26.35    32.67    32.40 3.13E-06 -113.767           3.446     0.175
      26.35    26.11    32.40    32.12 3.13E-06 -114.595           3.474     0.177
      26.11    25.87    32.12    31.84 3.13E-06 -115.446           3.503     0.178
      25.87    25.63    31.84    31.57 3.13E-06 -116.321           3.533     0.180
      25.63    25.38    31.57    31.29 3.13E-06 -117.221           3.563     0.181
      25.38    25.13    31.29    31.01 3.13E-06 -118.147           3.595     0.183
      25.13    24.88    31.01    30.72 3.13E-06 -119.100           3.627     0.184
      24.88    24.63    30.72    30.44 3.13E-06 -120.081           3.661     0.186
      24.63    24.38    30.44    30.16 3.13E-06 -121.093           3.695     0.188
      24.38    24.12    30.16    29.87 3.13E-06 -122.136           3.731     0.190
      24.12    23.86    29.87    29.58 3.13E-06 -123.213           3.767     0.191

 DROP TOTAL (BAR)  =   7.140                  PRESSURE DROP (PSI/100') =      0.175
 DROP TOTAL (PSI) = 103.591                   GAS VELOCITY(FT/S)       =     44.366
  TOTAL (DEG C)   =   8.417                                 (M/S)      =     13.523


                                              RECOMMENDED PRESSURE DROP FOR GAS LINES
 D VELOCITY RANGE 35 - 100 FT/S

        60 x MMSCFD x TEMP(DEG R)             OPERATING PRESS.              ACCEPTABLE DEL. P
 ORMULA :----------------                     (BARG)    (PSIG)              (PSI/100')
        DIAM.**2 x OP PRES(PSIA)              0 - 7      0 - 100            0.05 - 0.19
  (DIAM.IN INS.)                              7 - 35   101 - 500            0.20 - 0.49
                                              35 - 140 501 - 2000           0.50 - 1.20
```

(Program on disc
Lotus 1-2-3)

GLOBAL
GLOBAL ENGINEERING LIMITED

SHEET No. 1 OF
JOB No. 1096

CLIENT ______ PROJECT ______

SUBJECT ______

CALC'D BY DH CHK'D BY ______ APP'D BY ______ DATE 9th May 98

1. PIPELINE TEMPERATURE PROFILE

$\frac{W}{mK} \cdot m \cdot \frac{s}{kg} \cdot \frac{kgK}{J} \cdot \frac{J}{W \cdot s}$

Basic equations:

$$T_2 = T_{eg} + (T_1 - T_{eg}) \times e^{-\frac{U_L}{m C_p}} \qquad \text{(Ref: Dunkle)}$$

and

$$T_{eg} = T_g + \frac{m}{U_L}\left[\mu C_p \frac{dP}{dL}\right]$$

Units: $K = K + \frac{kg}{s} \cdot \frac{mK}{W}\left[\frac{m^2K}{N} \cdot \frac{J}{kgK} \cdot \frac{N}{m^2 \cdot m}\right] \frac{W \cdot s}{J}$

where T_2 = outlet temp, °C

T_1 = inlet temp, °C

T_{eg} = effective ground temp, °C

T_g = ground temp, °C

U_L = overall heat transfer coeff per unit length of pipe, W/m K

m = mass flow, kg/s

C_p = gas specific heat, J/kg K

μ = Joule-Thomson coeff, m²K/N

P = pressure, N/m²

L = length, m

2. TROLL B - TROLL 'A'

2.1 Consider 24"NB line, first 1.5 km

GLOBAL
ENGINEERING
LIMITED

SHEET No. 2 OF
JOB No. 1096

LIENT PROJECT

UBJECT

ALC'D BY CHK'D BY APP'D BY DATE 9th May 1985

T_{eq}

$T_g = 8°C$

$\dot{m} = 5\times10^9 \frac{Sm^3}{y} \times \frac{y}{365\times24\times3600\,s} \times \frac{17.4\,kg}{23.646\,Sm^3}$

$= 116.7\,kg/s$ (Pipesim used 115.2 kg/s)

$U_l = 10 \frac{W}{m^2\,°C} \times \frac{\pi\times24}{12\times3.281} \frac{m^2}{m}$

$= 19.15\,W/m\,°C$

$\mu =$ assume $5.6\times10^{-6}\,Km^2/N$ (typical)

$C_p = 2.9\,kJ/kg\,K$ (@ 117 bara & 45°C)

$= 2900\,J/kg\,K$

$\frac{dP}{dL} = -\frac{0.5\,bar}{1500\,m} \times \frac{10^5\,N}{m^2\cdot bar}$

$= -33.3\,N/m^3$

$\therefore T_{eq} = 8 + \frac{116.7}{19.15}\left[5.6\times10^{-6}\times2.9\times10^3\times-33.3\right]$

$\therefore T_{eq} = (8 - 3.3)°C$

$= 4.7°C$

T_2

$T_1 = 46.8°C$

$\therefore T_2 = 4.7 + (46.8 - 4.7)\,e^{-\left(\frac{19.15\times1500}{116.7\times2900}\right)}$

$\therefore T_2 = 4.7 + (42.1)\,e^{-0.0849}$

$\therefore T_2 = 4.7 + 38.7$

$\therefore T_2 = \underline{\underline{43.4°C}}$

SHEET No. 3 OF

JOB No. 1096

CLIENT PROJECT

SUBJECT

CALC'D BY CHK'D BY APP'D BY DATE 9th May 1985

Conclusion

PipeSim & above equations agree well, ie. within 0.1°C.

2.2. 24" N.B., 2nd segment.

T_g, m, U_l, μ and C_p are substantially the same as per the 1st segment

Teg

$$\frac{dP}{dL} = \left(\frac{-0.4}{1500} \times 10^5\right) = -26.7 \, N/m$$

$$\therefore T_{eg} = 8 - \frac{116.7}{19.15}\left[5.6 \times 10^{-6} \times 2900 \times -26.7\right]$$

$$\therefore T_{eg} = (8 - 2.64)\,°C$$

$$\therefore T_{eg} = 5.4°C$$

T_2

$$T_2 = 5.4 + (43.4 - 5.4)\,e^{(-0.0849)}$$

$$\therefore T_2 = 5.4 + (38.0)\,e^{-0.0849}$$

$$\therefore T_2 = 40.3\,°C$$

Again outlet temp agrees with PipeSim results within 0.1°C!

12. PACKAGE UNITS

1. APPLICABILITY

For many studies undertaken there will be a requirement for a gas or liquid dehydration ur in order to reduce the water content of the export phase to acceptable limits for pipelir transportation. Generally this design will be undertaken by a specialist vendor. However th engineer should be aware of some of the options available for dehydration schemes, some of the dos and donts of design and also how to undertake the basis sizing of the most commc unit (TEG). The majority of this section is concerned with gas dehydration using tri-ethylen glycol contact, this being the most widely used.

2. GENERAL DEHYDRATION NOTES

(English units are used throughout this section for convenience)

- Gas is normally dehydrated to 6 to 10 lb of H_2O per MMSCF in order to prevent hydrat formation in gas transmission lines, and reduce corrosion. Unless the gas is dehydrate liquid water may accumulate at low points and reduce the flow capacity of the line.
- Methods of dehydration in usage are :

1.	Adsorption	(Alumina, silica gel, mole sieve)
2.	Absorption	(di- or tri-ethylene glycol)
3.	Direct cooling	
4.	Compression followed by cooling	
5.	Chemical reaction	(for method injection see 4.0)

The last three methods have minor usage and are discussed elsewhere in literature.

- A summary of the advantages and disadvantages of various absorption liquids is given i Table 1.
- Tri-ethylene glycol is the preferred (most widely used) absorption liquid. Exampl flowsheets of di- and tri-ethylene glycol are given in Fig. 1 & 2.
- In order to limit the overhead glycol losses a max practical operating temp of 38 °C (10 °F) is used. A maximum of 50 °C (50 °F) is recommended to prevent problems due to the glycol viscosity.
- Glycol losses are usually in the order to 0.012 gal/MMCF (0.0016 m^3/MMm^3) due to vapourisation and in the overheads. Total losses due to leakage, vapourisation, solubilit run around 0.025 gal/MMCF (0.0033 m^3/MMm^3).
- Concentrations of TEG upto 99.1 % can be acheived without the use of stripping gas. For higher purities gas will be required.

- Glycol foams in the presence of light hydrocarbons, this can be minimised by good feed prescrubbing and addition of anti-foam agents.
- Actual gas exit dew points are usually 10-15 °F (5.5 - 8 °C) above the theoretical equilibrum dew point. Take this into account when setting the specification.
- The number of trays (or packing height) is usually small (4 trays) an excess of either is always provided in the design. Recommended efficiencies are 25 % for bubble caps 33 1/3 % for value trays. Use 24" tray spacing.
- Regenerator temperatures should not be above 400 °F (204 °C) at atmospheric pressure in order to prevent glycol degredation. Limit heat flux to 5000 - 7000 BTU/hrft2, aim for 6000. Provide at least 2000 BTU/gall pump capacity.
- To prevent hydrocarbon condensation in the glycol feed maintain the inlet temperature at 10-15 °F (5.5 - 8 °C) above the gas exit.
- Regenerator still column should run at 220 °F (104 °C) at top to prevent loss of glycol but maximise water rejection.
- Glycol circulation rates should be between 2-4 gall/lb H_2O removed 3 is a good number.

3. PRELIMINARY SIZING CALCULATIONS

An exact sizing of a TEG unit will normally be performed by the vendor on request. The CFP inhouse program "GLYCOL" also exists for estimating vessel sizes, circulation rates and utility consumptions. These are based on data from the BS+B design guide. The following hand method can be used however to estimate the required size :

1. Determine water content of inlet gas to contactor at required temp and pressure Fig. 7 lbs/MMSCF, kg/MMm^3.
2. Calculate total water mass in feed gas to contactor
3. Repeat calculation for exit gas using required exit dew point (add 10 °F) contingency). Calculate dew point depression °F, °C.
4. Calculate amount of water to be removed in contactor.
5. Use 3 galls TEG/lb H_2O evaluate glycol circulation rate.
6. Use Fig. 3 to determine required TEG concentration. %
   Use Fig. 4 to determine required stripping gas rate
7. Use 2000 BTU/gall TEG circulated to determine reboiler capacity.

8. Use Fig. 6 to determine number of trays required in contactor and Fig. 5 to determine contactor diameter.
9. Evaluate contactor height (see section 2 vessels) include integral KO pot in base c tower. Hence estimate weight of contactor.

A more detailed sizing method can be found in CAMPBELL VOL II.

4. METHANOL INJECTION (HYDRATE INHIBITION)

In order to prevent hydrate formation is gas transmission lines the product is normall dehydrated in a TEG or mole sieve unit as defined in previous sections. On some oc ior however (wellhead to plant) this is not possible due to the location of the source. If th minimum pipeline temperature is below to hydrate point the inhibition of water is require This is acheived by injection of inhibitors to depress the hydrate and freezing points.

- Common inhibitors are methanol, DEG, TEG. Recovery of inhibitors at the receivin plant is normal, the liquid being then recycled. Economics of methanol recovery are no favourable.
- Methanol is adequate for any temperature. DEG not good below - 10 °C due to viscosity limitations. Above - 10 °C better as lower vapourisation losses.
- Predict injection rate for hydrate depression as follows :

$$W = \frac{d\ M}{Ki + d\ M}\ 100$$

W = weight % inhibitor
d = °C hydrate depression
M = Mol wt of inhibitor
Ki = 1297 for Me OH
2220 for DEG, TEG

- To use above equation :
  1. Predict hydrate formation temp at max. press in line T1
  2. Estimate min flowing temperature in line T2
  3. d = T1 - T2
- The amount of inhibitor injected must be sufficient to depress the hydrate point as calculated above and also provide for vapour and liquid phase losses due to vapourisatio + dissolving. Adjust injection rate accordingly. For glycol use 0.0035 m^3/Mm^3 (0.2 lb/MMSCF), vapourisation. For methanol use vapour pressure charts (CAMPBELL pp 159).

5. SOLID BED DEHYDRATION

Solid bed dehydration is used when lower residual water concentrations are required than can be acheived by glycol units. This is generally around the - 40 °C mark or 1 ppm residual water. Solid bed dehydration can be used for less stringent design requirements providing the cost is competative when compared to TEG.

NOTES :

- LNG facilities always used molecular sieve dehydration to acheive 1 ppm H2O or less.
- Available dessicant medium :

	KgH2O/100 Kg bed	
Bauxite	4-6	cheapest
Alumina	4-7	
Gels	7-9	
Molecular Sieve	9-12	most expensive

- Beds can be severely degredated by heavy oils, amines, glycols corrosion inhibitors, salts and liquids. It is essential to have a good feed filter or scrubber prior to entering the dessicant bed.
- Bed life is usually 2-4 years depending on contamination.
- Gas flow through the bed is generally downwards. Regeneration gas flows upwards. This ensures the water is stripped from the media without having to pass all the way through the bed.
  Figures 8 and 9 show a typical molecular sieve arrangement.
- Regeneration temperature is usually 175 °C - 230 °C. Too high temp destroys the media, too low results in poor regeneration.
- Table 1 gives a summary of operating and regeneration practices.

6. USEFUL REFERENCES AND LITERATURE

6.1. CAMPBELL VOL II CHAPTERS 17 AND 18
6.2. HANDBOOK OF NATURAL GAS ENGINEERING KATZ et al. Chapter 16
6.3. PERRY
6.4. GAS DEHYDRATION "Fire tuning existing field installations"
D. CRAMER - World Oil - Jan 1981
6.5. "Cutting glycol costs I" C. SIMMONS O + GJ Sept 21 1981
"Cutting glycol costs II" Sept 28 1981
6.6. "Correlation eases absorber-equilibrum line calcs for TEG natural gas dehydration"W. BEHR 0 + 9JNOV 7 1983

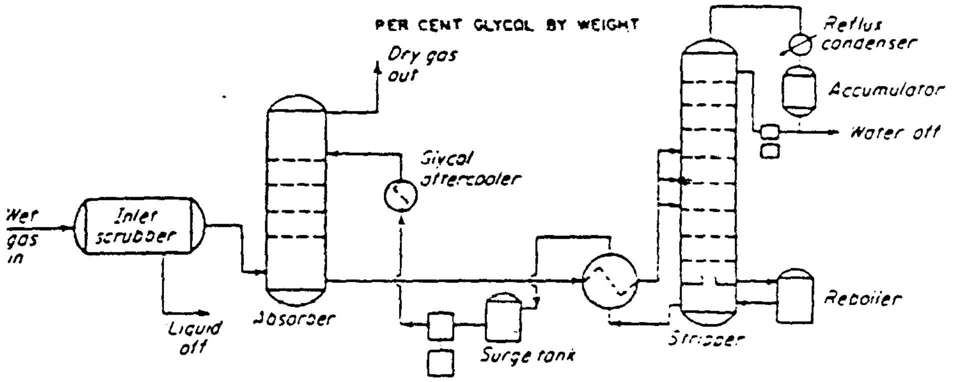


Fig. 1 Flow sheet for diethylene glycol dehydration plant. (Campbell and Laurence,

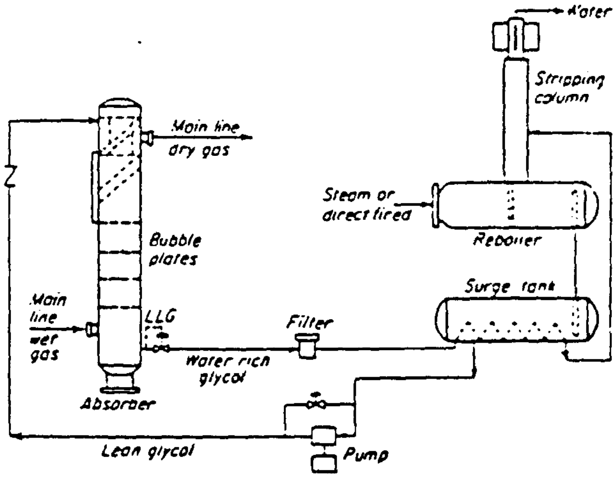
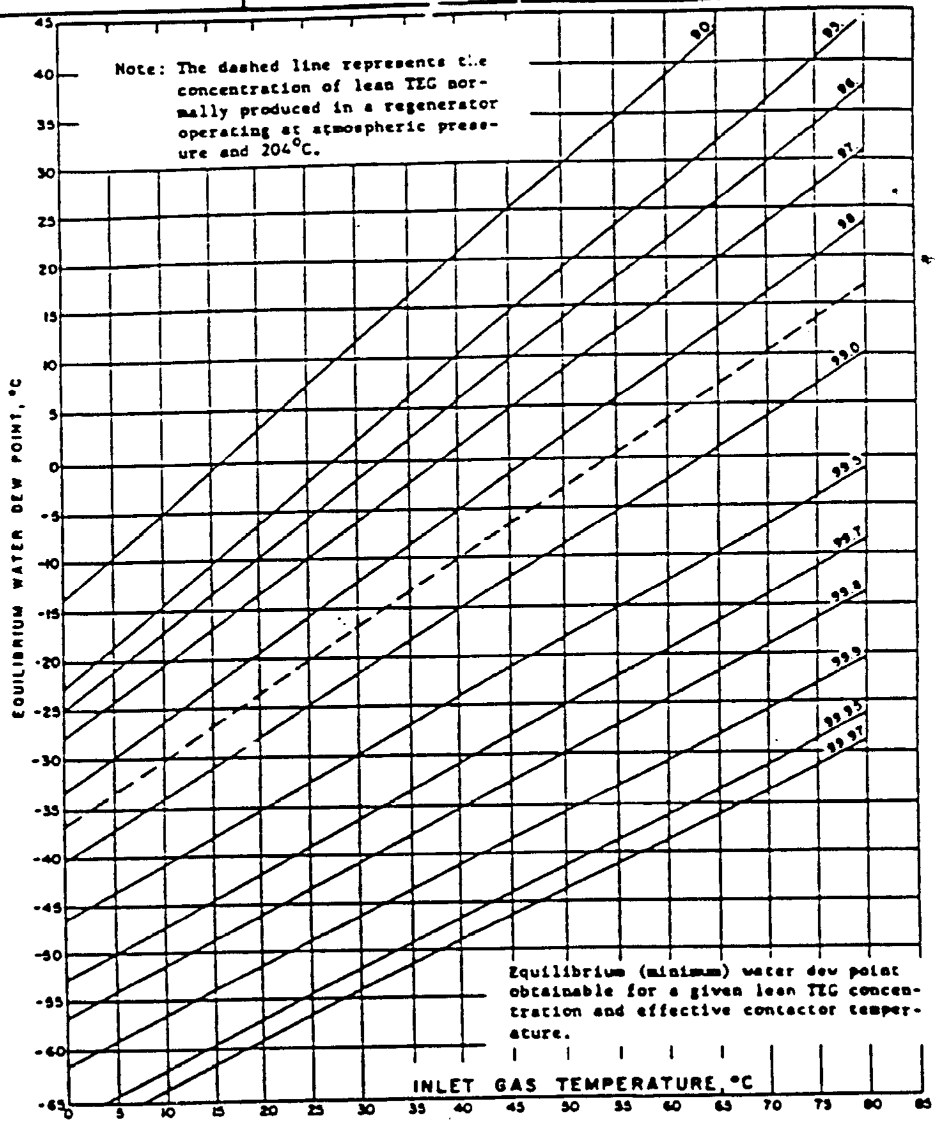


Fig. 2 Flow sheet for triethylene glycol dehydration plant. (Campbell and Laurence,

TABLE 1

SUMMARY OF ADVANTAGES AND DISADVANTAGES - ABSORPTION LIQUIDS

LIQUID	ADVANTAGES	DISADVANTAGES
Calcium Chloride	Cheap Makeup small	Emulsifies with oil. Corrodes electrolytically Low dew point depression Hydrogen sulfide forms precipitates
Lithium Chloride	High capacity for water Low corrosion rate Not hydrolyzed easily Dew point depression 22-36°C	Expensive Impurities in commercial grades cause corrosion.
10-30 percent MEA 60-85 Percent DEG 5-10 Percent Water	Removes CO_2, H_2S and water simultaneously. Glycol reduces foaming tendencies of amine. Dehydrates and purifies gas in one operation.	Greater carry over than with triethylene glycol. Limited for practical purposes to sour gas dehydration. At high regeneration temperatures needed corrosion is a problem. Dew point depression is limited and less than with triethylene glycol.
Diethylene Glycol	Stable Does not solidify in concentrated solution. Stable in presence of sulfur, oxygen and CO_2 at normal operating temperatures. Highly hydroscopic Carry over is small	Carry over greater than with triethylene glycol. Not over a 95 percent solution may be obtained easily. Dew point depression less than with triethylene glycol High initial cost
Triethylene Glycol	Does not solidify in concentrated solution. Stable in presence of sulfur, oxygen and CO_2 at normal operating temperatures. Highly hygroscopic Easily regenerated to 99 percent solution. Carry over small High dew point depression	High initial cost Exhibits some foaming tendency in presence of light hydrocarbon liquids. Defoaming agent must sometimes be added.



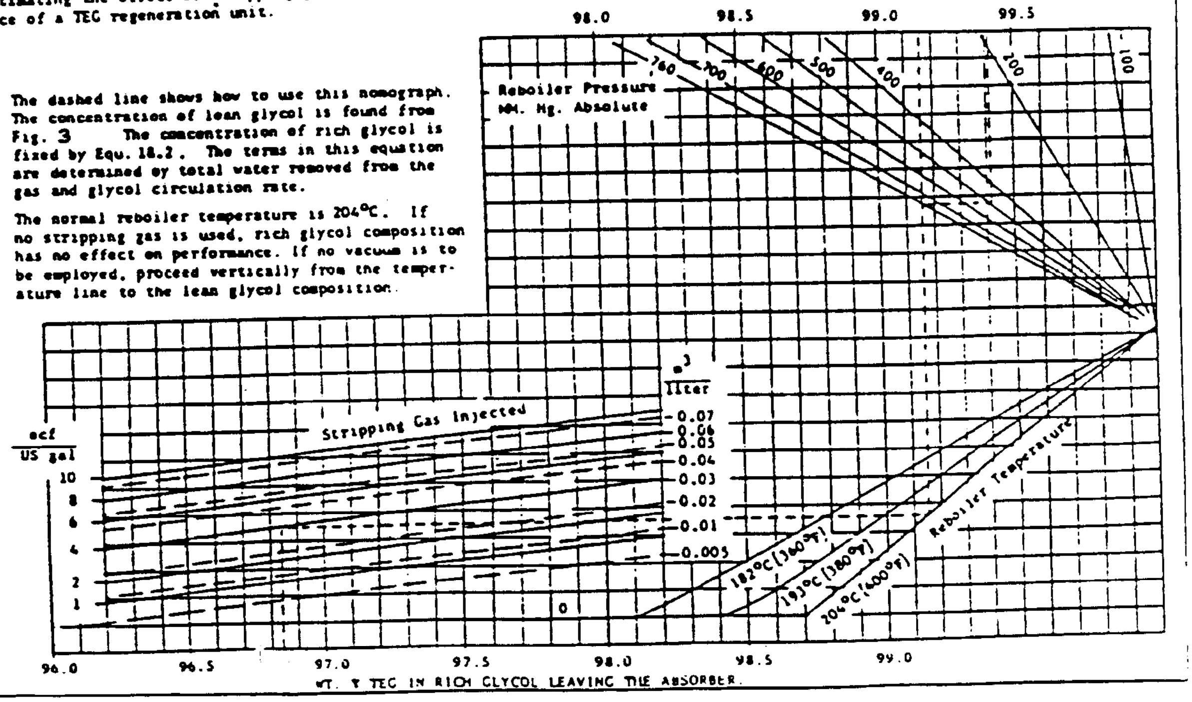
Note: The dashed line represents the concentration of lean TEG normally produced in a regenerator operating at atmospheric pressure and 204°C.

Equilibrium (minimum) water dew point obtainable for a given lean TEG concentration and effective contactor temperature.

FIGURE 3

MINIMUM TEG CONCENTRATION

Nomograph for estimating the effect of stripping gas, vacuum and reboiler temperature on the performance of a TEG regeneration unit.



The dashed line shows how to use this nomograph. The concentration of lean glycol is found from Fig. 3. The concentration of rich glycol is fixed by Equ. 18.2. The terms in this equation are determined by total water removed from the gas and glycol circulation rate.

The normal reboiler temperature is 204°C. If no stripping gas is used, rich glycol composition has no effect on performance. If no vacuum is to be employed, proceed vertically from the temperature line to the lean glycol composition.

FIGURE 5

CONTACTOR DIAMETER

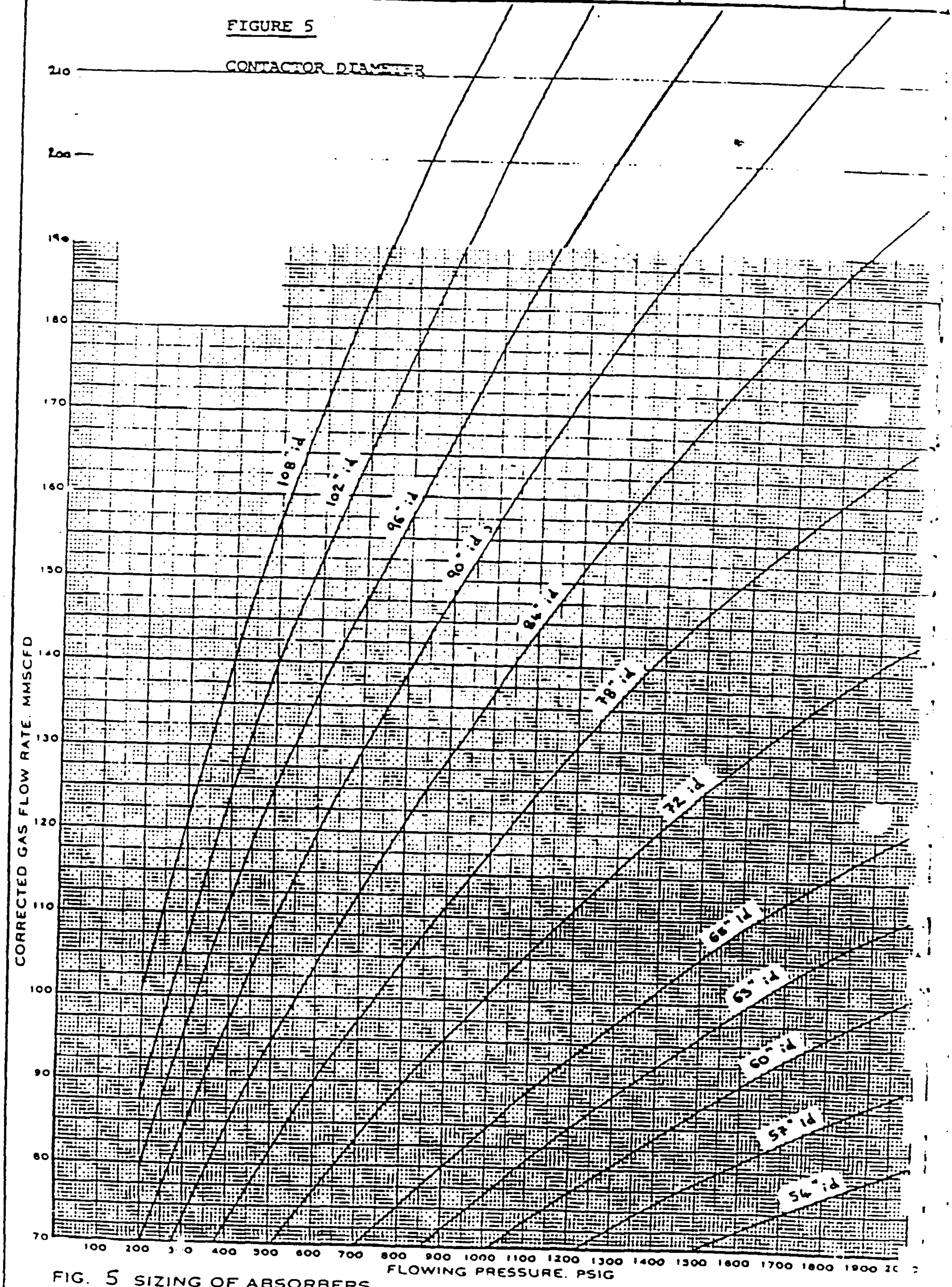


FIG. 5 SIZING OF ABSORBERS

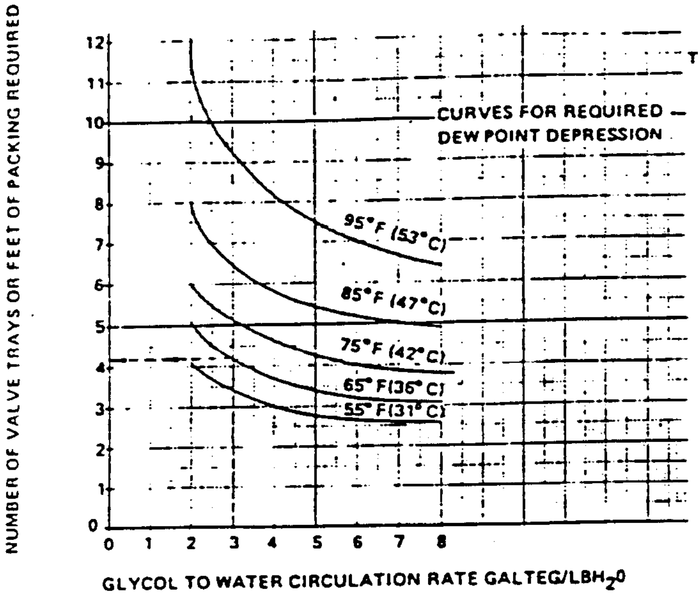


FIGURE 6
TRAYS OR PACKING REQUIRED FOR GLYCOL DEHYDRATORS

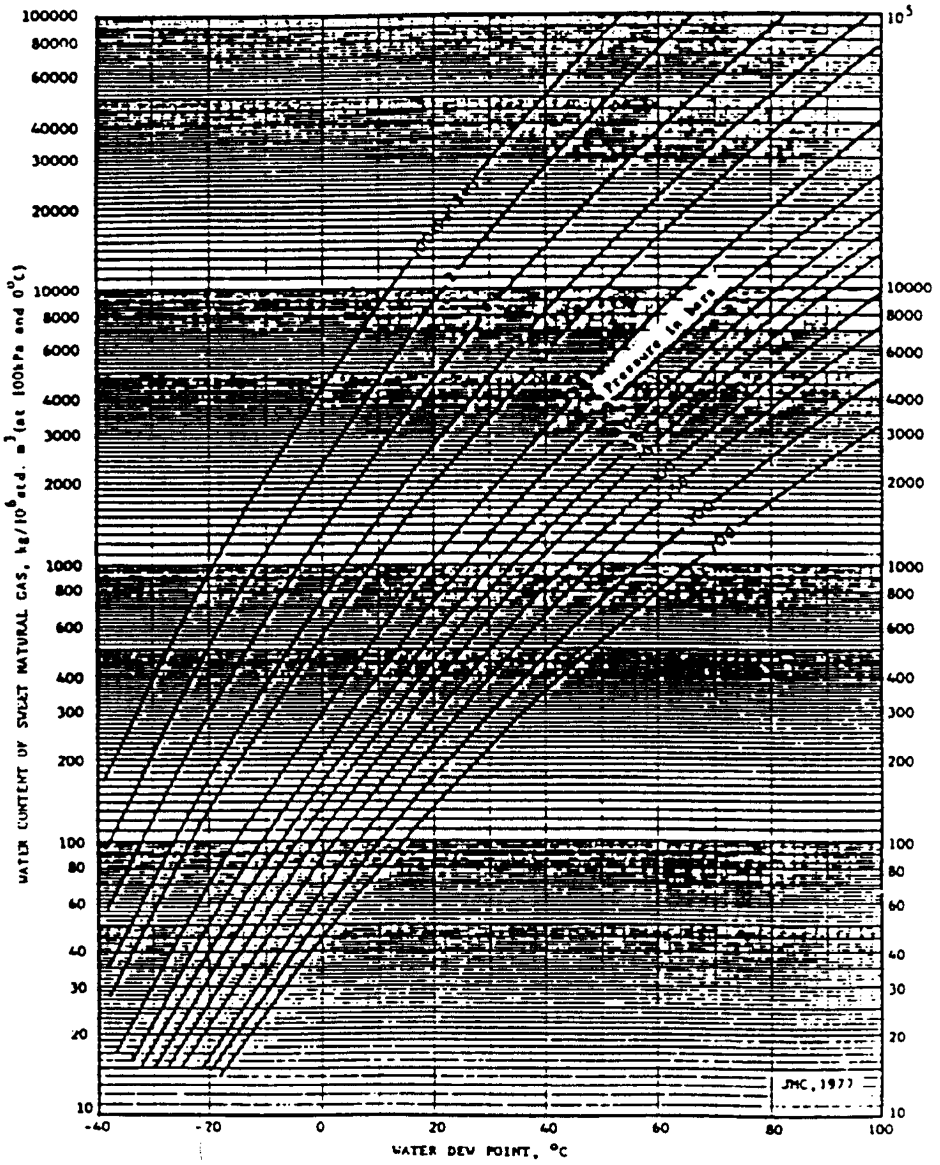


Figure 7 Water Content Of Lean, Sweet Natural Gas.

BASIC TYPE	Nominal Pore Diameter (Angstroms)	Available Form	Equilibrium H_2O Capacity* (% wt)	Molecules Adsorbed**	Molecules Excluded	APPLICATIONS
3A	3	Powder 1/16-in Pellets 1/8-in Pellets	23 20 20	Molecules with an effective diameter < 3 angstroms, including H_2O and NH_3	Molecules with an effective diameter > 3 angstroms, e.g. ethane	The preferred Molecular Sieve adsorbent for the commercial dehydration of unsaturated hydrocarbon streams such as cracked gas, propylene, butadiene, and acetylene. It is also used for drying polar liquids such as methanol and ethanol
4A	4	Powder 1/16-in Pellets 1/8-in Pellets 8 x 12 Beads 4 x 8 Beads 14 x 30 Mesh	28.5 22 22 22 22 22	Molecules with an effective diameter < 4 angstroms, including ethanol, H_2S, CO_2, SO_2, C_2H_4, C_2H_6 and C_3H_6	Molecules with an effective diameter > 4 angstroms, e.g. propane	The preferred Molecular Sieve adsorbent for static dehydration in a closed gas or liquid system. It is used as a static desiccant in household refrigeration systems, in packaging of drugs, electronic components and perishable chemicals; and as a water scavenger in paint and plastic systems. Also used commercially in drying saturated hydrocarbon streams
5A	5	Powder 1/16-in Pellets 1/8-in Pellets	28 21.5 21.5	Molecules with an effective diameter < 5 angstroms, including $n\text{-}C_4H_9OH$** $n\text{-}C_4H_{10}$** C_3H_8 to $C_{22}H_{46}$ R-12	Molecules with an effective diameter > 5 angstroms, e.g. iso compounds and all 4 carbon rings	Separates normal paraffins from branched-chain and cyclic hydrocarbons through a selective adsorption process
10X	8	Powder 1/16-in Pellets 1/8-in. Pellets	36 28 28	Iso paraffins and Olefins, C_6H_6 Molecules with an effective diameter < 8 angstroms	Di-n-butylamine and larger	Aromatic hydrocarbon separation
13X	10	Powder 1/16-in Pellets 1/8-in. Pellets	36 28.5 28.5	Molecules with an effective diameter < 10 angstroms	Molecules with an effective diameter > 10 angstroms, e.g. $(C_4F_9)_3N$	Used commercially for general gas drying, air plant feed purification (simultaneous removal of H_2O and CO_2) and liquid hydrocarbon and natural gas sweetening (H_2S and mercaptan removal).

FIGURE 8 Basic Characteriscics Of Molecular Sieves.

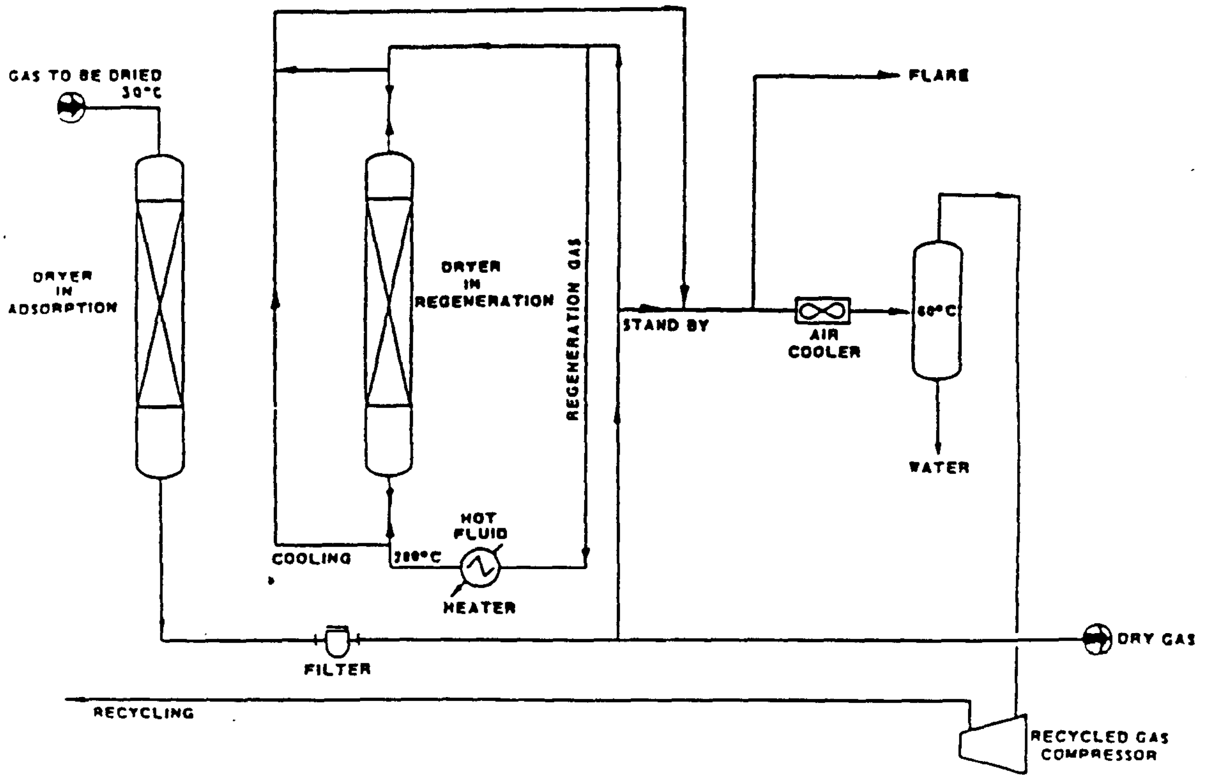


FIGURE 9 SCHEMATIC DIAGRAM OF MOLECULAR SIEVE ADSORPTION

1. APPLICABILITY

Generally natural gas, or associated gas contain acid components, mainly carbon dioxide (CO_2) hydrogen sulfide (COS), carbon disulfide (CS_2) and mercaptans.

To obtain a commercial product, gaseous or liquefied, the gas needs to be treated to eliminate these sour components for safety or process reasons. An other aspect of gas sweetening is linked with the development of the CO_2 injection to improve oil recovery. In this case CO_2 is extracted by a selective process. This section details different methods available for gas sweetening and lists their advantages and disadvantages. Guidelines are given on which system to select for specific services.

A detailed sizing method is beyond the scope of this section, but can be found in the referenced literature is required.

2. UNIT

The specification of treated gas can be given in grains/100 SCF for H_2S or sulfur content

1 grain/100 SCF = 16 ppm volume

3. GAS SWEETENING PROCESSES

Various processes are available :

- chemical absorption
- physical absorption
- chemical conversion using catalyst
- solid bed adsorption
- cryogenic fractionation

the absorption process is the most utilised

3.1. CHEMICAL ABSORPTION

In this type of process, the chemical solvent absorbe the acid components present in the feed gas by chemical reaction and releases them by heating at low pressure.

The main chemical solvents are :

- The Alkanolamines	Aqueous % normally used (wt)
. MEA (Monoethanolamine)	15-20
. DGA (R) (Diglycolamine), (FLUOR ECONAMINE)	up to 65
. MDEA (Methyldiethanolamine)	15-30
. DIPA (Diisopropanolamine), (ADIP)	30-40
- or alkaline salt solutions as potassium carbonate K_2CO_3	25-40

Alkanolamines cannot be used undiluted because :

. close to solid state at ambient conditions
. low stability at high temperature (heating is needed to extract the absorbed ac gases) with generation of highly corrosive products by decomposition.

Table 1 shows the advantages and disadvantages of these processes.

3.1.1. MEA PROCESS (see figure 1)

MEA solution was the first solvent used and is still widely used. Generally a 15 % weight solution is utilized.

a) Advantages

- high reactivity
- low solvent cost
- good chemical stability
- pubicly available (no licensing fees)

b) Disadvantages

- irreversible degradation into corrosion products by sulfur components such as COS, CS_2
- irreversible degradation for the solvent by oxygen (Direct contact witl air must be avoided)
- ineffectiveness for removing mercaptans
- high utility requirements
- high vaporisation losses
- need of reclaimer to purify the circulating solution
- no selectivity for absorption between H_2S and CO_2

c) Fields of utilization

- general use : MEA can be utilized for gases containing from 60 ppm to 15 % volume acid gases without COS, CS_2, mercaptans and with acid gas partial pressure up to 100 PSIA currently maximum capacity for a MEA unit is around 250 X10^6 SCFD.

3.1.2. DEA PROCESS

The second most widely used gas treating process with a tendancy to replace the MEA process and some improved processes exist.

Flow diagram very similar to MEA process without reclaimer.

a) Advantages

- no degradation by COS and CS_2 (hydrolysed into CO_2/H_2S)
- a significant amount of the light mercaptans present on the feed gas is absorbed
- a good chemical stability
- a very low absorption of hydrocarbons
- publicly available
- no need for a reclaimer
- reduced vaporisation losses

b) Disadvantages

- lower reactivity compared to MEA and thus higher circulation rates for the conventional system (Not applicable to SNEA-DEA process)
- an irreversible degradation of the solvent by oxygen
- higher utilities requirements
- no selectivity for absorption between H_2S and CO_2

c) Fields of utilization

The DEA process is used to treat gases containing H_2S, CO_2 and also COS, CS_2, RSH (up to a total acid gas content of 20 % volume)

H_2S content of the treated gas lower than the normal specification requirements (4 ppm volume) can be acheived. The CO_2 content of the treated gas can be as low as to 100 ppm volume. Performance of the process depends on the CO_2/H_2S ratio in the feed gas.

d) Improved processes

1. Split flow (see figure 2)

   For sour gases with high acid gas content (above 25 % mole), DEA flow rate can be reduced. Investment cost increases significantly (more equipment, complex columns, increased regenerator height).

2. SNEA - DEA process

   SNEA company has developed a process using a higher concentration of DEA (above 30 % weight).

   The process licensor claim to give in one step, for gases containing 0 to 35 % of H_2S and 0 to 35 % of CO_2, a treated gas matching the most stringent H_2S specification (4 ppm by volume).

3.1.3. DIGLYCOLAMINE (DGA) PROCESS (FLUOR ECONAMINE)

The DGA process has a limited number of units compared with MEA an DEA.

Although in the public domaine, the process was developed by FLUOR and i refered to as the FLUOR ECONAMINE process advantages and disadvantage: to compare with MEA are :

a) Advantages

- low solution circulation rate due to the concentration (same abs tio capacity as MEA)
- low utilities consumption

b) Disadvantages

- needs cooling of the solution during the absorption phase
- high solubility hydrocarbons and aromatics are dissolved
- high solvent cost.

c) Criteria of selection

Like MEA, DGA reacts both with CO2 and CS2 and a reclaimer is required. The process is applicable to gases with acid gas content from 1.5 to 30 % volume and CO_2/H_2S ratios between 300/1 and 0.1/1 at operating pressures above 15 PSIG.

3.1.4. DIPA PROCESS

This process has been developed by SHELL under the ADIP trademark name. It is characterized by the selective absorption of H_2 in presence of CO_2.

3.1.5. MDEA PROCESS

As with DIPA, MDEA is characterized by its selectivity for H_2S in presence of CO_2.

MDEA processes are proposed by process licensors :

- SNEA (D)
- UNION CARBIDE : UCARSOL

3.1.6. HOT POTASSIUM CARBONATE PROCESS (see figure 3)

An activator specific to each process licensor is added to increase the reactivity of the solution

- BENFIELD (amine and other activators)
- CATACARB (amine and other activators)
- GIAMMARCO-VETROCOKE (arsenic and others activators)

The main characteristic of the process is that the absorber and the regenerator operate at the same temperature (110/115°C)

a) Advantages

- no degradation by COS and CS_2 which are hydrolysed
- good-chemical stability
- no reaction with air
- low hydrocarbon absorption
- no need for a reclaimer
- low heat requirements (isothermal)
- selective CO_2 absorption (GIAMMARCO)

b) Disadvantages

- licensing fees required
- low reactivity with H_2S
- high water content of treated gas
- no mercaptan absorption

c) Fields of utilization

Applicable mainly on gas with high CO_2 content. Low H_2S absorption makes it difficult to achieve specification of 4 ppm volume.

Generally a two stage process will be used

- K_2CO_3 for CO_2 removal
- amine for H_2S removal

This dual system (amine / K_2CO_3) can be in some instances more attractive cost wise than an amine process.

3.1.7. CONSTRUCTION MATERIALS

Carbon steel generally utilized in the chemical absorption units. Regenerator can have a strainless steel cladding and trays. Reboiler tubes can be stainless, but still subject to corrosion. Monel is an alternative by costly, copper alloys shall be avoided.

Generally solutions treating gas with high CO_2/H_2S ratio will be more corrosive.

When the CO_2/H_2S ratio is high, stainless steel will be preferred for the following equipment : amine/amine exchanger tubes, expansion valve internals, regenerator trays and reboiler tubes.

3.2. PHYSICAL ABSORPTION

In this type of process, the solvent extracts the acid components by simple physical contact and releases them by simple expansion at low pressure.

High pressure and low temperature favour the physical absorption. Table 2 lists the advantages and disadvantages of physical solvents. These processes are applicable especially in the case of high acid gas partial pressure (above 5 bars also).

Not suitable for sweetening at low or medium pressure (10 bars abs) gases containing large amount of heavy hydrocarbons. Can be considered for a selective absorption.

The main processes are :

3.2.1. WATER WASH

Can be used as primary treatment. For absorbers water wash can be realized by addition of trays in the top section.

Because of its low efficiency, water wash should be used mainly on gases ~~sh~~ with a large amount of H_2S. Corrosion problems for this process should be considered carefully.

3.2.2. SELEXOL PROCESS (see figure 4)

- developed by NORTON CHEMICAL PROCESS can be applied to gases with large acid gas content.
- has been applied for sweetening of gases containing up to 65 % of CO_2 and 9 % of H_2S at pressure ranging from 25 to 100 bars abs.
- treated gas specification can reach 0,02 % CO_2 and 1 ppm H_2S. When used to absorb selectively H_2S or CO_2 it can also dehydrate.
- other sulfur compounds (COS, mercaptans) are also eliminated.

3.2.3. FLUOR SOLVENT

- developed by FLUOR, propylene carbonate is used as the solvent
- primarily intended for the removal of CO_2 from gas containing up to 50 % volume residual CO_2 content around 1 % volume in treated gas.
- CO_2 solubility is higher than that obtained with MEA or potassium carbonate.
- can be used to treat gas containing H_2S and CO_2. H_2S content would require a finishing treatment douwstream to obtain 6 ppm of H_2S. COS and mercaptans also absorbed.
- requires an extensive use frotating equipment.

3.2.4. PURISOL PROCESS

- proposed by LURGI uses n-methyl-2-pyr-rolidone as solvent
- as the solubility of H_2S is higher than CO_2 can be considered as a selective process to remove H_2S even in case of low H_2S/CO_2 ratio.

3.2.5. RECTISOL PROCESS

- developed by LURGI, uses a refrigerated solution of methanol as solvent. High selectivity for CO_2, primarly used on synthesis gas or on precooled gas (cooling by an external refrigerant cycle for example).
- major disadvantage of the process, when not integrated in a plant already equipped with refrigeration cycles, needs refrigeration and methanol injection.

3.2.6. ESTASOLVAN

Developed by F. UHDE GMBH uses tri-n-butylphosphate as solvent. Selective process for H_2S extraction. If CO_2 specifications on the treated gas are stringent, additional unit downstream will be required.

3.3. PHYSICO - CHEMICAL PROCESSES

3.3.1. SULFINOL

- this process has been developed by SHELL
- involves a physical solvent (sulfolane) and a chemically reactive agent (DIPA alkanolamine) in aqueous solution.

Sulfolane permits deep absorption of CO_2 and H_2S. Amine facilitates the extraction of the acid gases from solvent during regeneration

- performances for selective and non selective H_2S absorption depends on operating conditions
- process also permits extraction of mercaptans and other sulfur compounds (COS). As for physical absorption, absorption of heavy hydrocarbons occurs (mainly aromatics). Does not dehydrate the treated gas. Compared to amine processes, SULFINOL shows a low foaming tendency
- SULFINOL solution freezes at about -2°C.

3.4. SOLID BED PROCESS

3.4.1. MOLECULAR SIEVES

- not widely used for gas sweetening
- can be used as a finishing treatment to remove mercaptans
- absorption in molecular sieves is particularly well adapted for LPG as finishing treatment to obtain the sulfur content specifications of propane and butane
- good absorption capacity for H_2S low for CO_2. They remove water preferentially
- sieve life is reduced for gases with high CO_2 and H_2S content

3.4.2. IRON SPONGE PROCESS

- could be also classified as absorption process or as a conversion process (H_2S is converted to sulfur)
- mainly applied to gas with low H_2S content
- discontinuous process, iron oxide has to be regenerated or replaced. Spontaneous combustion of the fouled product occurs with air.

4. CRITERIA FOR SELECTION OF ABSORPTION PROCESSES

- there is no multipurpose process for gas sweetening, each case is specific and shall be studied accordingly
- final selection is done on the basis of economical criterial from short list of processes which seem appropriate to satisfy the treated gas specifications

- chemical processes are characterized by their ability to absorb acid gases with a low influence of the gas pressure. They require a large heat quantity for regeneration
- physical processes performances are more dependant on gas pressure. At high pressure with high acid gas partial pressure, the absorption is better than for chemical processes
- selection criteria listed herebelow can be used for preselection of sweetening processes but shall not be considered as definitive.

4.1. CO_2 ABSORPTION (NO H_2S IN THE GAS) (see figure 5)

4.2. SIMULTANEOUS ABSORPTION OF CO_2 AND H_2S (see figure 6)

4.3. H_2S ABSORPTION (NO CO_2 IN THE GAS) (see figure 7)

This is not a frequent situation with natural gases.

4.4. H_2S SELECTIVE ABSORPTION (H_2S AND CO_2 IN THE GAS) (see figure 8)

Physical solvents are particularly well adapted in this case.

Among the chemical processes, only MDEA and DIPA seem to be adapted for this service.

5. REFERENCES AND USEFUL LITERATURE

(1) Natural gas production transmission and processing
F.W. COLE, D.L. KATZ, L.S. REID, C.H. HINTON

(2) Gas conditioning and processing (volume 4) gas and liquid sweetening by ROBERT N. MADDOX edited by JOHN M. CAMPBELL.

FIGURE 1

MEA PROCESS
FLOW DIAGRAM

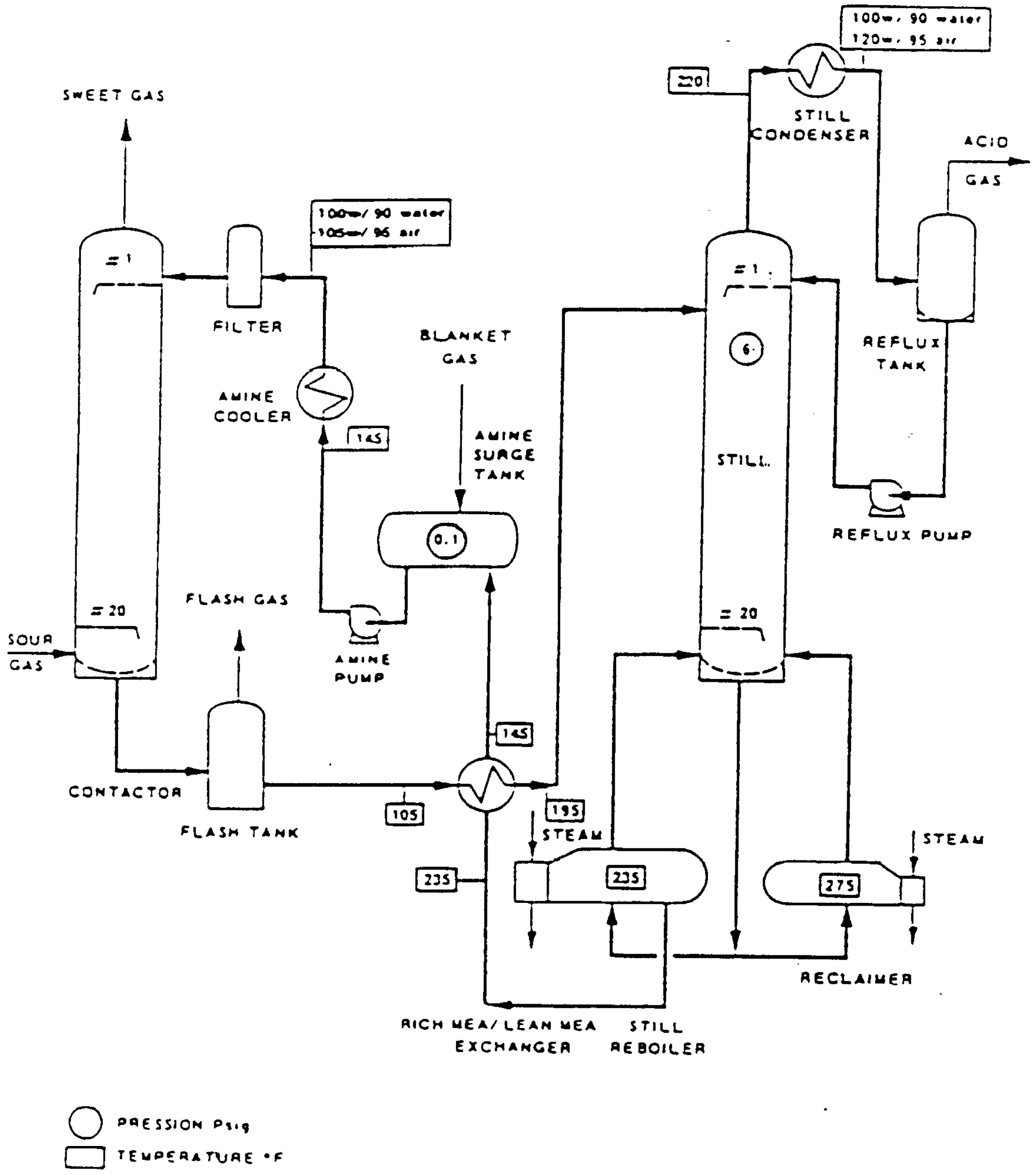


TABLE 1

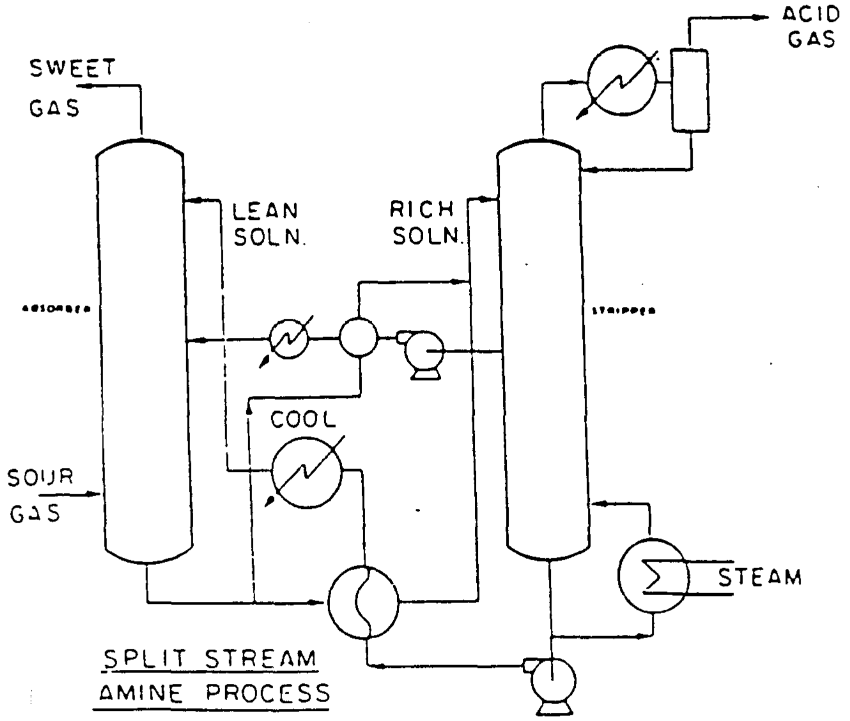
ADVANTAGES AND DISADVANTAGES OF ABSORPTION PROCESSES BY CHEMICAL SOLVENT

ADVANTAGES	DISADVANTAGES
Low sensitivity to acid gas partial pressure	Exothermal chemical reaction requiring high energy quantity for regeneration (to extract acid gases from the solution)
Slight incidence of the operating pressure on the treated gas purity	Risk of foaming when the solution is polluted (hydrocarbons, suspended solids)
Very slight absorption of heavy hydrocarbons possibly present in the feed gas	Corrosion problems resulting from solution oxidation (contact with air) or from solution superheating during regeneration.
Most of the processes available in the public domain (no licensing fees)	Pure water required to dilute the solvent
Acid gas/solvent equilibrium data available in the litterature	High water content of the treated gas
Reasonable cost of solvents	Heating required for solvent storage.

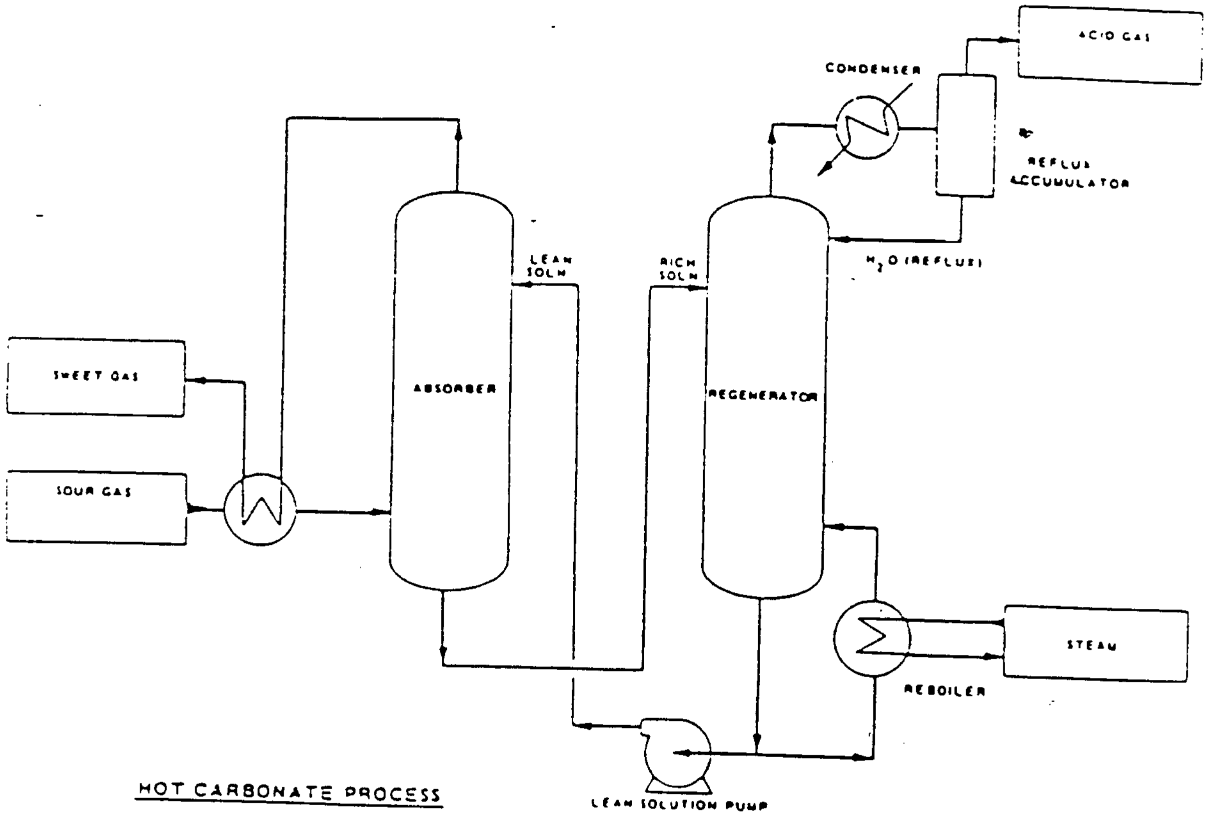
TABLE 2

PHYSICAL SOLVENTS

ADVANTAGES	DISADVANTAGES
. Low energy requirements for regeneration	. High sensibility to acid gas partial pressure
. Reduced corrosion compared with chemical processes (amines)	. Not suited for gas at medium pressure (associated gas)
. Low water content of the treated gas	. Absorption of heavy hydrocarbons from the feed gas
. No need for pure water for dilution	. Licensing fees
. No heating required for solvent storage	. High cost of the solvents
. Selective processes.	

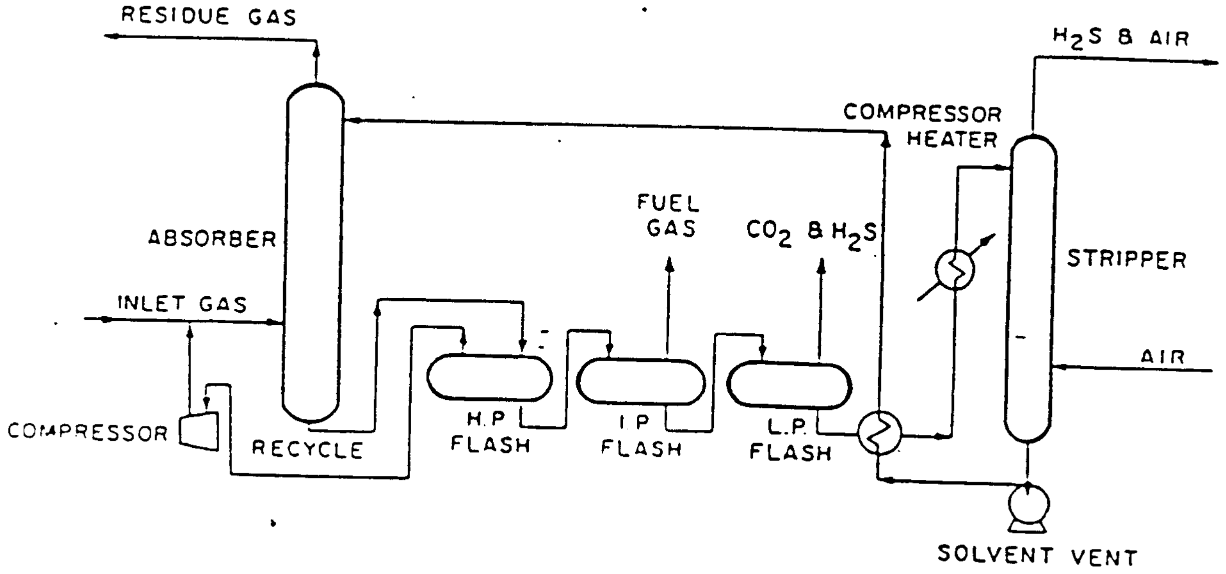


SPLIT STREAM AMINE PROCESS



HOT CARBONATE PROCESS

FIGURE 3



SELEXOL PROCESS FLOW FOR HIGH CO_2 AND HIGH H_2S

FIGURE 4

FIGURE 6

H_2S SELECTIVE ABSORPTION FROM GAS CONTAINING H_2S AND CO_2

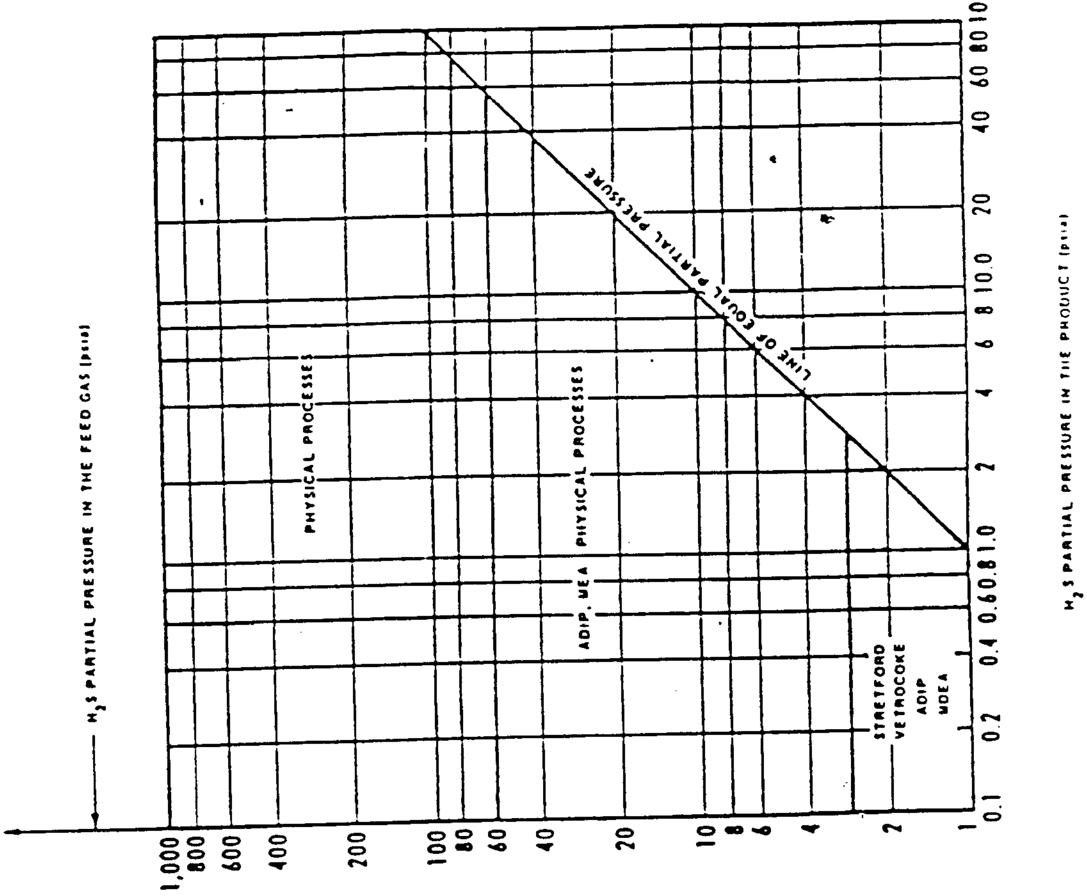


FIGURE 5

SWEETENING PROCESSES FOR GAS CONTAINING H_2S ONLY (NO CO_2)

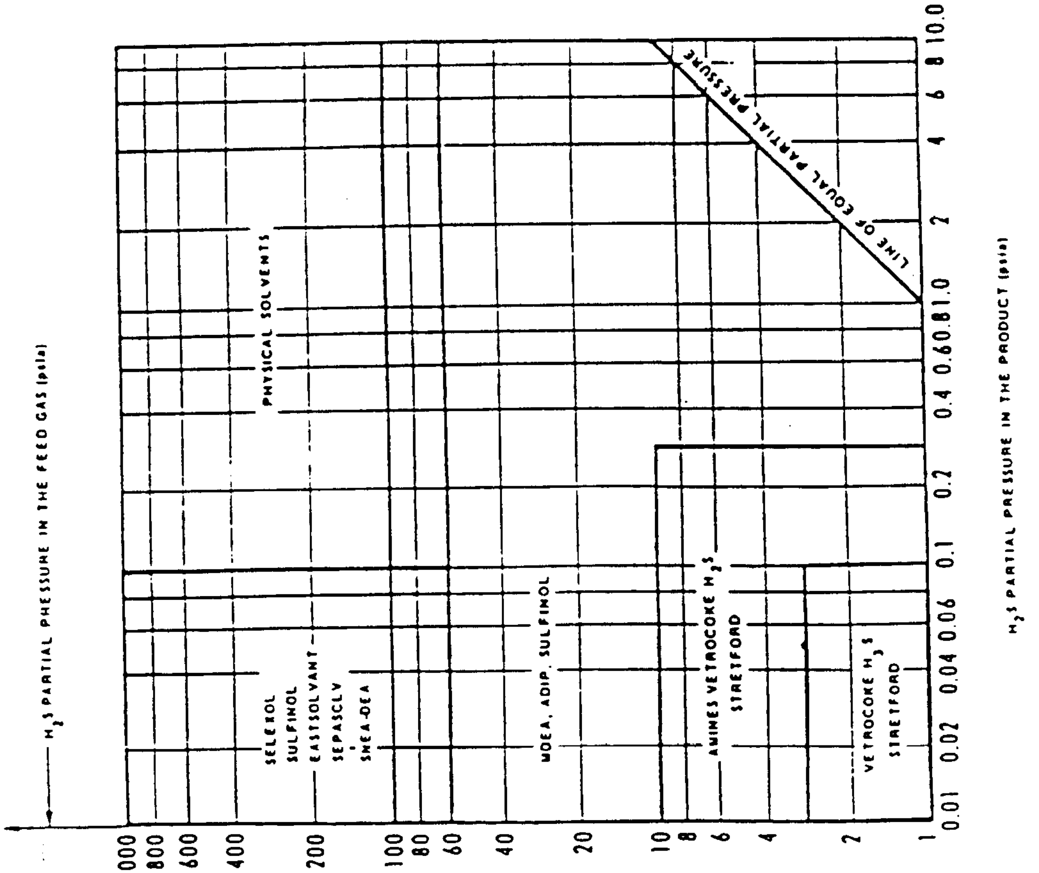


FIGURE 8
SWEETENING PROCESSES FOR GAS CONTAINING CO_2 AND H_2S

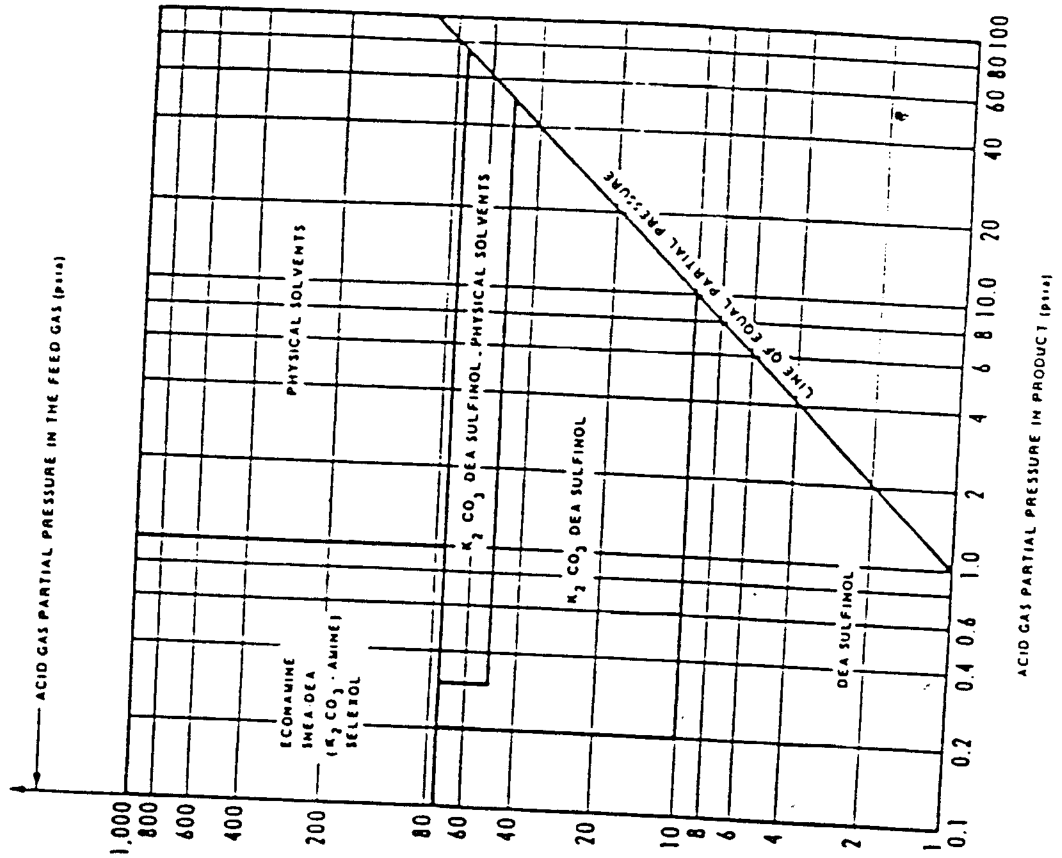
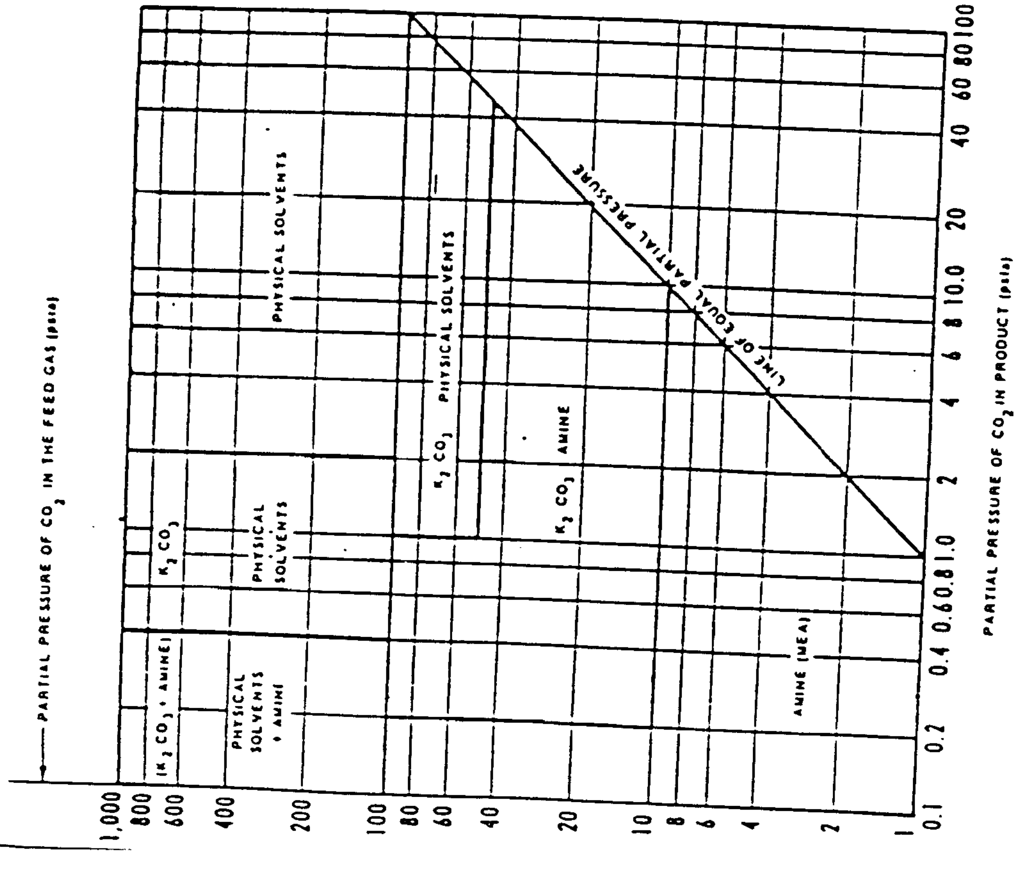


FIGURE 7
SWEETENING PROCESSES FOR GAS CONTAINING CO_2 ONLY (NO H_2S)



1. APPLICABILITY

For both feasibility and pre-project studies, the engineer will be required to select a process scheme (i.e. : choice between cold frac and refrigeration system) to estimate the power, utilities, weight of this package.

2. DESCRIPTION

The description is based on simple cycle. A refrigeration cycle is based on the exchange between a hot source and a cold source. The cold source is the air or the water, the hot source is the gas to refrigerated. (see figs 1, 1A)

Figure 1 shows such a cycle where :

t_1 : is refrigerated stream temperature
t_2 : is condensed refrigerated stream temperature
P_1 : is vapor pressure of the refrigerated stream at t_1
P_2 : is vapor pressure of the refrigerated stream at t_2
P_d : compressor discharge pressure
t_d : compressor discharge temperature
T_1/T_2 : initial and final temperature of the hot source
TR_1/TR_2 : initial and final temperature of the cold source

On the figure 1A, it is easy to explain the cycle on a pressure enthalpy diagram :

ΔQ_1 : is the duty of the process to be refrigerated
ΔQ_2 : is the duty of the condenser
ΔQ_2 : $\Delta Q_1 + HP$
HP is the power of the compressor

3. MODIFICATIONS - ECONOMISER

During the discharge of the cryogenic refrigerant, a mixed phase is generated (vapor and liquid). Only the liquid phase participates in the cooling duty. The vapor phase being compressed from low pressure to the high pressure without participation at the refrigerated duty.

It is possible to remove a part of this vapor phase by addition of an intermediate pressure stage removing the vapor from the low pressure stage compressor which is called economiser (see figures 2 and 2A). An economiser is widely used in the industry.

4. SELECTION OF THE REFRIGERANT

Depends on the required final temperature of the hot source and the disponibility of t country where the units are installed.

Tables 1 shows the performances of different refrigerants in various conditions.

It is recommended that the compressor suction pressure be maintained above atmosphe pressure.

5. CHOICE OF DIFFERENT PARAMETERS

5.1. REFRIGERATED STREAM TEMPERATURE t_1 AND COMPRESSOR SUCTION PRESSURE P_2

t_1 to be 3 to 6°C lower than the final temperature of the hot source T_2.

With the selected refrigerant and t_1 read on the MOLLIER diagram of the selecte refrigerant the vapor pressure.

5.2. CONDENSED REFRIGERATED STREAM TEMPERATURE t_2 AND COMPRESSOR DISCHARGE PRESSURE P_d

In first, estimation take $t_2 = TR_1$ (initial temperature of cold source) + 15 or 20° t_2 to be checked later if it is compatible with the cold source flowrate and the pinch of the condenser (pinch are shell and tube § 3 note 2) which should be 3°c minim .

With t_2 determine P_2 which is the vapor pressure of the refrigerant at t_2 (read MOLLIER diagram). Compressor discharge pressure = $P_d = P_2 + \quad P$ through th condenser.

5.3. PRESSURE IN THE ECONOMISER P_e

This pressure will be finalized with the compressor manufacture but for an estimatio take :

$$P_e = P_1 \sqrt{\frac{P_2}{P_1}}$$

6. CALCULATION WITH ECONOMISER (see figure 2A)

Step 1 Determine refrigerant circulation through the economiser = m_2

$$m_2 = \frac{\Delta Q_1}{H_2 - H5}$$

$$\text{\% vapor at evaporator inlet} = \frac{H_5 - H_3}{H_2 - H_3} \times 100$$

$$\text{\% liquid at evaporator inlet} = \frac{H_2 - H_5}{H_2 - H_3} \times 100$$

Step 2 Determine vapor refrigerant circulation rate through the economiser = m_1

$$\text{\% vapor at economiser inlet} = \frac{H_1 - H_5}{H_4 - H_5} \times 100$$

$$\text{\% liquid at economiser inlet} = \frac{H_4 - H_1}{H_4 - H_5} \times 100$$

$$m_1 = m_2 \frac{\text{\% vapor at economiser inlet}}{\text{\% liquid at economiser inlet}} = m_2 \times \frac{H_1 - H_5}{H_4 - H_1}$$

Step 3 Determine refrigerant circulation through the condenser = m

$$m = m_1 + m_2$$

Step 4 Duty of condenser

$$\Delta Q2 = m (H_d - H_1)$$

Step 5 Calculate the compressor discharge temperature and power (see compressor chapter).

Step 6 Check the pinch in the condenser and the cold source flowrate (if not acceptable select a new t_2 and P_d and repeat the calculation).

Step 7 Size drums evaporator and condenser (see vessels and shell and tube exchanger chapter 5).

7. SELECTION OF MATERIAL

The material selection to be made carefully. We recommend to take the temperatur corresponding at the vapor pressure at the atmosphere pressure (i.e. : for propane it is recommended to select the low temperature killed carbon steel).

8. MULTISTAGE CYCLE

If we look at the enthalpic curves of the exchanger (process refrigerant) with or withou economiser, it is obvious to see that the area between the process stream and the refrigerant stream is proportional to the compressor work (in a first approximation).
It is possible to reduce this area by addition of several pressure levels between the discharg and the suction of the compressor. However, there is more equipment (drums, exchangers, regulation) and the compressor is more complicated.
The number of pressure levels is an economical problem but the maximum is 3 or 4 selections of these pressures : the pressure ratio between each pressure is for a firs estimation.

$$\pi = \sqrt[n]{\frac{P_1}{P_2}}$$

n = number of compresser suction
P_1 = first compressor suction pressure
P_2 condenser pressure

9. REFERENCES AND USEFUL LITERATURE

9.1. Gas conditionning and processing volume 2 by Dr. John M. CAMPBELL

9.2. Applied process design for chemical and petrochemical plants volume 3 by Ernest E. LUDWIG

9.3. Chemical Engineers Handbook by Robert H. PERRY/CECIL M. CHILTON

REFRIGERATION CYCLE
FIGURE 1

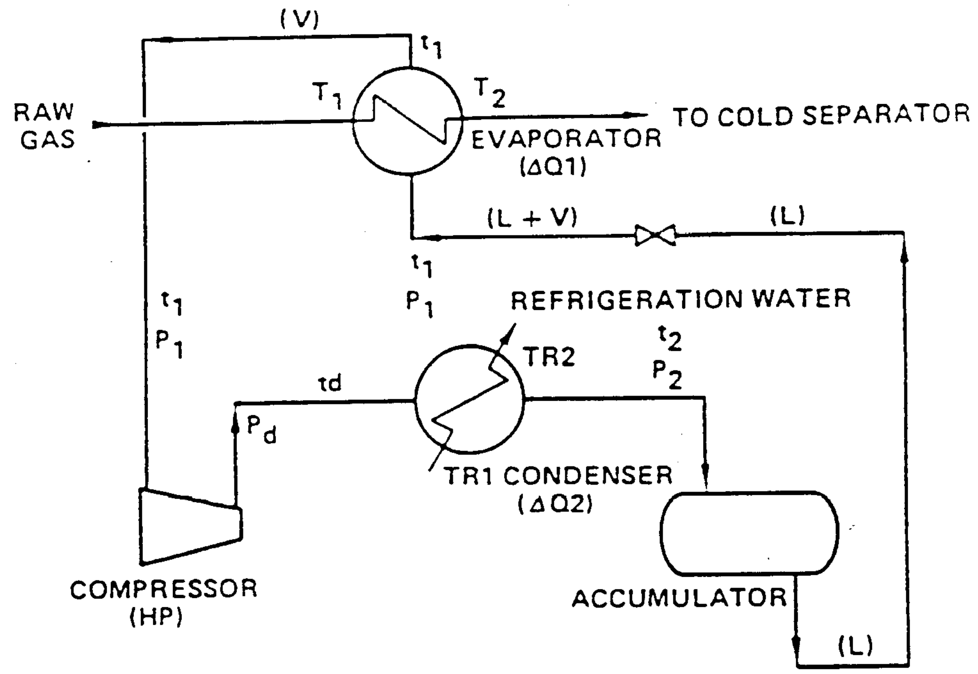
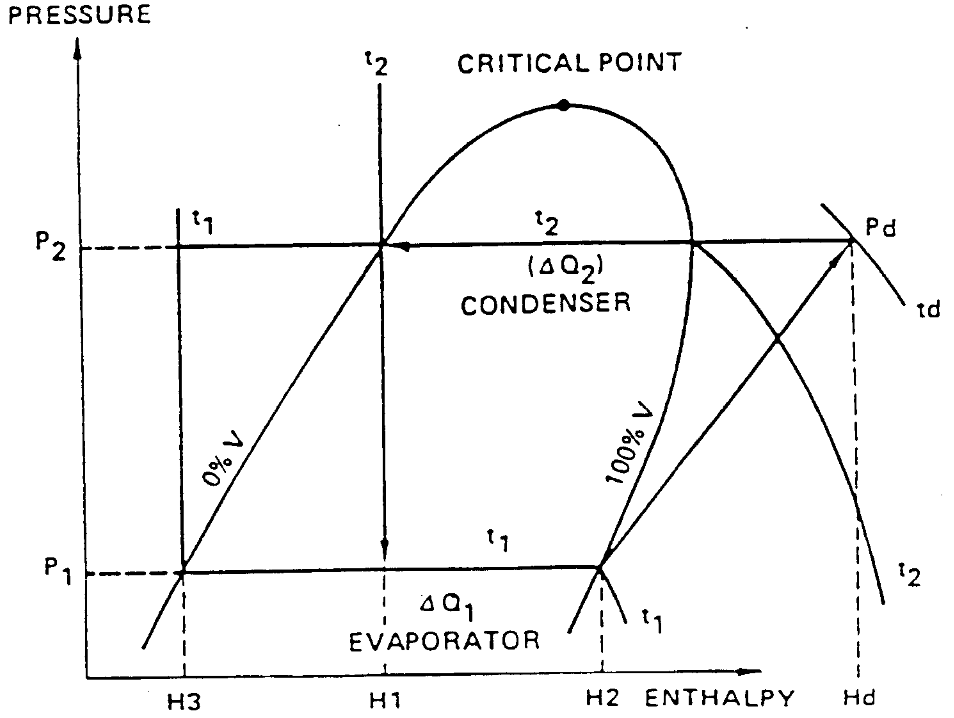


FIGURE 1.A



REFRIGERATION CYCLE WITH ECONOMISER
FIGURE 2

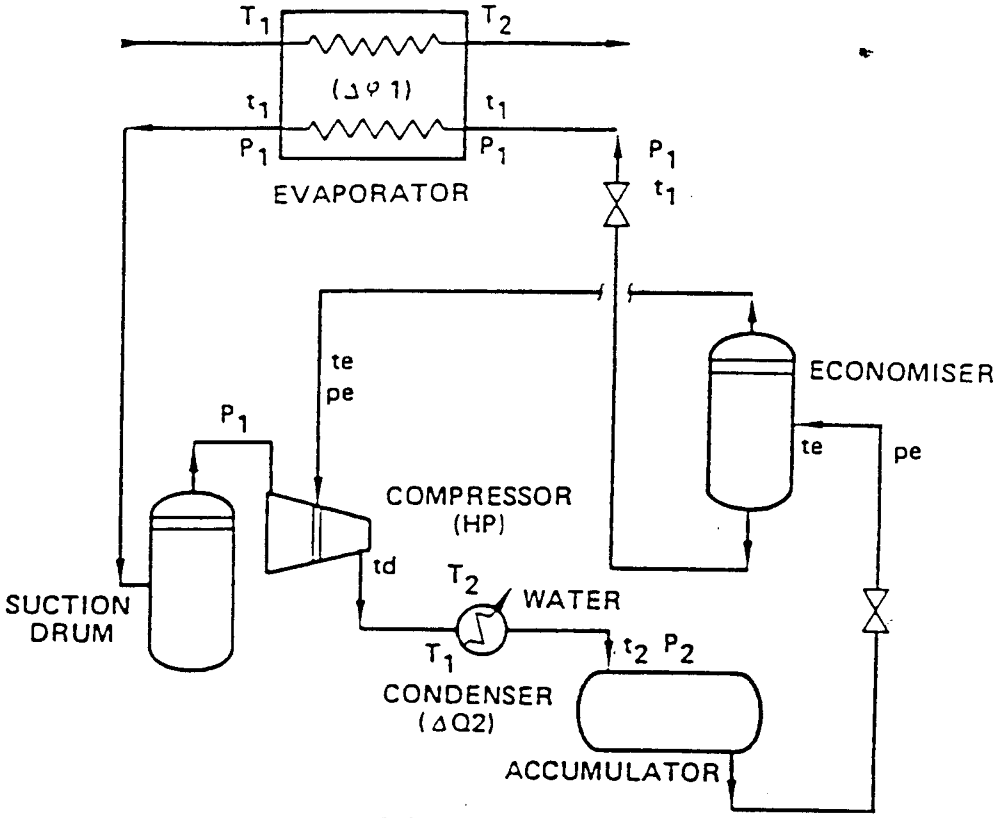
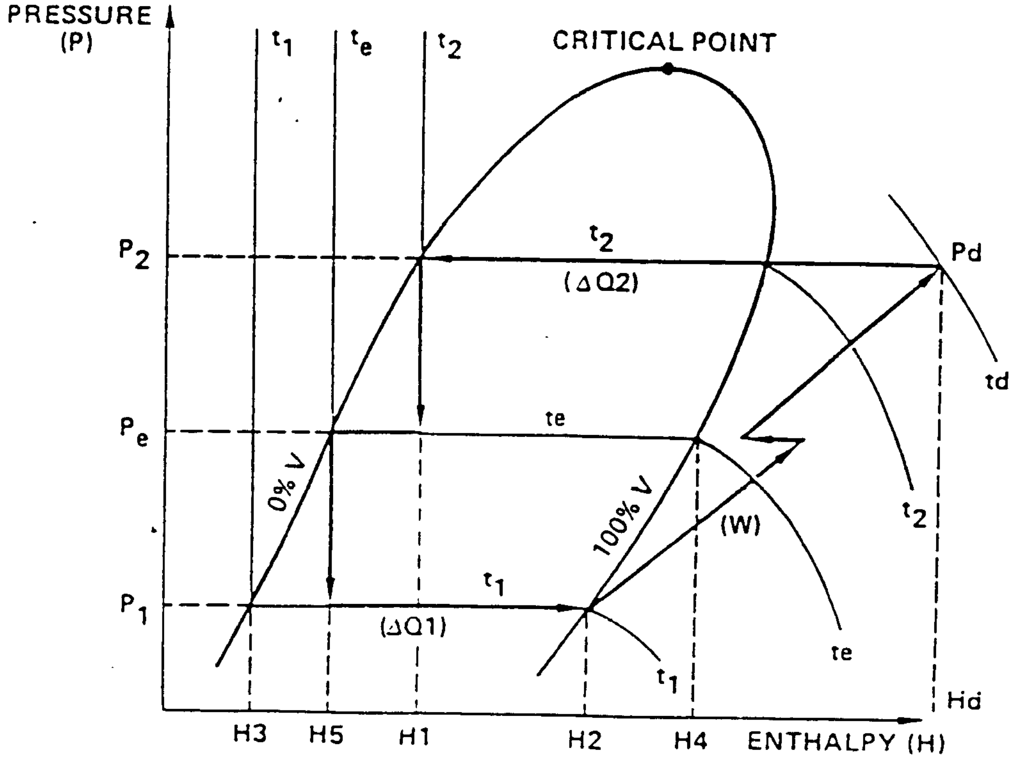


FIGURE 2.A



13. UTILITIES

1. APPLICABILITY

For both feasibility and pre-project studies the engineer will be required to make an estimate of utility requirements both in consumptions and equipment required.

This section details a few guidelines and notes on the following utilities :

WATER TREATMENT
UTILITY AND INSTRUMENT AIR
INERT GAS GENERATORS
WATER SYSTEMS
FUELS

2. WATER TREATMENT

The following details the common used effluent water treatment equipment given in order of effluent quality :

API gravity settler

- Usually the first line of clean up. Simply a settling tank with a top skimmer to remove any floating oil or debris and a bottom skimmer to remove sludge.
- Effluent quality around 150 microns globules and 150 ppm oil.
- Large bulky items, cannot be used offshore. Either circular or rectangular in design.
- Simple, cheap very common in onshore use.

Tilted Plate Separator (TPS), Corrugated Plate Interceptor (CPI)

- Widely used both offshore and onshore.
- Uses plate packs, usually at 45° mounted in a tank and relies on gravity settling between oil + water within the spaces between the plates.
- Effluent quality down to 60 micron oil globules and 50-200 ppm.
- Can have problems with high solids content if upstream settling tank is not installed.
- TPS units are usually used as the first treatment stage offshore.

Flotation units

- Uses induced or dissolved air flotation to remove any residual solids/oil in the effluent. Works in reverse to a gravity settler (small air bubbles trap debris and float to top of ank).

  - Effluent quality better than 40 ppm. Vendors usually guarantee < 30 ppm.

- Can be used both offshore and onshore. Usually installed downstream of a TPS unit or API separator.

Filtration (Use for Water injection systems only)

- Filtration units either uses media beds (sand, anthracite, garnet, walnut shells) to achieve water quality or filters (fibre socks, mesh, stainless steel cage).
- Usually not required for effluent water treatment unless very low residual solids content enforced by local effluent standards.
- More commonly used for water re-injection where high quality, low solids level is required. Can achieve 1-2 ppm in certain beds, 10-15 ppm is more common.
- Units are generally compact but heavy due to media bed weight.
- Good pre-filtration is required to prevent fouling up of main bed units.

Effluent standards

Listed below are maximum residual oil content in effluent water for dumping to sea :

NORTH SEA	40 ppm
INDONESIA + MIDDLE EAST	30 ppm
CHINA	20 ppm
LOCAL ESTUARY (river)	5 ppm (European standard)

Process drains, produced water, deck (site) drains (see figure 1)

- Produced water may need degassing before treatment. If the amount of dissolved gas is small it may be possible to handle it in the TPS unit.
- Deck or site drains normally flow to a separate sump tank before de-oiling. If the deck drainage is small or produced water flow is small can combine both streams through one TPS unit.
- Process drains are normally manually initiated and pass directly to the return oil stop tank. These drains are generally water free.
- Always try to use gravity feed between units. Pumping can cause emulsions and make oil-water separation harder. Similarly avoid fast flowing lines and turbulent pipe arrangements.

3. UTILITY AND INSTRUMENT AIR (see figure 2)

Compressed air is used on plants for instrument control, turbine and engine start-up an utility services eg : pneumatic tools, cleaning, etc. :

- For turbine/engine start-up 17-25 bar supply will be required.
- For general instrument and utility air, compressor discharge around 9 bar is adequate.
- Consumption : use 0.6 scfm (0.017 m^3/min) for each air pilot (= valve)
  (instrument air) 0.8 scfm (0.022 m^3/min) for valve positioner
  generally 1 m^3/h per valve unit will do as first estimate.
  Add 25% to compressor capacity for design.
- Utility air : consumption is intermittent and difficult to estimate at early project stage
  Add 75-100 scfm (130-170 m^3/h) to compressor capacity for initial estimate.
- All plants should have 100% instrument air standby capacity.
- Utility and instrument air can be supplied from same compressor or separate one depending on capacity requirement.
- Instrument air must be dried before use. Dew point of air is dependant on minimum air temperature in location of unit. Generally dessicant bed driers are used giving dew point as low as -60°C.
- Size air receivers to give 10-15 minutes of instrument air assuming the compressor goe down. Pressure in the instrument air receiver should not fall below 80 psig (5.5 bar g) during this period.
- For long air transmission headers in cold climates intermediate KO puts may be required.
- An estimate of compressor and dryer weights and power are given in figure 3.

4. INERT GAS GENERATORS (N_2, CO_2)

Inert gas is required in all plants for purging and inerting of equipment. For smal requirements N_2 bottles can be used in racks, this however is not feasible for large units an so gas generators must be supplied. The main types of generator in use are :

- cryogenic distillation of air
- oxygen absorbtion on sieve
- gas combustion

- For purging purposes estimate capacity based on 3 times the volume of the largest vesse to be purged in one hour.

- Cryogenic distillation is used only for large volume requirements, specifically LNG plants. Not used offshore.
- Gas combustion produces a N_2, CO_2 mixture for inerting and purging purposes. Not used much these days except for onshore large volumes.
- Pressure swing absorbtion is the must common used method for N_2 generation. Skid mounted units are sometimes supplied with dedicated air compressor, or can use existing plant air compressor for supply. Air consumption is 4-5 times inert gas production rate, residual O_2 in gas is 1-2%-3%.
- Details and weights of common units are given in figures 4 and 5.

5. WATER SYSTEMS

Seawater

- Used for cooling purposes both onshore and offshore. Can also be used as wash water, sanitation water and feed to potable water units.
- Seawater is also used for fire water systems but is usually a separate system. The seawater cooling circuit is normally connected to the fire water ring for emergency supply only.
- Always coarse filter the seawater before circulating to the plant. This removes any debris or marine life.
- Treat with chlorine at 1-2 ppm concentration - maintain a residual Cl^- level in the water exit at 0.3-0.5 ppm.
- Seawater exit temperatures to outfall canals or drain caissons should not be above 40°C to prevent corrosion.
- Once through water systems are preferred for small cooling duties with only 3-4 exchangers. For large duties and number of units where the cost of corrosion proofing is prohibitive consider using a closed loop cooling medium system. Common used is 25% TEG in water.
- For cooling medium/seawater exchangers consider using titanium or similar plate exchangers. These are especially ideal offshore due to reduced weight and space requirements.

Potable water

- Depending on location of plant potable water can be made in situ or supplied by tanker for storage, or taken direct from a mains supply.
- For onshore plants most common method of water supply is evaporative distillation. Problem with these units is size and weight are high and residual TDS (Total Dissolved Solids) is 5-10 ppm. This results is a bland distilled water which is not pleasant to drink.
- Increasingly popular now are Reverse Osmosis units (RO) which are lighter and need less maintenance than evaporative units. Water quality is 400-500 TDS (World Health Organization TDS for drinking water is 500-1000) operating costs of RO units is 1.5 times that of evaporative distillation.
- Power consumptions : for a 100 gph (0.38 m^3/h) unit.

Evaporative Distillation unit	(EO)	3.5	kw
Reverse Osmosis	(RO)	5	kw
Vapor Compression unit	(VC)	15	kw

- Most common unit offshore at present is the VC unit which is very reliable and easy to operate. Unit operates at 100°C and is more susceptible to corrosion.
- RO units are relatively new, operate at ambient temperature and has few mechanical parts for servicing. Average membrane life is 3 years.
- Consumption : estimate on 50-60 gallons per day per man (0.2 m^3)
  Storage : allow 10-15 days for offshore units
  10 days for onshore remote areas
- Potable water can be dosed with hypochlorite at 0.4-0.5 ppm to inhibit bacterial growth.

Waste water and sewage

- Before discharging to river, sea, or underground sewage + waste water must be treated to meet local health regulations prevalant to the area of siting.
- Limits are imposed on BOD (Biological Oxygen Demand), COD (Chemical Oxygen Demand), coliform bacteria count and TDS.

Example limits are :	bacteria	< 200 per 100 ml
	TDS	< 150 ppm
	BOD	< 100 mg/l
	Cl^- residual	> 0.5 mg/l < 1.0 mg/l

- Sewage is treated by physical attrition, airation and chlorine dosing to 30-40 ppm raw sewage.
- Provide 15-20 hr retention time for enzymic action to reducė BOD.
- Allow 30-50 gall/day per person (0.15 m^3) for sewage, shower, laundry and kitchen wastes. Use upper limit for hot unsociable climates.

6. FUELS

Diesel

- Used for emergency generators, pumps motors and air compressors, cranes, and alternative fuel for turbines.
- For emergency equipment provide individual day tanks sized on providing fuel for 24 hr operation.
- Main diesel tank (for feed to day tanks) should hold 10-12 days supply. This is dependant on location of plant and normal supply periods.
- Diesel should be filtered to ~ 5 Microns. Can be centrifuged to remove residual water and smaller particles. This is especially recommended offshore where longer storage times, supply boat debris, seawater contact and poor supply quality can lead to operation problems.
- For storage use atmospheric venting tanks with vacuum-PSV vent. Use crane pedestrals, platform legs or inter-deck space for offshore storage.

Gas

- Fuel gas is supplied as normal fuel to generators, turbines and any gas driven motors.
- Always pass FG through a scrubber before use. Filter gas supply to turbines to 10 microns (generally turbine manufacturer will state quality required and <u>may</u> include his own filters) - do not rely on this and provide separate treatment anyway.
- Maintain FG temperature at least 15°C above dew point. Minimum temperature of gas to be 5°C.
- Common supply pressures are 15-20 bar (some jet engines need 35 bar).
- Size fuel gas supply on maximum design duty of all users operating. Allow + 10% margin.
- FG used for flare purge and pilots, etc., does not need to be filtered to 10 microns - use gas straight off scrubber overheads.

Fig 1

Process water and deck drain water treatment

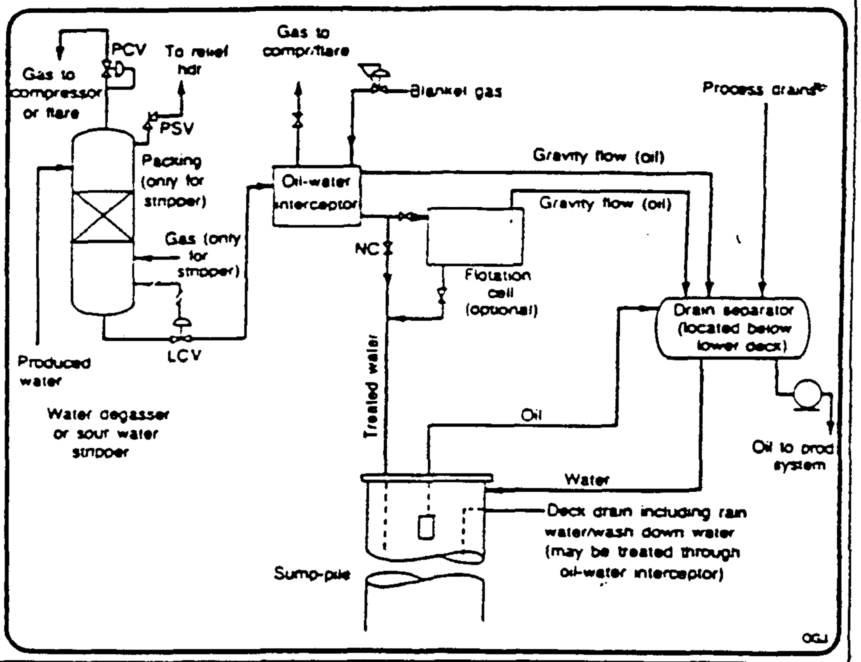


Fig. 2

Compressed Air System

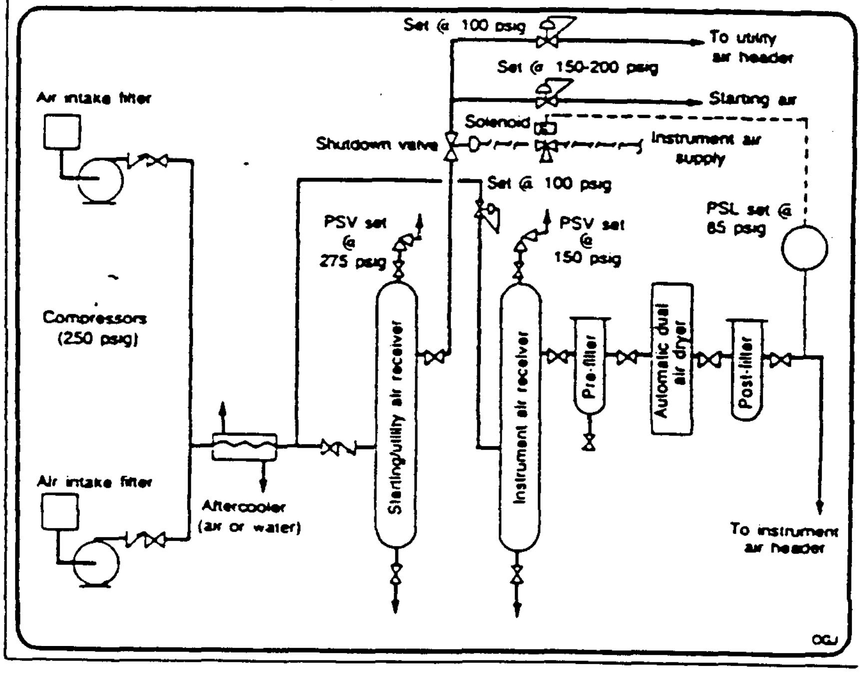
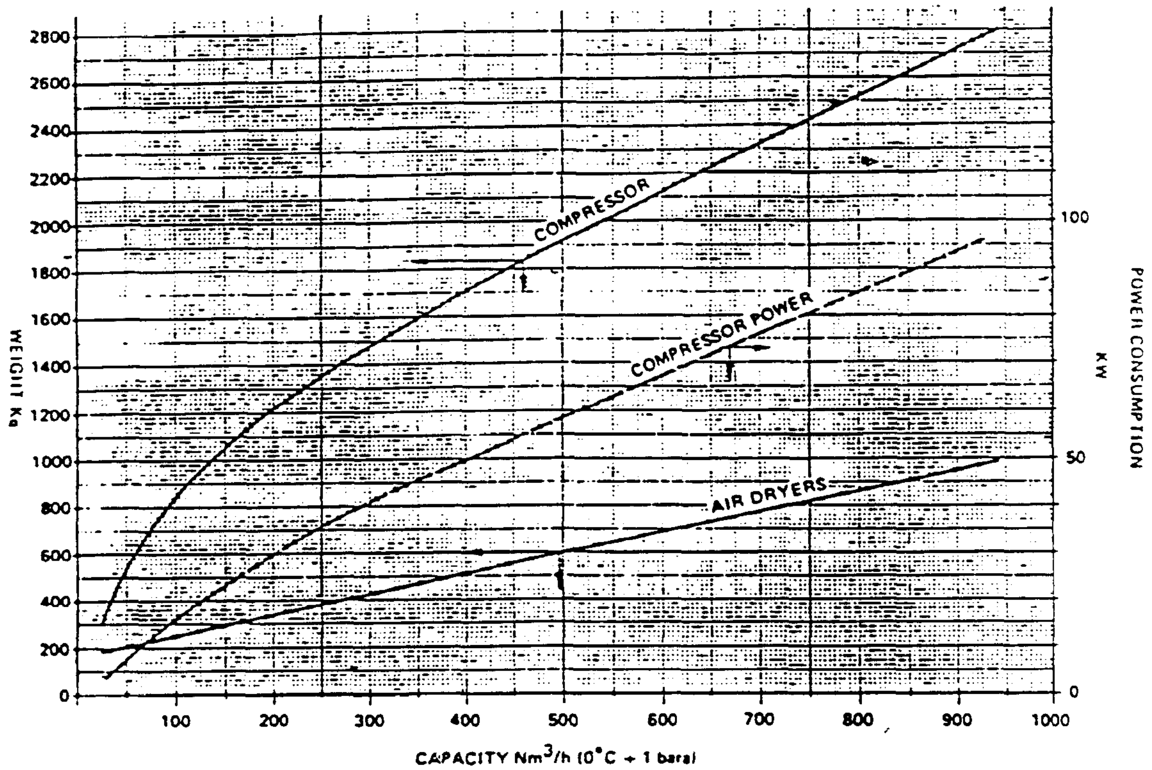
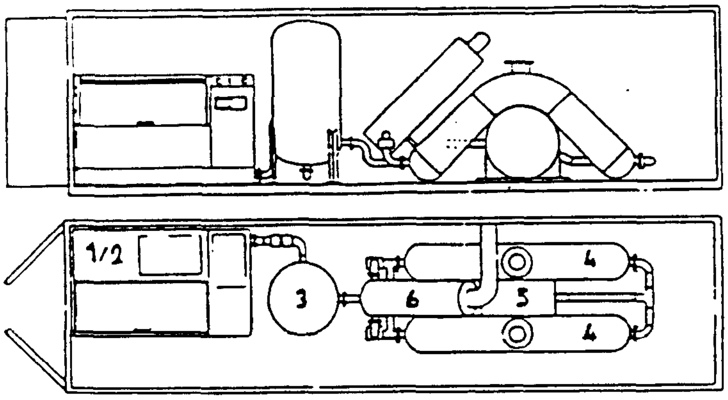


FIGURE 3
ESTIMATE OF WEIGHT AND POWER FOR INSTRUMENT AIR UNITS



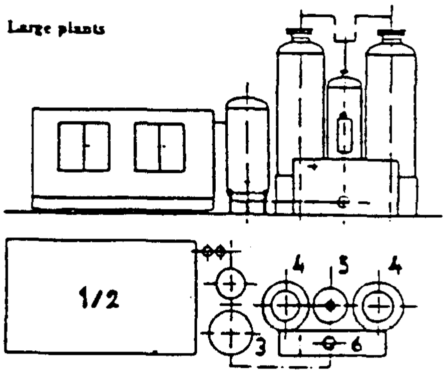
Small plants



Dimensions and weights

Type	Adsorber plant L x W x H (m)	Weight (kg)	Compressor with buffer (m)	Weight (kg)	Complete plant with container L x W x H (m)	Weight (kg)
DWN 20	3.0 x 1.1 x 1.1	750	1.9 x 1.7 x 1.8	700	6.0 x 2.4 x 2.6	3000
DWN 50	3.6 x 1.3 x 1.4	1200	2.0 x 2.0 x 2.2	900	6.0 x 2.4 x 2.6	3600
DWN 90	4.1 x 1.5 x 1.6	1800	4.0 x 1.5 x 2.2	1700	9.1 x 2.4 x 2.6	5900
DWN 180	4.9 x 1.7 x 2.0	2600	4.5 x 1.9 x 2.2	3500	12 x 2.4 x 2.6	9300

Large plants



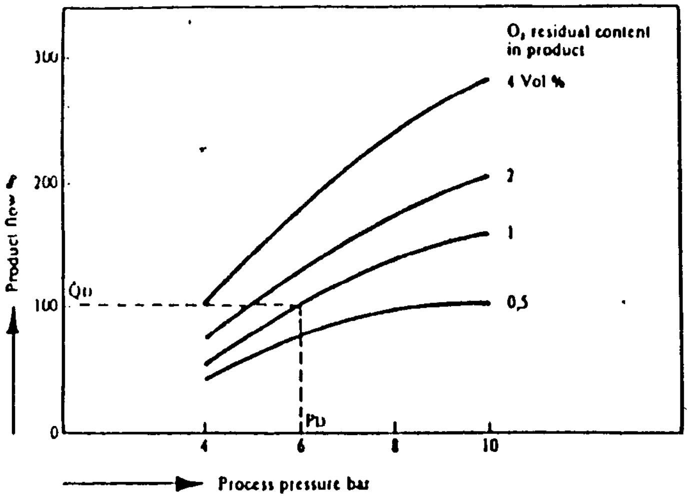
Dimensions and weights

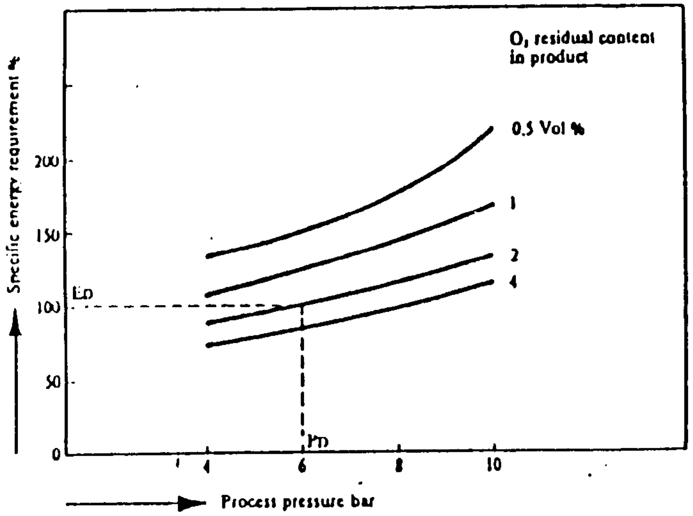
Type	Length (m)	Width (m)	Height (m)	Weight (kg)
DWN 400	8.3	2.6	4.3	9000
DWN 700	9.6	2.8	5.2	25000
DWN 1600	10.9	5.0	6.7	35000

Layout plan

1 Air compressor
2 Aftercooler
3 Buffer vessel
4 Adsorber
5 Product buffer vessel
6 Silencer

Principle relationship between process pressure product flow and spec. energy requirement of Linde DWN plants





Qd ... Product flow
Ed ... Spec. energy requirement
Pd ... Process pressure
} under design conditions

FIGURE 5 - INERT GAS GENERATORS

Adsorptive separation of air in carbon molecular sieves.

Technical data

Small plants: Ready-for-operation plants in one unit.
Fully function- and performance-tested.
No installation work.
No cooling water required.

Type designation	Max. O_2% content in product	N_2 product pressure 4.5 bar Product quantity m^3/h	Power consumption approx. kW	N_2 product pressure 6 bar Product quantity m^3/h	Power consumption approx. kW
DWN 20	0.5	13	8	16	11.5
	1	17	8.5	20	12
	2	20	9	30	14
DWN 50	0.5	29	15	35	21
	1	37	18	45	24
	2	50	20	65	29
DWN 90	0.5	50	25	60	36
	1	68	30	85	42
	2	90	33	115	48
DWN 180	0.5	100	48	120	70
	1	130	54	160	77
	2	180	60	230	86

Large plants: Ready-for-operation plants from function-tested units.
Minimal installation work.

Type designation	Max. O_2% contents in product	N_2 product pressure 4.5 bar Product quantity m^3/h	Power consumption approx. kW	N_2 product pressure 6 bar Product quantity m^3/h	Power consumption approx. kW	Cooling water m^3/h N_2 product pressure 4.5 bar	6 bar
DWN 400	0.5	240	110	295	170	12	16
	1	300	125	385	190	14	18
	2	370	135	500	200	15	21
DWN 700	0.5	410	185	500	260	17	25
	1	510	200	650	280	19	29
	2	620	215	850	300	22	33
DWN 1600	0.5	910	390	1100	530	40	53
	1	1110	410	1440	580	44	61
	2	1370	450	1900	670	47	71

The performance figures listed are measured at 20 °C air temperature, 70% air humidity and 1 bar pressure at the shaft. The product nitrogen has a dew point of -50 °C at atmospheric pressure and contains <20 ppm carbon dioxide as well as a residual portion of rare gases.

The high flexibility of DWN plants enables them to be economically operated at part load up to 60% of product quantity. In addition to the standard units listed, customized DWN plants can be supplied based on customer specifications for product quantities, supply pressures and product units.

Nitrogen 1 m^3/h = 1.25 kg/h   Air 1 m^3/h = 1.29 kg/h   1 bar = 1.02 kp/cm²

15. DATA

PERSONAL
NOTES:

Values of the gas constant R in PV = n RT

Basis of units listed below is 22.4140 liters at 0°C and 101.3250 kPa for the volume of 1 mole.* All other values calculated from conversion factors listed in tables.

n	Temperature	Pressure	Volume	R	n	Temperature	Energy	R
mol*	K	atm	liter	0.082 057 477	mol	K	calorie	1 985 9
mol	K	atm	cm³	82.057	mol	K	joule	8.314 5
mol	K	mm Hg	liter	62 364				
mol	K	bar	liter	0 083 145	lb mol	°R	Btu	1 985 9
mol	K	kg/cm²	liter	0 084 784	lb mol	°R	hp · h	0.000 780 48
mol	K	kPa	m³	0 008 314 5	lb mol	°R	kW · h	0 000 582 00
lb mol	°R	atm	ft³	0 730 24	lb mol	°R	ft · lb	1 545 3
lb mol	°R	in Hg	ft³	21 850				
lb mol	°R	mm Hg	ft³	554 98	kmol	K	joule	8 314 5
lb mol	°R	lb/in²	ft³	10.732				
lb mol	°R	lb/ft²	ft³	1 545.3				
lb mol	K	atm	ft³	1.314 4				
lb mol	K	mm Hg	ft³	998 97				
kmol	K	kPa	m³	8.314 5				
kmol	K	bar	m³	0.083 145				

*In SI, 1 mole is equivalent to the gm mole; the molecular mass of carbon-12, for example, is 12 g/mol

The relation of Degrees Baumé or A.P.I. to Specific Gravity is expressed by the following formulas:

For liquids lighter than water:

$$\text{Degrees Baumé} = \frac{140}{G} - 130,$$

$$G = \frac{140}{130 + \text{Degrees Baumé}}$$

$$\text{Degrees A.P.I.} = \frac{141.5}{G} - 131.5,$$

$$G = \frac{141.5}{131.5 + \text{Degrees A.P.I.}}$$

For liquids heavier than water:

$$\text{Degrees Baumé} = 145 - \frac{145}{G}.$$

$$G = \frac{145}{145 - \text{Degrees Baumé}}$$

G = Specific Gravity = ratio of the weight of a given volume of oil at 60° Fahrenheit to the weight of the same volume of water at 60° Fahrenheit.

To determine the resulting gravity by mixing oils of different gravities:

$$D = \frac{md_1 + nd_2}{m + n}$$

D = Density or Specific Gravity of mixture
m = Volume proportion of oil of d_1 density
n = Volume proportion of oil of d_2 density
d_1 = Specific Gravity or density of m oil
d_2 = Specific Gravity or density of n oil

MOLAR VOLUME		
P = 14696 psia, 1 atm, 1.0325 bara		
lb mole	0°C (60°F) 15.5°C	359.03 ft³ 379.49 ft³
Kg mole	0°C (60°F) 15.5°C	22.414 m³ 23.696 m³

°C = 5/9 (°F − 32)
°F = 9/5 (°C) + 32
K = °C + 273.15 = 5/9 R
°R = °F + 459.67 − 1.8 K

SI Prefixes and Multiplication Factors:

	Prefix	Symbol
10^{12}	tera	T
10^{9}	giga	G
10^{6}	mega	M
10^{3}	kilo	k
10^{2}	hecto	h
10^{1}	deca	da
10^{-1}	deci	d
10^{-2}	centi	c
10^{-3}	milli	m
10^{-6}	micro	µ
10^{-9}	nano	n
10^{-12}	pico	p
10^{-15}	femto	f
10^{-18}	atto	a

USEFUL NOTES

R_i	Individual gas constant	= R/mol. wt of gas		
N_a	Avogadro constant	= 6.02252×10^{23} molecules/mol		
g	Accelaration of gravity	= 9.81 m/s² 32.17 ft/s²		
V_s	Velocity of sound in dry air at 0 °C and 1 atm	= 1 089 ft/s = 331·92 m/s		

VOLUME

VOLUME m³	m³	cm³	l	in³	ft³	fl oz (U.S.)	gal (U.S.)	gal (U.K.)	bbl
re cube	1	1,000 000 E+06	1,000 000 E+03	6,102 376 E+04	3,531 466 E+01	3,381 402 E+04	2,641 720 E+02	2,199 692 E+02	6,289 811 E+00
metre cube	1,000 000 E-06	1	1,000 000 E-03	6,102 376 E-02	3,531 466 E-05	3,381 402 E-02	2,641 720 E-04	2,199 692 E-04	6,289 811 E-06
	1,000 000 E-03	1,000 000 E+03	1	6,102 376 E+01	3,531 466 E-02	3,381 402 E+01	2,641 720 E-01	2,199 692 E-01	6,289 811 E-03
c inch	1,638 706 E-05	1,638 706 E+01	1,638 706 E-02	1	5,787 035 E-04	5,541 124 E-01	4,329 003 E-03	3,604 648 E-03	1,030 715 E-04
c feet	2,831 685 E-02	2,831 685 E+04	2,831 685 E+01	1,728 000 E+03	1	9,575 066 E+02	7,480 520 E+00	6,228 833 E+00	1,781 076 E-01
ounce (U.S.)	2,957 353 E-05	2,957 353 E+01	2,957 353 E-02	1,804 688 E+00	1,044 379 E-03	1	7,812 500 E-03	6,505 264 E-03	1,860 119 E-04
n (U.S. liquid)	3,785 412 E-03	3,785 412 E+03	3,785 412 E+00	2,310 000 E+02	1,336 805 E-01	1,280 000 E+02	1	8,326 739 E-01	2,380 952 E-02
rial gallon liquid)	4,546 092 E-03	4,546 092 E+03	4,546 092 E+00	2,774 196 E+02	1,605 437 E-01	1,537 217 E+02	1,200 950 E+00	1	2,859 406 E-02
el (oil)	1,589 873 E-01	1,589 873 E+05	1,589 873 E+02	9,702 000 E+03	5,614 583 E+00	5,376 000 E+03	4,200 000 E+01	3,497 230 E+01	1

MASS

MASSE kg	kg	g	t	lb	oz	ton (short)	ton (long)
ogramme	1	1,000 000 E+03	1,000 000 E−03	2,204 622 E+00	3,527 397 E+01	1,102 311 E−03	9,842 064 E−04
mme	1,000 000 E−03	1	1,000 000 E−06	2,204 622 E−03	3,527 397 E−02	1,102 311 E−06	9,842 064 E−07
ne métrique	1 000 000 E+03	1,000 000 E+06	1	2,204 622 E+03	3,527 397 E+04	1,102 311 E+00	9,842 064 E−01
nd (avoir)	4,535 924 E−01	4,535 924 E+02	4,535 924 E−04	1	1,600 000 E+01	5,000 000 E−04	4,464 286 E−04
nce (avoir)	2,834 952 E−02	2,834 952 E+01	2,834 952 E−05	6,250 000 E−02	1	3,125 000 E−05	2,790 178 E−05
rt ton (U.S.) 00 lbm)	9,071 847 E+02	9,071 847 E+05	9,071 847 E−01	2,000 000 E+03	3,200 000 E+04	1	8,928 570 E−01
g ton (G.B.) 40 lbm)	1,016 047 E+03	1,016 047 E+06	1,016 047 E+00	2,240 000 E+03	3,584 000 E+04	1,120 000 E+00	1

AREA

AIRE m^2	m^2	cm^2	a	acre	in^2	ft^2
metre carré	1	1,000 000 E -04	1,000 000 E -02	2,471 054 E -04	1,550 003 E -03	1,076 391 E -01
centimetre carré	1,000 000 E -04	1	1,000 000 E -06	2,471 054 E -08	1,550 003 E -01	1,076 391 E -03
are	1,000 000 E +02	1,000 000 E +06	1	2,471 054 E -02	1,550 003 E +05	1,076 391 E +03
acre	4,046 856 E +03	4,046 856 E +07	4,046 856 E +01	1	6,272 639 E +06	4,356 000 E +04
square inch	6,451 600 E -04	6,451 600 E +00	6,451 600 E -06	1,594 225 E -07	1	6,944 444 E -03
square foot	9,290 304 E -02	9,290 304 E +02	9,290 304 E -04	2,295 684 E -05	1,440 000 E +02	1

LENGTH

LONGUEUR m	m	cm	µ	Å	in (″)	ft (′)	mile
metre	1	1,000 000 E +02	1,000 000 E +06	1,000 000 E +10	3,937 008 E +01	3,280 840 E +00	6,213 712 E -04
centimetre	1,000 000 E -02	1	1,000 000 E +04	1,000 000 E +08	3,937 008 E -01	3,280 840 E -02	6,213 712 E -06
micron	1,000 000 E -06	1,000 000 E -04	1	1,000 000 E +04	3,937 008 E -05	3,280 840 E -06	6,213 712 E -10
angstrom	1,000 000 E -10	1,000 000 E -08	1,000 000 E -04	1	3,937 008 E -09	3,280 840 E -10	6,213 712 E -14
inch (pouce)	2,540 000 E -02	2,540 000 E +00	2,540 000 E +04	2,540 000 E +08	1	8,333 333 E -02	1,578 283 E -05
foot (pied)	3,048 000 E -01	3,048 000 E +01	3,048 000 E +05	3,048 000 E +09	1,200 000 E +01	1	1,893 939 E -04
mile (U.S. statute)	1,609 344 E +03	1,609 344 E +05	1,609 344 E +09	1,609 344 E +13	6,336 000 E +04	5,280 000 E +03	1

VELOCITY

VITESSE LINEAIRE $m\,s^{-1}$	m/s	km/h	in/s	ft/s	mile/h
metre par seconde	1	3,600 000 E +00	3,937 008 E +01	3,280 840 E +00	2,236 936 E +00
kilometre par heure	2,777 778 E -01	1	1,093 613 E +01	9,113 445 E -01	6,213 712 E -01
inch per second	2,540 000 E -02	9,144 000 E -02	1	8,333 333 E -02	5,681 818 E -02
foot per second	3,048 000 E -01	1,097 280 E +00	1,200 000 E +01	1	6,818 187 E -01
mile per hour (U.S. statute)	4,470 400 E -01	1,609 344 E +00	1,760 000 E +01	1,466 667 E +00	1

VOLUMETRIC FLOW

DEBIT VOLUME $m^3.s^{-1}$	m^3/s	m^3/h	l/s	m^3/mn	gpm	ft^3/s	ft^3/h	bbl/d
metre cube par seconde	1	3.600 000 E+03	1 000 000 E+03	6.000 000 E+01	1,585 032 E+04	3.531 466 E+01	1,271 328 E−05	5 434 396 E+05
metre cube par heure	2,777 778 E−04	1	2,777 778 E−01	1,666 667 E−02	4.407 868 E+00	9.809 629 E−03	3.531 466 E+01	1,509 554 E+02
litre par seconde	1,000 000 E−03	3.600 000 E+00	1	6,000 000 E−02	1,585 032 E+01	3.531 466 E−02	1,271 328 E+02	5.434 396 E+02
metre cube par minute	1,666 667 E−02	6,000 000 E+01	1,666 667 E+01	1	2,641 721 E+02	5,885 778 E−01	2,118 880 E+03	9,057 328 E+03
gallon per minute (U.S. liquid)	6,308 020 E−05	2,271 247 E−01	6,308 020 E−02	3,785 411 E−03	1	2,228 009 E−03	8,020 832 E+00	3,428 571 E+01
cubic foot per second (cusec)	2,831 685 E−02	1,019 407 E+02	2,831 685 E+01	1,699 011 E+00	4,488 312 E+02	1	3,600 000 E+03	1,538 850 E+04
cubic foot per hour	7,866 782 E−06	2,831 685 E−02	7,866 782 E−03	4,719 474 E−04	1,246 753 E−01	2,777 778 E−04	1	4,274 583 E+00
barrel per day	1,840 131 E−06	6,624 471 E−03	1,840 131 E−03	1,104 078 E−04	2,916 667 E−02	6,498 361 E−05	2,339 410 E−01	1

DENSITY

MASSE VOLUMIQUE $m^{-3}.kg$	kg/m^3	g/cm^3	lb/in^3	lb/ft^3	lb/gal (U.S.)
kilogramme par mètre cube	1	1,000 000 E−03	3,612 730 E−05	6,242 797 E−02	8,345 406 E−03
gramme par centimètre cube	1,000 000 E+03	1	3,612 730 E−02	6,242 797 E+01	8,345 406 E+00
pound per cubic inch	2,767 990 E+04	2,767 990 E+01	1	1,728 000 E+03	2,310 000 E+02
pound per cubic foot	1,601 846 E+01	1,601 846 E−02	5,787 037 E−04	1	1,336 806 E−01
pound per gallon (U.S.)	1,198 264 E+02	1,198 264 E−01	4,329 004 E−03	7,480 519 E+00	1

PRESSURE

TOTAL

PRESSION/ CONTRAINTE $m^{-1}.kg.s^{-2}$	Pa	bar	dyn/cm²	atm	kgf/cm²	mmHg (torr)	mmH₂0	psi	psf
pascal	1	1,000 000 E 05	1 000 000 E +01	9 869 233 E 06	1 019 716 E 05	7 500 615 E 03	1 019 716 E -01	1 450 377 E -04	2 088 543 E -02
bar	1,000 000 E -05	1	1 000 000 E +06	9 869 233 E -01	1 019 716 E +00	7 500 615 E +02	1 019 716 E +04	1 450 377 E -01	2 088 543 E +03
dyne par centimetre carre	1,000 000 E -01	1,000 000 E -06	1	9 869 233 E -07	1 019 716 E 06	7 500 615 E -04	1 019 716 E -02	1 450 377 E -05	2 088 543 E -03
atmosphère normale	1,013 250 E +05	1,013 250 E +00	1 013 250 E +06	1	1 033 227 E +00	7,600 000 E +02	1,033 227 E +04	1 469 595 E +01	2,11[illegible] E -03
kilogramme-force par centimetre carré	9,806 650 E +04	9,806 650 E -01	9,806 650 E +05	9 678 411 E -01	1	7,355 591 E +02	1 000 000 E +04	1 422 334 E +01	2,048 161 E +03
millimetre mercure à 0°C (torr)	1,333 224 E +02	1,333 224 E -03	1,333 224 E +03	1,315 790 E -03	1,359 510 E -03	1	1,359 510 E +01	1,933 678 E -02	2,784 496 E +00
millimetre eau à 4°C	9 806 650 E +00	9,806 650 E -05	9,805 650 E +01	9,678 411 E -05	1,000 000 E -04	7,355 591 E -02	1	1,422 334 E -03	2,048 161 E -01
pound-force per square inch	6,894 757 E +03	6,894 757 E -02	6,894 757 E +04	6,804 596 E -02	7,030 695 E -02	5,171 492 E +01	7,030 695 E +02	1	1,440 000 E +02
pound-force per square feet	4,788 026 E +01	4,788 026 E -04	4 788 026 E +02	4,725 414 E -04	4 882 428 E -04	3,591 314 E -01	4,882 428 E +00	6,944 445 E -03	1

VISCOSITY (Dynamique)

VISCOSITE DYNAMIQUE $m^{-1}.kg.s^{-1}$	Pa . s	cP	kg/m.h	lb/ft.s
pascal seconde	1	1,000 000 E +03	3,600 000 E +03	6,719 689 E -01
centipoise	1,000 000 E -03	1	3,600 000 E +00	6,719 689 E -04
kilogramme par metre-heure	2,777 778 E -04	2,777 778 E -01	1	1 866 581 E -04
pound per feet-second	1,488 164 E +00	1,488 164 E +03	5,357 390 E +03	1

VISCOSITY (Kinematic)

VISCOSITE CINÉMATIQUE/ DIFFUSIVITE $m^2 s^{-1}$	m^2/s	cSt	ft^2/s	ft^2/h
mètre carré par seconde	1	1,000 000 E +06	1,076 391 E +01	3,875 008 E +04
centistokes	1,000 000 E −06	1	1,076 391 E −05	3,875 008 E −02
square foot per second	9,290 304 E −02	9,290 304 E +04	1	3,600 000 E +03
square foot per hour	2,580 640 E −05	2,580 640 E +01	2,777 778 E −04	1

ENERGY/WORK/HEAT

ENERGIE-TRAVAIL-QUANTITE DE CHALEUR $m^2\ kg.s^{-2}$	J_{abs}	erg	kWh_{abs}	cal_{th}	$cal_{I.T.}$	$kcal_{I.T.}$	th	ft.lbf	$Btu_{I.T.}$	hph
joule (absolu)	1	1,000 000 E +07	2,777 778 E −07	2,390 057 E −01	2,388 459 E −01	2,388 459 E −04	2,389 201 E −07	7,375 621 E −01	9 478 170 E −04	3,725 061 E −07
erg	1 000 000 E −07	1	2,777 778 E −14	2,390 057 E −08	2,388 459 E −08	2,388 459 E −11	2,389 201 E −14	7,375 621 E −08	9,478 170 E −11	3,725 061 E −14
watt-heure (abs)	3 600 000 E +06	3 600 000 E +13	1	8,604 207 E +05	8,598 452 E +05	8,598 452 E +02	8,601 173 E −01	2,655 224 E +06	3 412 141 E +03	1,341 022 E +00
calorie thermochimique	4 184 000 E +00	4 184 000 E +07	1,622 222 E −06	1	9,993 312 E −01	9,993 312 E −04	9,996 416 E −07	3,085 960 E +00	3,965 666 E −03	1,558 565 E −06
calorie I.T.	4 186 800 E +00	4 186 800 E +07	1,163 000 E −06	1 000 669 E +00	1	1,000 000 E −03	1,000 311 E −06	3,088 075 E +00	3,968 320 E −03	1 559 608 E −06
kilocalorie I.T.	4 186 800 E +03	4 186 800 E +10	1 163 000 E −03	1,000 669 E +03	1,000 000 E +03	1	1,000 311 E −03	3,088 075 E +03	3,968 320 E +00	1,559 608 E −03
thermie (AFNOR)	4 185 500 E +06	4 185 500 E +13	1 162 639 E +00	1,000 359 E +06	9,996 895 E +05	9 996 895 E +02	1	3,087 066 E +06	3,967 088 E +03	1 559 124 E +00
foot-pound force	1 355 818 E +00	1 355 818 E +07	3 766 161 E −07	3 240 483 E −01	3,238 316 E −01	3 238 316 E −04	3 239 321 E −07	1	1,285 067 E −03	5 050 504 E −07
British thermal unit (I T)	1 055 056 E +03	1 055 056 E +10	2 930 711 E −04	2 521 644 E +02	2 519 958 E +02	2 519 958 E −01	2,520 741 E −04	7 781 693 E +02	1	3 930 148 E −04
horsepower hour	2 684 520 E +06	2 684 520 E +13	7 457 000 E −01	6 416 157 E +05	6 411 866 E +05	6 411 866 E +02	6 413 857 E −01	1 980 000 E +06	2 544 434 E +03	1

SPECIFIC HEAT CAPACITY

CAPACITE THERMIQUE MASSIQUE/ CHALEUR SPECIFIQUE $m^2\,s^{-2}\,K^{-1}$	J/kg K	kcal/kg.°C	Btu/lb.°F
joule par kilogramme-kelvin	1	2,390 057 E -04	2 388 459 E -04
kilocalorie par kilogramme-degré Celsius	4 184 000 E +03	1	9 993 312 E -01
British thermal unit per pound-degree Fahrenheit	4 186 800 E +03	1 000 669 E +00	1

THERMAL CONDUCTIVITY

CONDUCTIVITE THERMIQUE $m.kg.s^{-3}\,K^{-1}$	W/m.K	kcal/h.m.°C	Btu/h.ft.°F
watt par mètre-kelvin	1	8,604 208 E -01	5,777 792 E -01
kilocalorie par heure-mètre-degré Celsius	1,162 222 E +00	1	6,715 193 E -01
British thermal unit per hour-foot-degree Fahrenheit	1,730 735 E +00	1,489 160 E +00	1

HEAT TRANSFER COEFFICIENT

COEFFICIENT DE TRANSMISSION THERMIQUE $kg.s^{-3}\,K^{-1}$	W/m².K	kcal/h.m².°C	Btu/h.ft².°F
watt par mètre carré-kelvin	1	8,604 208 E -01	1,761 102 E -01
kilocalorie par heure-mètre carré-degré Celsius	1,162 222 E +00	1	2,046 791 E -01
British thermal unit per hour-square foot-degree Fahrenheit	5,678 263 E +00	4,885 696 E +00	1

FORCE

FORCE $m.kg.s^{-2}$	N	dyn	kgf	lbf	pdl
newton	1	1,000 000 E +05	1,019 716 E -01	2,248 089 E -01	7,233 011 E +00
dyne	1 000 000 E -05	1	1 019 716 E -06	2,248 089 E -06	7,233 011 E -05
kilogramme-force	9 806 650 E +00	9,806 650 E +05	1	2,204 622 E +00	7 093 161 E +01
pound force	4 448 222 E +00	4,448 222 E +05	4,535 924 E -01	1	3,217 404 E +01
poundal	1,382 550 E -01	1,382 550 E +04	1,409 809 E -02	3,108 096 E -02	1

HEAT CAPACITY/ENTROPY

ENTROPIE/ CAPACITE THERMIQUE $m^2 \cdot kg \cdot s^{-2} \cdot K^{-1}$	J/K	kcal/K	Btu/°F
joule par kelvin	1	2,390 057 E−04	1,706 071 E−03
kilocalorie par degré kelvin	4,184 000 E+03	1	7,138 200 E−00
British thermal unit per degree Fahrenheit	5,861 422 E+02	1,400 913 E−01	1

POWER/HEAT FLOW RATE

PUISSANCE- FLUX ENERGETIQUE $m^2 \cdot kg \cdot s^{-3}$	W	erg/s	cal/s	kcal/h	ch	hp	ft.lbf/s	Btu/h	t of refrig.
watt (joule par seconde)	1	1,000 000 E+07	2 390 057 E 01	8 604 208 E 01	1 359 621 E 03	1 341 022 E 03	7 375 621 E−01	3 412 141 E+00	2,843 451 E−04
erg par seconde	1 000 000 E−07	1	2 390 057 E 08	8 604 208 E 08	1 359 621 E 10	1 341 022 E 10	7 375 621 E 08	3 412 141 E−07	2 843 451 E−11
par seconde	4 184 000 E+00	4 184 000 E+07	1	3 600 000 E+00	5 688 655 E 03	5 610 835 E 03	3 085 960 E+00	1,427 640 E+01	1 189 700 E−03
kilocalorie par heure	1 162 222 E+00	1 162 222 E+07	2,777 778 E−01	1	1 580 182 E 03	1 558 565 E 03	8 572 109 E 01	3 965 666 E+00	3,304 722 E−04
cheval vapeur metrique	7 354 990 E+02	7 354 990 E+09	1 757 885 E+02	6 328 386 E+02	1	9 863 207 E 01	5 424 767 E+02	2 509 627 E+03	2 091 356 E−01
horsepower (UK)	7 457 000 E+02	7 457 000 E+09	1 782 266 E+02	6 416 158 E+02	1 013 869 E+00	1	5 500 000 E+02	2,544 434 E+03	2 120 362 E−01
foot-pound force per second	1 355 818 E+00	1,355 818 E+07	3,240 483 E 01	1 166 574 E+00	1 843 399 E 03	1 818 182 E 03	1	4 626 243 E+00	3 855 202 E−04
British thermal unit per hour	2 930 711 E−01	2,930 711 E+06	7 004 567 E−02	2 521 544 E−01	3 984 657 E−04	3 930 148 E 04	2 161 581 E−01	1	8 333 334 E−05
ton (refrigeration)	3 516 853 E+03	3 516 853 E+10	8 405 480 E+02	3 025 974 E+03	4 781 588 E+00	4 716 177 E+00	2 593 898 E+03	1 2000 000 E+04	1

N.B. – joule absolu
– calorie ou kilocalorie thermochimique
– Btu I.T

PSEUDO-CRITICALS AND OIL PROPERTIES

. Definitions :

True vapour pressure : - actual vapour pressure of a crude oil at the actual temperature of the fluid.

Reid vapour pressure : - reference vapour pressure of an oil at a controlled temperature of 100 °F (used as a basis for product specification).

Molal average boiling point : - equal to the sum of the mole fraction of each component x its atmospheric boiling point °R.

Volume average boiling point : VABP : - average temperature at which the ASTM 10 %, 30 %, 50 %, 70 % and 90 % volumes boil.

$$VABP = \frac{T10\,\% + T30\,\% + T50\,\% + T70\,\% + T90\,\%}{5}$$

Mean average boiling point : MABP : - the slope of the ASTM distillation curve is used to correct the VABP to give MABP. See Fig. 7

Cubic average boiling point : CABP : - another corrected form of VABP.

UOP K or WATSON CHARACTERISATION FACTOR

$$K = \frac{CABP^{1/3}}{sg}$$

sg at 60/60
CABP in °R

This issued as a characterisation factor when defining crude oils. It is required for various other data evaluations.

PHYSICAL CONSTANTS OF HYDROCARBONS(27)

	See Note No. →		1	2.		3			
							Critical constants		
No.	Compound	Formula	Molecular mass	Boiling point, °C 101.3250 kPa (abs)	Vapor pressure, kPa (abs) 40 °C	Freezing point, °C 101.3250 kPa (abs)	Pressure, kPa (abs)	Temperature, K	Volume, m³/kg
1	Methane	CH_4	16.043	−161.52(28)	(35 000.)	-182.47^d	4 604	190.55	0.006 17
2	Ethane	C_2H_6	30.070	−88.58	(6 000.)	-182.80^d	4 880.	305.43	0.004 92
3	Propane	C_3H_8	44.097	−42.07	1 341.	-187.68^d	4 249.	369.82	0.004 60
4	*n*-Butane	C_4H_{10}	58.124	−0.49	377.	−138.36	3 797.	425.16	0.004 39
5	Isobutane	C_4H_{10}	58.124	−11.81	528.	−159.60	3 648.	408.13	0.004 52
6	*n*-Pentane	C_5H_{12}	72.151	36.06	115.66	−129.73	3 369.	469.6	0.004 21
7	Isopentane	C_5H_{12}	72.151	27.84	151.3	−159.90	3 381.	460.39	0.004 24
	Neopentane	C_5H_{12}	72.151	9.50	269.	−16.55	3 199.	433.75	0.004 20
	n-Hexane	C_6H_{14}	86.178	68.74	37.28	−95.32	3 012.	507.4	0.004 29
	2-Methylpentane	C_6H_{14}	86.178	60.26	50.68	−153.66	3 010.	497.45	0.004 26
11	3-Methylpentane	C_6H_{14}	86.178	63.27	45.73	—	3 124.	504.4	0.004 26
12	Neohexane	C_6H_{14}	86.178	49.73	73.41	−99.870	3 081.	488.73	0.004 17
13	2,3-Dimethylbutane	C_6H_{14}	86.178	57.98	55.34	−128.54	3 127.	499.93	0.004 15
14	*n*-Heptane	C_7H_{16}	100.205	98.42	12.34	−90.582	2 736.	540.2	0.004 31
15	2-Methylhexane	C_7H_{16}	100.205	90.05	17.22	−118.27	2 734.	530.31	0.004 20
16	3-Methylhexane	C_7H_{16}	100.205	91.85	16.16	—	2 814.	535.19	0.004 03
17	3-Ethylpentane	C_7H_{16}	100.205	93.48	15.27	−118.60	2 891.	540.57	0.004 15
18	2,2-Dimethylpentane	C_7H_{16}	100.205	79.19	26.32	−123.81	2 773.	520.44	0.004 15
19	2,4-Dimethylpentane	C_7H_{16}	100.205	80.49	24.84	−119.24	2 737.	519.73	0.004 17
20	3,3-Dimethylpentane	C_7H_{16}	100.205	86.06	20.93	−134.46	2 945.	536.34	0.004 13
21	Triptane	C_7H_{16}	100.205	80.88	25.40	−24.91	2 954.	531.11	0.003 97
22	*n*-Octane	C_8H_{18}	114.232	125.67	4.143	−56.76	2 486.	568.76	0.004 31
23	Diisobutyl	C_8H_{18}	114.232	109.11	8.417	−91.200	2 486.	549.99	0.004 22
24	Isooctane	C_8H_{18}	114.232	99.24	12.96	−107.38	2 568.	543.89	0.004 10
25	*n*-Nonane	C_9H_{20}	128.259	150.82	1.40	−53.49	2 288.	594.56	0.004 27
26	*n*-Decane	$C_{10}H_{22}$	142.286	174.16	0.4732	−29.64	2 099.	617.4	0.004 24
27	Cyclopentane	C_5H_{10}	70.135	49.25	73.97	−93.866	4 502.	511.6	0.003 71
28	Methylcyclopentane	C_6H_{12}	84.162	71.81	33.85	−142.46	3 785.	532.73	0.003 79
29	Cyclohexane	C_6H_{12}	84.162	80.73	24.63	6.554	4 074.	553.5	0.003 68
30	Methylcyclohexane	C_7H_{14}	98.189	100.93	12.213	−126.59	3 472.	572.12	0.003 75
31	Ethene (Ethylene)	C_2H_4	28.054	−103.77(29)	—	-169.15^d	5 041.	282.35	0.004 67
32	Propene (Propylene)	C_3H_6	42.081	−47.72	1 596.	-185.25^d	4 600.	364.85	0.004 30
33	1-Butene (Butylene)	C_4H_8	56.108	−6.23	451.9	-185.35^d	4 023.	419.53	0.004 28
34	*cis*-2-Butene	C_4H_8	56.108	3.72	337.6	−138.91	4 220.	435.58	0.004 17
35	*trans*-2-Butene	C_4H_8	56.108	0.88	365.8	−105.55	4 047.	428.63	0.004 24
	Isobutene	C_4H_8	56.108	−6.91	452.3	−140.35	3 999.	417.90	0.004 26
	1-Pentene	C_5H_{10}	70.135	29.96	141.65	−165.22	3 529.	464.78	0.004 22
38	1,2-Butadiene	C_4H_6	54.092	10.85	269.	−136.19	(4 502.)	(444.)	(0.004 05)
39	1,3-Butadiene	C_4H_6	54.092	−4.41	434.	−108.91	4 330.	425.	0.004 09
40	Isoprene	C_5H_8	68.119	34.07	123.77	−145.95	(3 850.)	(484.)	(0.004 06)
41	Acetylene	C_2H_2	26.038	-84.88^e	—	-80.8^d	6 139.	308.33	0.004 34
42	Benzene	C_6H_6	78.114	80.09	24.38	5.533	4 898.	562.16	0.003 28
43	Toluene	C_7H_8	92.141	110.63	7.895	−94.991	4 106.	591.80	0.003 43
44	Ethylbenzene	C_8H_{10}	106.168	136.20	2.87	−94.975	3 609.	617.20	0.003 53
45	*o*-Xylene	C_8H_{10}	106.168	144.43	2.05	−25.18	3 734.	630.33	0.003 48
46	*m*-Xylene	C_8H_{10}	106.168	139.12	2.53	−47.87	3 536.	617.05	0.003 54
47	*p*-Xylene	C_8H_{10}	106.168	138.36	2.65	13.26	3 511.	616.23	0.003 56
48	Styrene	C_8H_8	104.152	145.14	1.85	−30.61	3 999.	647.6	0.003 38
49	Isopropylbenzene	C_9H_{12}	120.195	152.41	1.47	−96.035	3 209.	631.1	0.003 57
50	Methyl alcohol	CH_4O	32.042	64.54	35.43	−97.68	8 096.	512.64	0.003 68
51	Ethyl alcohol	C_2H_6O	46.069	78.29	17.70	−114.1	6 383.	513.92	0.003 62
52	Carbon monoxide	CO	28.010	−191.49	—	-205.0^d	3 499.(33)	132.92(33)	0.003 32(33)
53	Carbon dioxide	CO_2	44.010	-78.51^e	—	-56.57^d	7 382.(33)	304.19(33)	0.002 14(33)
54	Hydrogen sulfide	H_2S	34.076	−60.31	2 881.	-85.53^d	9 005.	373.5	0.002 87
55	Sulfur dioxide	SO_2	64.059	−10.02	630.8	-75.48^d	7 894.	430.8	0.001 90
56	Ammonia	NH_3	17.031	−33.33(30)	1 513.	-77.74^d	11 280.	405.6	0.004 25
57	Air	$N_2 + O_2$	28.964	−194.2(2)	—	—	3 771.(2)	132.4(2)	0.003 23(3)
58	Hydrogen	H_2	2.016	-252.87^v	—	-259.2^d	1 297	33.2	0.032 24
59	Oxygen	O_2	31.999	-182.962^v	—	-218.8^d	5 081.	154.7(33)	0.002 29
60	Nitrogen	N_2	28.013	−195.80(31)	—	-210.0^d	3 399.	126.1	0.003 22
61	Chlorine	Cl_2	70.906	−34.03	1 134	-101.0^d	7 711.	417	0.001 75
62	Water	H_2O	18.015	100.00^v	7.377	0.00	22 118.	647.3	0.003 18
63	Helium	He	4.003	−268.93(32)	—	—	227.5(32)	5.2(32)	0.014 36(32)
64	Hydrogen chloride	HCl	36.461	−85.00	6 304	-114.18^d	8 309.	324.7	0.002 22

PHYSICAL CONSTANTS OF HYDROCARBONS(27)

4				5.	6.	7	8			9	
Density of liquid 101 3250 kPa (abs), 15 °C							Ideal gas * 101 3250 kPa (abs) 15 °C			Specific heat capacity 101 3250 kPa (abs) 15 °C C_p kJ/(kg · °C)	
Relative density 15 °C/15 °C ^{a,b}	kg/m³ ^{a,d} (mass in vacuum)	kg/m³ ^{a,e,c} (Apparent mass in air)	m³/kmol	Temperature coefficient of density, at 15 °C ^{a} −1/°C	Pitzer acentric factor	Compressibility factor of real gas, Z 101 3250 kPa (abs), 15 °C	Relative density Air = 1	Specific volume m³/kg	Volume ratio gas/(liquid in vacuum)	Ideal gas	Liquid
(0.3)^{i}	(300.)^{i}	(300.)^{i}	(0.051)^{i}	—	0.0126	0.9981	0.5539	1.474	(442.)^{i}	2.204	—
0.3581^{h}	357.8^{h,x}	356.6^{h}	0.084 04^{h}	—	0.0978	0.9915	1.0382	0.7863	281.3^{h}	1.706	3.807
0.5083^{h}	507.8^{h,x}	506.7^{h}	0.086 84^{h}	0.002 74^{h}	0.1541	0.9810	1.5225	0.5362	272.3^{h}	1.625	2.476
0.5847^{h}	584.2^{h}	583.1^{h}	0.099 49^{h}	0.002 11^{h}	0.2015	0.9641	2.0068	0.4068	237.6^{h}	1.652	2.366(41)
0.5637^{h}	563.2^{h}	562.1^{h}	0.103 2^{h}	0.002 14^{h}	0.1840	0.9665	2.0068	0.4068	229.1^{h}	1.616	2.366(41)
0.6316	631.0	629.9	0.114 3	0.001 57	0.2524	0.942†	2.4911	0.3277	206.8	1.622	2.292(41)
0.6250	624.4	623.3	0.115 6	0.001 62	0.2286	0.948†	2.4911	0.3277	204.6	1.600	2.239
0.5972^{h}	596.7^{h}	595.6^{h}	0.120 9^{h}	0.001 87^{h}	0.1967	0.9538	2.4911	0.3277	195.5^{h}	1.624	2.317
0.6644	663.8	662.7	0.129 8	0.001 35	0.2998	0.910†	2.9753	0.2744	182.1	1.613	2.231
0.6583	657.7	656.6	0.131 0	0.001 40	0.2784	—	2.9753	0.2744	180.5	1.602	2.205
0.6694	668.8	667.7	0.128 9	0.001 35	0.2741	—	2.9753	0.2744	183.5	1.578	2.170
0.6545	653.9	652.8	0.131 8	0.001 40	0.2333	—	2.9753	0.2744	179.4	1.593	2.148
0.6668	666.2	665.1	0.129 4	0.001 35	0.2475	—	2.9753	0.2744	182.8	1.566	2.146
0.6886	688.0	686.9	0.145 6	0.001 24	0.3494	0.852†	3.4596	0.2360	162.4	1.606	2.209
0.6835	682.8	681.7	0.146 8	0.001 22	0.3303	—	3.4596	0.2360	161.1	1.595	2.183
0.6921	691.5	690.4	0.144 9	0.001 24	0.3239	—	3.4596	0.2360	163.2	1.584	2.137
0.7032	702.6	701.5	0.142 6	0.001 26	0.3107	—	3.4596	0.2360	165.8	1.613	2.150
0.6787	678.0	676.9	0.147 8	0.001 30	0.2876	—	3.4596	0.2360	160.0	1.613	2.161
0.6777	677.1	676.0	0.148 0	0.001 30	0.3031	—	3.4596	0.2360	159.8	1.651	2.193
0.6980	697.4	696.3	0.143 7	0.001 17	0.2681	—	3.4596	0.2360	164.6	1.603	2.099
0.6950	694.4	693.3	0.144 3	0.001 24	0.2509	—	3.4596	0.2360	163.9	1.578	2.088
0.7073	706.7	705.6	0.161 6	0.001 12	0.3981	0.783†	3.9439	0.2070	146.3	1.601	2.191
0.6984	697.7	696.6	0.163 7	0.001 17	0.3564	—	3.9439	0.2070	144.4	1.573	2.138
0.6966	696.0	694.9	0.164 1	0.001 17	0.3041	—	3.9439	0.2070	144.1	1.599	2.049
0.7224	721.7	720.6	0.177 7	0.001 13	0.4452	—	4.4282	0.1843	133.0	1.598	2.184
0.7346	733.9	732.8	0.193 9	0.000 99	0.4904	—	4.9125	0.1662	122.0	1.595	2.179
0.7508	750.2	749.1	0.093 49	0.001 26	0.1945	0.949†	2.4215	0.3371	252.9	1.133	1.763
0.7541	753.4	752.3	0.111 7	0.001 28	0.2308	—	2.9057	0.2809	211.7	1.258	1.843
0.7838	783.1	782.0	0.107 5	0.001 22	0.2098	—	2.9057	0.2809	220.0	1.211	1.811
0.7744	773.7	772.6	0.126 9	0.001 13	0.2364	—	3.3900	0.2408	186.3	1.324	1.839
—	—	—	—	—	0.0869	0.9938	0.9686	0.8428	—	1.514	—
0.5231^{h}	522.6^{h,x}	521.5^{h}	0.080 69^{h}	0.003 40^{h}	0.1443	0.9844	1.4529	0.5619	293.6^{h}	1.480	2.443
0.6019^{h}	601.4^{h}	600.3^{h}	0.093 30^{h}	0.002 09^{h}	0.1949	0.9703	1.9372	0.4214	253.4^{h}	1.483	2.237
0.6277^{h}	627.1^{h}	626.0^{h}	0.089 47^{h}	0.001 76^{h}	0.2033	0.9660	1.9372	0.4214	264.3^{h}	1.366	2.241
0.6105^{h}	610.0^{h}	608.9^{h}	0.091 98^{h}	0.001 93^{h}	0.2126	0.9661	1.9372	0.4214	257.1^{h}	1.528	2.238
0.6010^{h}	600.5^{h}	599.4^{h}	0.093 44^{h}	0.002 16^{h}	0.2026	0.9688	1.9372	0.4214	253.1^{h}	1.547	2.296
0.6462	645.6	644.5	0.108 6	0.001 60	0.2334	0.948†	2.4215	0.3371	217.7	1.519	2.241(43)
0.6576^{h}	657.^{h}	656.^{h}	0.082 33^{h}	0.001 76^{h}	(0.2540)	(0.969)	1.8676	0.4371	287.2^{h}	1.446	2.262
0.6280^{h}	627.4^{h}	626.3^{h}	0.086 22^{h}	0.002 03^{h}	0.1971	(0.965)	1.8676	0.4371	274.2^{h}	1.426	2.124
0.6866	686.0	684.9	0.099 30	0.001 55	(0.1567)	0.949†	2.3519	0.3471	238.1	1.492	2.171
0.615^{h}	—	—	—	—	0.1893	0.9925	0.8990	0.9081	—	1.659	—
0.8850	884.2	883.1	0.088 34	0.001 19	0.2095	0.929†	2.6969	0.3027	267.6	1.014	1.715
0.8723	871.6	870.5	0.105 7	0.001 08	0.2633	0.903†	3.1812	0.2566	223.7	1.085	1.677
0.8721	871.3	870.5	0.121 9	0.000 97	0.3031	—	3.6655	0.2227	194.0	1.168	1.721
0.8850	884.2	883.1	0.120 1	0.000 99	0.3113	—	3.6655	0.2227	196.9	1.218	1.741
0.8691	868.3	867.2	0.122 3	0.000 97	0.3257	—	3.6655	0.2227	193.4	1.163	1.696
0.8661	865.3	864.2	0.122 7	0.000 97	0.3214	—	3.6655	0.2227	192.7	1.157	1.708
0.9115	910.6	909.5	0.114 4	0.001 03	0.1997	—	3.5959	0.2270	206.7	1.133	1.724
0.8667	866.0	864.9	0.139 0	0.000 97	0.3260	—	4.1498	0.1967	170.4	1.219	1.732
0.7967	796.0	794.9	0.040 25	0.001 17	0.5648	—	1.1063	0.7379	587.4	1.352	2.484
0.7922	791.5	790.4	0.058 20	0.001 07	0.6608	—	1.5906	0.5132	406.2	1.389	2.348
0.7893^{m}	788.6^{m}(34)	—	0.035 52^{m}	—	0.0442	0.9995	0.9671	0.8441	—	1.040	—
0.8226^{h}	821.9^{h}(35)	820.8^{h}	0.053 55^{h}	—	0.2667	0.9943	1.5195	0.5373	441.6^{h}	0.8330	—
0.7897^{h}	789.0^{h,x}(36)	787.9^{h}	0.043 19^{h}	—	0.0920	0.9903	1.1765	0.6939	547.5^{h}	0.9960	2.08(36)
1.397^{h}	1396.^{h,x}(36)	1395.^{h}	0.045 89^{h}	—	0.2548	0.9801†	2.2117	0.3691	515.3^{h}	0.6062	1.359(36)
0.6183^{h}	617.7^{h,x}(30)	616.6^{h}	0.027 57^{h}	—	0.2576	0.9899(30)	0.5880	1.388	857.4	2.079	4.693(30)
0.856^{m}(36)	855.^{m}	—	0.033 9^{m}	—	—	0.9996	1.0000	0.8163	—	1.005	—
0.07106^{m}	71.00^{m}(37)	—	0.028 39^{m}	—	−0.219^{w}	1.0006	0.0696	11.73	—	14.24	—
1.1420^{m}(25)	1141.^{m}(38)	—	0.028 04^{m}	—	0.0200	0.9993(39)	1.1048	0.7389	—	0.9166	—
0.8093^{m}(26)	808.6^{m}(31)	—	0.034 64^{m}	—	0.0372	0.9997	0.9672	0.8441	—	1.040	—
1.426	1424.5	1423.5	0.049 78	—	0.0737	(0.9875)†(36)	2.4481	0.3335	475.0	0.4760	—
1.000	999.1	998.0	0.018 03	0.000 14	0.3434	—	0.6220	1.312	1311.	1.862	4.191
0.1251^{m}	125.0^{m}(32)	—	0.032 02^{m}	—	0	1.000 5(40)	0.1382	5.907	—	5.192	—
0.8538	853.0^{x}	851.9	0.042 74	0.006 03	0.1232	—	1.2588	0.6485	553.2	0.7991	—

PHYSICAL CONSTANTS OF HYDROCARBONS(27)

	See Note No →	10				11.	12.	13	Flammability limits, vol % in air mixture		ASTM octane number	
		Heating value, 15 °C*										
		Net	Gross^p									
No.	Compound	MJ/m^3 Ideal gas, 101.3250 kPa (abs)	MJ/m^3 Ideal gas, 101.3250 kPa (abs)	MJ/kg Liquid (Mass in vacuum)	MJ/m^3 Liquid (Mass in vacuum)	Heat of vaporization, 101.3250 kPa (abs) at boiling point, kJ/kg	Refractive index *, n_D 15 °C	Air required for combustion ideal gas *, m^3(air)/m^3(gas)	Lower *	Higher	Motor method D 357	Research method D 908
1	Methane	33.936	37.694	—	—	509.86	—	9.54	5.0	15.0	—	—
2	Ethane	60.395	66.032	51.586^h	18 458^h	489.36	1.214 04^h	16.70	2.9	13.0	+.05^f	+1.6^{d,f}
3	Propane	86.456	93.972	50.008^h	25 394^h	425.73	1.219 05^h	23.86	2.1	9.5	97.1	+1.8^{d,f}
4	*n*-Butane	112.384	121.779	49.158^h	28 718^h	385.26	1.332 92^h	31.02	1.8	8.4	89.6^f	93.8^f
5	Isobutane	112.031	121.426	49.044^h	27 621^h	366.40	—	31.02	1.8	8.4	97.6	+0.10^{d,f}
6	*n*-Pentane	138.380	149.654	48.667	30 709	357.22	1.360 24	38.18	1.4	8.3	62.6^f	61.7^f
7	Isopentane	138.044	149.319	48.579	30 333	342.20	1.356 58	38.18	1.4	(8.3)	90.3	92.3
8	Neopentane	137.465	148.739	48.427^h	28 896^h	315.34	1.345	38.18	1.4	(8.3)	80.2	85.5
9	-Hexane	164.402	177.556	48.344	32 091	334.81	1.377 46	45.34	1.2	7.7	26.0	24.8
	Methylpentane	164.075	177.229	48.273	31 749	322.52	1.374 17	45.34	1.2	(7.7)	73.5	73.4
	3-Methylpentane	164.188	177.341	48.300	32 303	325.82	1.379 18	45.34	(1.2)	(7.7)	74.3	74.5
2	Neohexane	163.683	176.836	48.191	31 512	305.24	1.371 57	45.34	1.2	(7.7)	93.4	91.8
13	2,3-Dimethylbutane	164.025	177.179	48.269	32 157	316.50	1.377 59	45.34	(1.2)	(7.7)	94.3	+0.3^f
14	*n*-Heptane	190.398	205.431	48.104	33 095	316.33	1.390 17	52.50	1.0	7.0	0.0	0.0
15	2-Methylhexane	190.099	205.132	48.051	32 809	306.06	1.387 43	52.50	(1.0)	(7.0)	46.4	42.4
16	3-Methylhexane	190.243	205.276	48.082	33 249	307.27	1.391 19	52.50	(1.0)	(7.0)	55.8	52.0
17	3-Ethylpentane	190.327	205.359	48.101	33 796	308.94	1.395 94	52.50	(1.0)	(7.0)	69.3	65.0
18	2,2-Dimethylpentane	189.630	204.662	47.964	32 520	291.03	1.384 75	52.50	(1.0)	(7.0)	95.6	92.8
19	2,4-Dimethylpentane	189.803	204.836	48.000	32 501	294.41	1.384 08	52.50	(1.0)	(7.0)	83.8	83.1
20	3,3-Dimethylpentane	189.885	204.918	48.019	33 488	295.87	1.393 42	52.50	(1.0)	(7.0)	86.6	80.8
21	Triptane	189.690	204.722	47.982	33 319	288.90	1.391 96	52.50	(1.0)	(7.0)	+0.1^f	+1.8^f
22	*n*-Octane	216.374	233.286	47.919	33 865	301.26	1.399 81	59.65	0.96	—	—	—
23	Diisobutyl	215.797	232.709	47.832	33 372	285.69	1.394 88	59.65	(0.98)	—	55.7	55.2
24	Isooctane	215.732	232.644	47.843	33 299	271.44	1.393 92	59.65	1.0	—	100.	100.
25	*n*-Nonane	242.398	261.189	47.783	34 485	288.82	1.407 73	66.81	0.87^e	2.9	—	—
26	*n*-Decane	268.396	289.066	47.670	34 985	276.06	1.414 11	73.97	0.78^e	2.6	—	—
27	Cyclopentane	131.114	140.509	46.955	35 225	389.20	1.409 27	35.79	(1.4)	—	84.9^f	+0.1^f
28	Methylcyclopentane	156.757	168.032	46.825	35 278	345.51	1.412 40	42.95	(1.2)	8.35	80.0	91.3
29	Cyclohexane	156.034	167.308	46.606	36 497	355.95	1.428 92	42.95	1.3	7.8	77.2	83.0
30	Methylcyclohexane	181.567	194.720	46.525	35 997	317.03	1.425 66	50.11	1.2	—	71.1	74.8
31	Ethene (Ethylene)	55.942	59.700	—	—	482.77	—	14.32	2.7	34.0	75.6	+0.03^f
32	Propene (Propylene)	81.482	87.119	—	—	437.68	—	21.48	2.0	10.0	84.9	+0.2^f
33	1-Butene (Butylene)	107.475	114.991	48.081^h	28 916^h	390.60	—	28.63	1.6	9.3	80.8^f	97.4
34	*cis*-2-Butene	107.191	114.707	47.927^h	30 055^h	416.10	—	28.63	(1.6)	—	83.5	100.
35	*trans*-2-Butene	106.957	114.473	47.843^h	29 184^h	405.56	—	28.63	(1.6)	—	—	—
36	Isobutene	106.755	114.271	47.769^h	28 685^h	394.18	—	28.63	(1.6)	—	—	—
	-Pentene	133.465	142.860	47.788	30 852	359.25	1.374 61	35.79	1.4	8.7	77.1	90.9
	,2-Butadiene	104.118	109.755	47.504^h	31 210^h	(449.6)	—	26.25	(2.0)	(12.)	—	—
	1,3-Butadiene	101.917	107.555	46.608^h	29 242^h	(418.7)	—	26.25	2.0	11.5	—	—
40	Isoprene	127.330	134.846	46.408	31 836	(385.2)	1.425 36	33.41	(1.5)	—	81.0	99.1
41	Acetylene	53.098	54.978	—	—	—	—	11.93	2.5	80.	—	—
42	Benzene	134.055	139.692	41.843	36 998	393.32	1.504 32	35.79	1.3^g	7.9^g	+2.8^f	—
43	Toluene	159.534	167.050	42.450	37 000	360.14	1.499 73	42.95	1.2^g	7.1^g	+0.3^f	+5.8^f
44	Ethylbenzene	185.555	194.950	43.014	37 478	334.98	1.498 56	50.11	0.99^g	6.7^g	97.9	+0.8^f
45	*o*-Xylene	185.092	194.487	42.900	37 935	346.80	1.507 95	50.11	1.1^g	6.4^g	100.	—
46	*m*-Xylene	185.020	194.415	42.891	37 245	342.47	1.499 80	50.11	1.1^g	6.4^g	+2.8^f	+4.0^f
47	*p*-Xylene	185.050	194.445	42.901	37 122	338.92	1.498 39	50.11	1.1^g	6.6^g	+1.2^f	+3.4^f
48	Styrene	180.290	187.806	42.213	38 439	(351.23)	1.549 69	47.72	1.1	6.1	+0.2^f	>+3.^{f}
49	Isopropylbenzene	211.328	222.603	43.410	37 591	312.25	1.494 00	57.27	0.88^g	6.5^g	99.3	+2.1^f
50	Methyl alcohol	28.601	32.360	22.685	18 057	1075.97	1.330 28	7.16	6.72(5)	36.50	—	—
51	Ethyl alcohol	54.062	59.699	29.707	23 513	840.54	1.363 45	14.32	3.28(5)	18.95	—	—
52	Carbon monoxide	11.959	11.959	—	—	215.70	1.000 36	2.39	12.50(5)	74.20	—	—
53	Carbon dioxide	0	0	—	—	573.27^n	1.000 49	—	—	—	—	—
54	Hydrogen sulfide	21.912	23.791	—	—	548.01	1.000 61	7.16	4.30(5)	45.50	—	—
55	Sulfur dioxide	—	—	—	—	387.74	1.000 62	—	—	—	—	—
56	Ammonia	17.301	20.121	—	—	1366.	1.000 36	3.58	15.50(5)	27.00	—	—
57	Air	—	—	—	—	214.	—	—	—	—	—	—
58	Hydrogen	10.230	12.091	—	—	450.4	1.000 13	2.39	4.00(5)	74.20	—	—
59	Oxygen	—	—	—	—	213.	1.000 27	—	—	—	—	—
60	Nitrogen	—	—	—	—	204	1.000 28	—	—	—	—	—
61	Chlorine	—	—	—	—	288.0	1.387 8^y	—	—	—	—	—
62	Water	0	1.879	0	0	2257.	1.333 47	—	—	—	—	—
63	Helium	—	—	—	—	—	1.000 03	—	—	—	—	—
64	Hydrogen chloride	—	—	—	—	431.5	1.000 42	—	—	—	—	—

REFERENCES

(2) International Critical Tables.
(3) Hodgman, "Handbook of Chemistry and Physics", 31st edition, 1949
(5) Jones, *Chem. Rev.*, 1938, 22, 1
(23) Din, F., Ed. "Thermodynamic Functions of Gases", Butterworths London, 1956
(25) Roder, H. M.; Weber, L. A. *NASA Spec. Publ.* SP-3071, 1972
(26) Jacobsen, R. T.; Stewart, R. B.; McCarty, R. D.; Hanley, H. J. M. *Nat. Bur. Stand. U.S. Tech. Note* No. 648, 1973
(27) Values for hydrocarbons, compound numbers 1-49, were selected or calculated from data in the tables of the Thermodynamics Research Center Hydrocarbon Project (formerly API Research Project 44) and the values for the remaining compounds, numbers 50-64, were selected or calculated from data in the tables of the Thermodynamics Research Center Data Project unless indicated otherwise
(28) Angus, S.; Armstrong, B.; de Reuck, K. M., Eds. "Methane International Thermodynamic Tables of the Fluid State—5", Pergamon Press: Oxford, 1976
(29) Angus, S.; Armstrong, B.; de Reuck, K. M., Eds. "International Thermodynamic Tables of the Fluid State—Ethylene, 1972", Butterworths: London, 1974
(30) Haar, L.; Gallagher, J. S. *J. Phys. Chem. Ref. Data* 1978, *7*, 635.
(31) Angus, S.; de Reuck, K. M.; Armstrong, B., Eds. "Nitrogen International Thermodynamic Tables of the Fluid State—6", Pergamon Press: Oxford, 1979
(32) Angus, S.; de Reuck, K. M., Eds. "Helium. International Thermodynamic Tables of the Fluid State—4", Pergamon Press: Oxford, 1977.
(33) Kudchadker, A. P.; Alani, G. H.; Zwolinski, B. J. *Chem. Rev.* 1968, *68*, 659.
(34) Hust, J. G.; Stewart, R. B. "Thermodynamic Property Values for Gaseous and Liquid Carbon Monoxide from 70 to 300 K with Pressures to 300 Atmospheres," *Nat. Bur. Stand. U.S. Tech. Note* No. 202, November 30, 1963.
(35) Angus, S.; Armstrong, B.; de Reuck, K. M., Eds. "Carbon Dioxide. International Thermodynamic Tables of the Fluid State—3", Pergamon Press: Oxford, 1976.
(36) "The Matheson Unabridged Gas Data Book"; Matheson Gas Products: New York, 1974.
(37) Dean, J. W. "A Tabulation of the Thermodynamic Properties of Normal Hydrogen from Low Temperatures to 300 K and from 1 to 100 Atmospheres"; *Nat. Bur. Stand. U.S. Tech. Note* No. 120, November 1961.
(38) McCarty, R. D.; Weber, L. A. "Thermophysical Properties of Oxygen from the Freezing Line to 600°R for Pressures to 5000 Psia"; *Nat. Bur. Stand. U.S. Tech Note* No. 384, July 1971.
(39) Weber, L. A. *Nat. Bur. Stand. U.S. Rept* No. 9710, 1968.
(40) Tsederberg, N. V.; Popov, V. N.; Morozova, N. A. "Thermodynamic and Thermophysical Properties of Helium"; Atomizat: Moscow, 1969; Israel Program for Scientific Translation, 1971.
(41) Touloukian, Y. S.; Makita, T. "Thermophysical Properties of Matter, Vol. 6, Specific Heat, Non-metallic Liquids and Gases"; IFI/Plenum: New York, 1970.
(42) Schlinger, W. G.; Sage, B. H. *Ind. Eng. Chem.* 1952, *44*, 2454.
(43) Todd, S. S.; Oliver, G. D.; Huffman, P. M. *J. Am. Chem. Soc.* 1947, *69*, 1519.

NOTES

a Air saturated liquid.
b Absolute values from weights in vacuum.
c The apparent values from weight in air are shown for users' convenience and compliance with ASTM-IP Petroleum Measurement Tables. In the United States and Great Britain, all commercial weights are required by law to be weights in air. All other mass data are on an absolute mass (weight in vacuum) basis.
d At saturation pressure (triple point).
e Sublimation point.
f The + sign and number following signify the ASTM octane number corresponding to that of 2,2,4-trimethylpentane with the indicated number of cm³ of TEL added per gal.
g Determined at 100°C.
h Saturation pressure and 15°C.
i Apparent value at 15°C.
j Average value from octane numbers of more than one sample
k Relative density (specific gravity), 48.3°C/15°C (sublimation point; solid C_2H_4/liquid H_2O).
m Densities of liquid at the boiling point.
n Heat of sublimation.
p See Note 10.
s Extrapolated to room temperature from higher temperature.
t Gross calorific values shown for ideal gas volumes are not direct conversions of each other using only the gas volume per liquid volume value shown herein. The values differ by the heat of vaporization to ideal gas at 288.15 K.
u Fixed points on the 1968 International Practical Temperature Scale (IPTS-68).
w Value for normal hydrogen (25% para, 75% ortho). The value for equilibrium mixture of para and ortho is −0.218; however in most correlations, 0 is used.
x Densities at the boiling point in kg/m³ for: Ethane, 546.4; propane, 581.0; propene, 608.8; hydrogen sulfide, 960.; sulfur dioxide, 1462.; ammonia, 681.6; hydrogen chloride, 1192.
* Calculated values.
( ) Estimated values.
† Values are estimated using 2nd virial coefficients.

1. Molecular mass (M) is based on the following atomic weights: C = 12.011, H = 1.008; O = 15.9994; N = 14.0067; S = 32.06; Cl = 35.453.
2. Boiling point—the temperature at equilibrium between the liquid and vapor phases at 101.3250 kPa (abs).
3. Freezing point—the temperature at equilibrium between the crystalline phase and the air saturated liquid at 101.3250 kPa (abs).
4. All values for the density and molar volume of liquids refer to the air saturated liquid at 101.3250 kPa(abs), except when the boiling point is less than 15°C. In such cases, the density and molar volume are given for the liquid in equilibrium with its vapor at 15°C.

The relative density 15°C is defined as $\dfrac{\rho(\text{liquid, }15°C)}{\rho(\text{water, }15°C)}$

The density of water at 15°C is taken as 999.10 kg/m³.

The apparent density in air ρ(mass in air) is related to the density in vacuum ρ(mass in vacuum) by

$$\rho(\text{mass in air}) = [\rho(\text{mass in vacuum}) - \rho(\text{air})]\,\frac{\rho(\text{brass})}{\rho(\text{brass}) - \rho(\text{air})}$$

"Mass in air" is the ratio of gravitational force on the object to the force of a standard brass weight, with both objects immersed in air.

The density of air at 15°C and 101.3250 kPa(abs) is taken as 1.22 kg/m³, and the density of brass as [illegible] kg/m³.

The molar volume is related to density by

$$V(\text{m}^3/\text{kmol}) = \frac{M}{\rho(\text{kg/m}^3)}$$

5. The temperature coefficient of density is related to the expansion coefficient by $\dfrac{1}{\rho}\left(\dfrac{d\rho}{dT}\right)_P = -\dfrac{1}{V}\left(\dfrac{dV}{dT}\right)_P$ in units of −1/°C
6. Pitzer acentric factor ω $\omega = -\log_{10}\left(\dfrac{P}{P_c}\right) - 1 \quad \text{at } T_r = \dfrac{T}{T_c} = 0.7$

   P is calculated at T from same vapor pressure equation used to calculate vapor pressure at 40°C in Fig. 16-1.
7. The compressibility factor of the real gas, Z, is $Z = \dfrac{PV}{RT}$
8. The density of an ideal gas relative to air is M(gas)/M(air). The molecular mass of air is taken as 28.964. The specific volume of an ideal gas is $V = \dfrac{RT}{MP}$

   The volume ratio is V(ideal gas)/V(liquid in vacuum)
9. The specific heat capacity of the ideal gas was calculated from the constant pressure molar heat capacity reported in the references cited. These were calculated from the molecular partition functions. This quantity is independent of pressure. The specific heat capacities of liquids are derived from experimental measurements. For liquids boiling below 15°C, the values given are C_s/M, where C_s is the molar heat capacity of the liquid under saturated vapor pressure conditions.
10. The heating value is the negative of the enthalpy of combustion ($-\Delta H_c^\circ$) of a gas or liquid in its standard state at 15°C to give combustion products in their standard states, all at 101.3250 kPa (abs). For a compound where the formula is $C_nH_mO_hS_jN_k$, the combustion reaction is

$$C_nH_mO_hS_jN_k(\text{s, l, or g}) + \left(n + \frac{m}{4} - \frac{h}{2} + j\right)O_2(g) \rightarrow nCO_2(g) + \frac{m}{2}H_2O(\text{g or l}) + \frac{k}{2}N_2(g) + jSO_2(g)$$

For the gross heating value, the water produced by the combustion is in the liquid form, and for the net heating value, the water is in the gas state.

The enthalpies of combustion at 15°C were calculated from those at 25°C, as reported in the references cited, by

$$\Delta H_c^\circ(15°C) = \Delta H_c^\circ(25°C) - 10\Delta C_p^\circ, \text{ where}$$

$$\Delta C_p^\circ = nC_p^\circ(CO_2, g) + \frac{m}{2}C_p^\circ(H_2O, \text{g or l}) + \frac{k}{2}C_p^\circ(N_2, g) + jC_p^\circ(SO_2, g) - C_p^\circ(C_nH_mO_hS_jN_k, \text{s, l, or g}) - \left(n + \frac{m}{4} - \frac{h}{2} + j\right)C_p^\circ(O_2, g) \text{ at } 25°C$$

$$\Delta H_c^\circ(\text{MJ/m}^3) = \Delta H_c^\circ(\text{MJ/kg})/\rho(\text{kg/m}^3, \text{vacuum}).$$

The above equations apply to the combustion of an (initially) "bone-dry" gas. The heat of combustion of 1 m³ of an ideal gas, saturated with water vapor, is

$$\Delta H_c^\circ(\text{g, sat})(\text{MJ/m}^3) = 0.98322\,\Delta H_c^\circ(g)(\text{MJ/m}^3)$$

The vapor pressure of water at 15°C is 1.70 kPa (abs). The gross heat of combustion of 1 m³ of gas, saturated with water vapor for which all of the water is condensed to liquid in the final products, is

$$\Delta H_c^\circ(\text{g, sat})(\text{MJ/m}^3) = 0.98322\,\Delta H_c^\circ(g)(\text{MJ/m}^3) - 0.0315$$

The heat of vaporization of water to an ideal gas at 15°C is 44.43 kJ/mol.

11. The heat of vaporization is the enthalpy of the saturated vapor at the boiling point at 101.3250 kPa (abs) minus the enthalpy of the liquid at the same condition.
12. The refractive index reported refers to the liquid or gas and is measured for light of wavelength corresponding to the sodium D-line (589.26 nm).
13. Air required for combustion of ideal gas for compounds of formula $C_nH_mO_hS_jN_k$ $\dfrac{V(\text{air})}{V(\text{gas})} = \dfrac{n + 1/4\,m - 1/2\,h + j}{0.2095}$

COMMENTS

Units—all dimensional values are reported in SI units, which are derived from the following basic units:
mass—kilogram, kg
length—meter, m
temperature—International Practical Temperature Scale of 1968 (IPTS-68), where 0°C = 273.15 K

Other derived units are:
volume—cubic meter, m³
pressure—Pascal, Pa (1 Pa = N/m²)

Physical constants for molar volume = 22.41383 ± 0.00031
gas constant, R = 8.31441 J/(K · mol)
8.31441 × 10⁻³ m³ · kPa/(K · mol)
1.98719 cal/(K · mol)
1.98596 Btu(IT)/(°R · lb-mol)

Conversion factors
1 m³ = 35.31467 ft³ = 264.1720 gal
1 kg = 2.204623 lb
1 kg/m³ = 0.06242795 lb/ft³ = 0.001 g/cm³
1 kPa = 0.01 bar = 9.869233 × 10⁻³ atm = 0.1450377 lb/in²
1 atm = 101.3250 kPa = 14.69595 lb/in² = 760 Torr
1 kJ = 0.2390057 kcal(thermochemical) = 0.2388459 kcal(IT) = 0.9478171 Btu(IT)

see Rossini, F. D. "Fundamental Measures and Constants for Science and Technology", CRC Press, Cleveland, Ohio, 1974

COMPRESSIBILITY FACTORS FOR NATURAL GAS

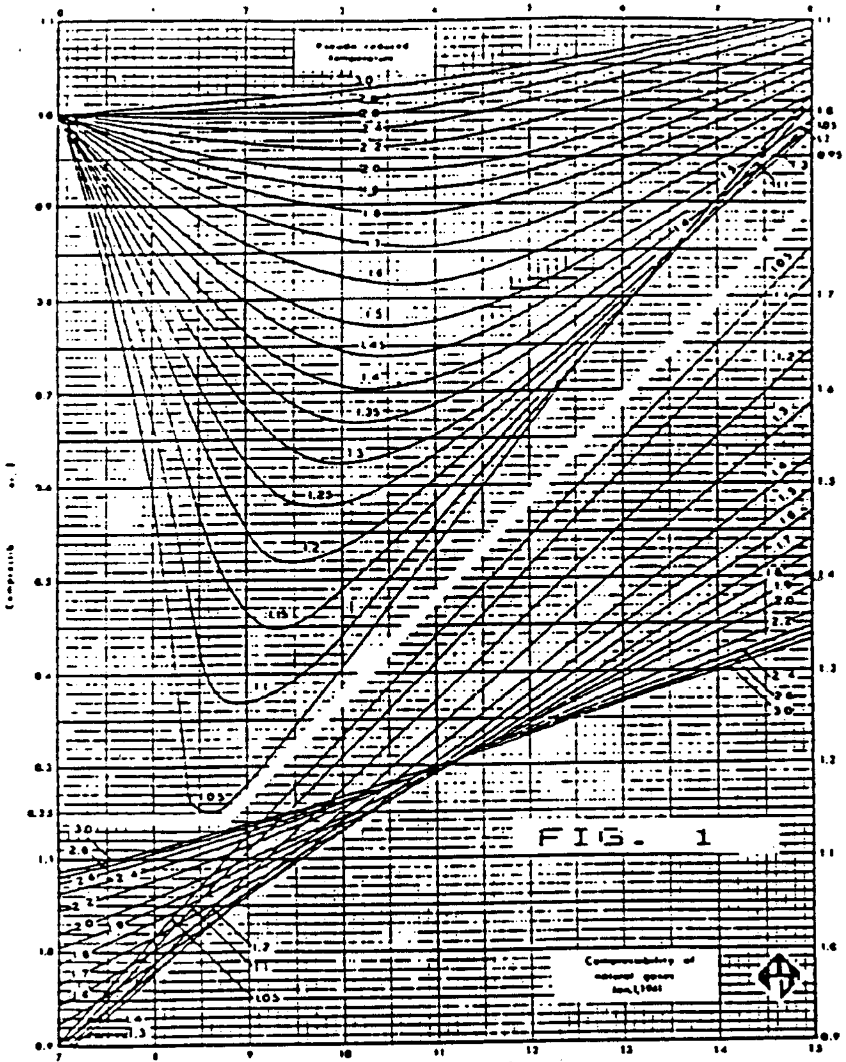


FIG. 1

Compressibility factors at low reduced pressures

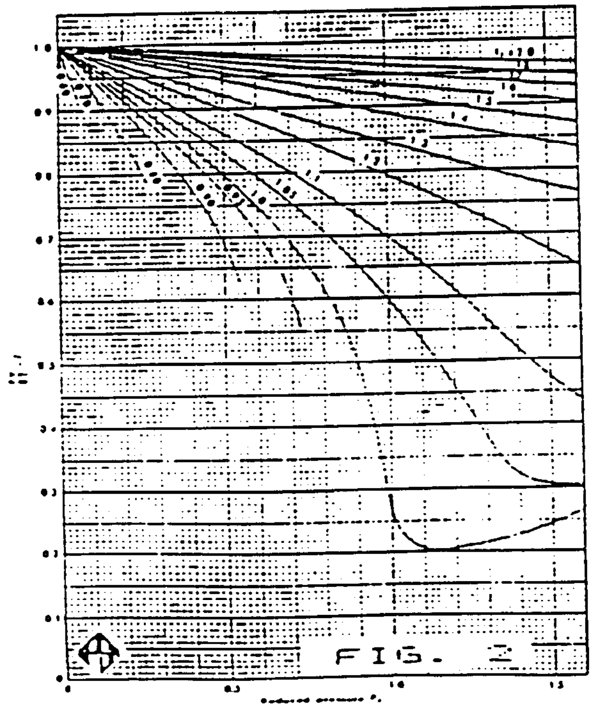
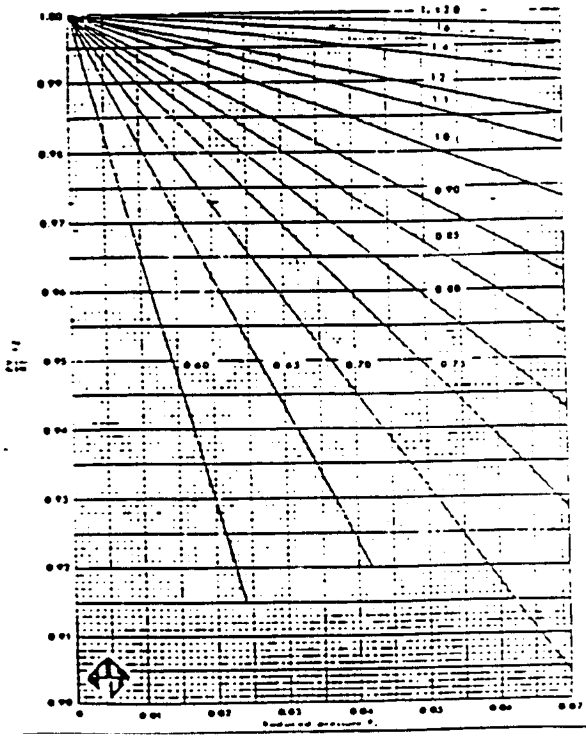


FIG. 2

FIG. 3

Compressibility factors for gases near atmospheric pressure



Pseudo critical pressure FIG. 4

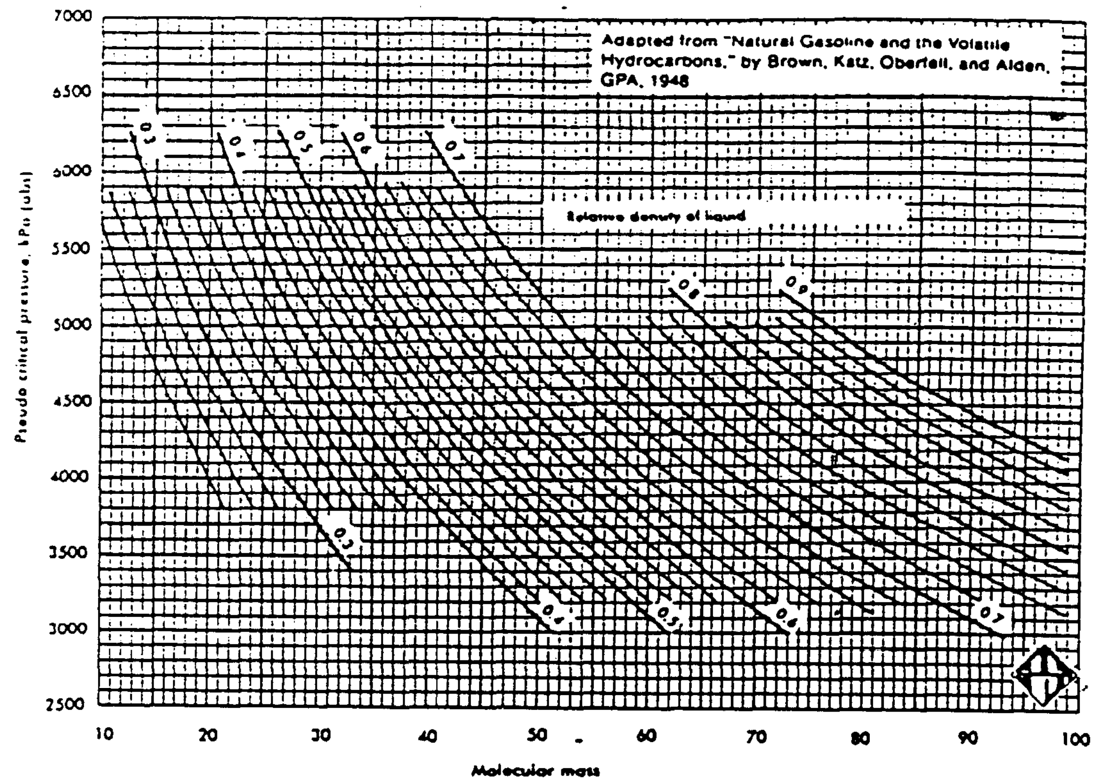
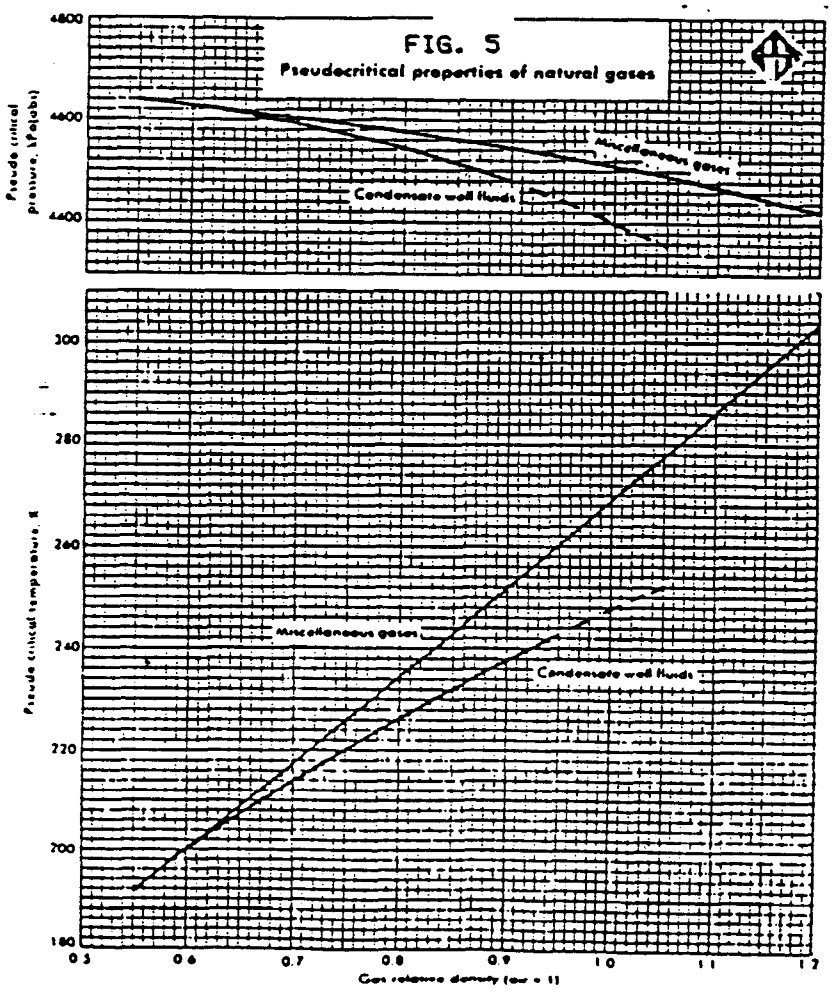


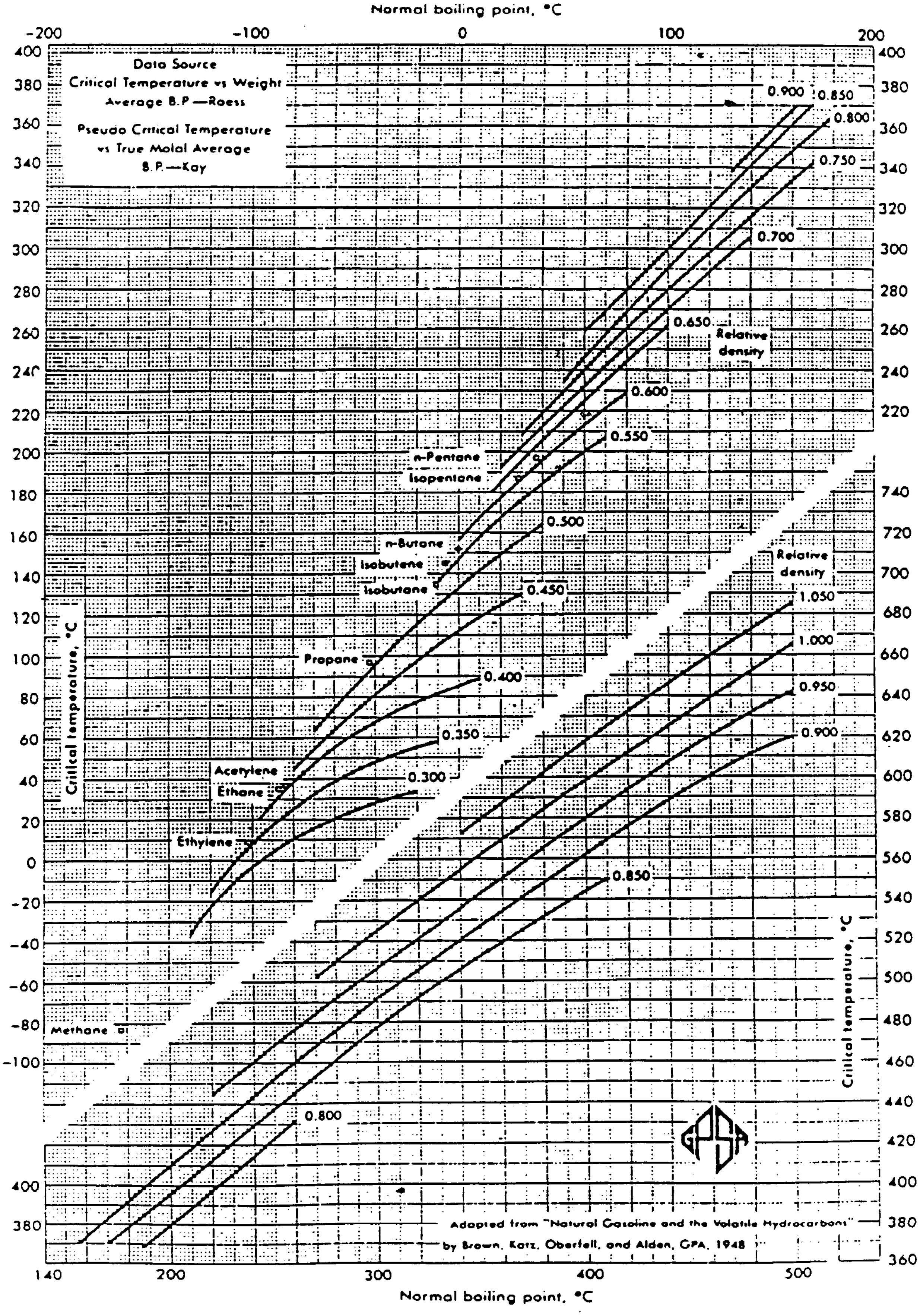
FIG. 5

Pseudocritical properties of natural gases



Critical temperatures

FIG. 6



Data Source
Critical Temperature vs Weight Average B.P.—Roess
Pseudo Critical Temperature vs True Molal Average B.P.—Kay

Adapted from "Natural Gasoline and the Volatile Hydrocarbons" by Brown, Katz, Oberfell, and Alden, GPA, 1948

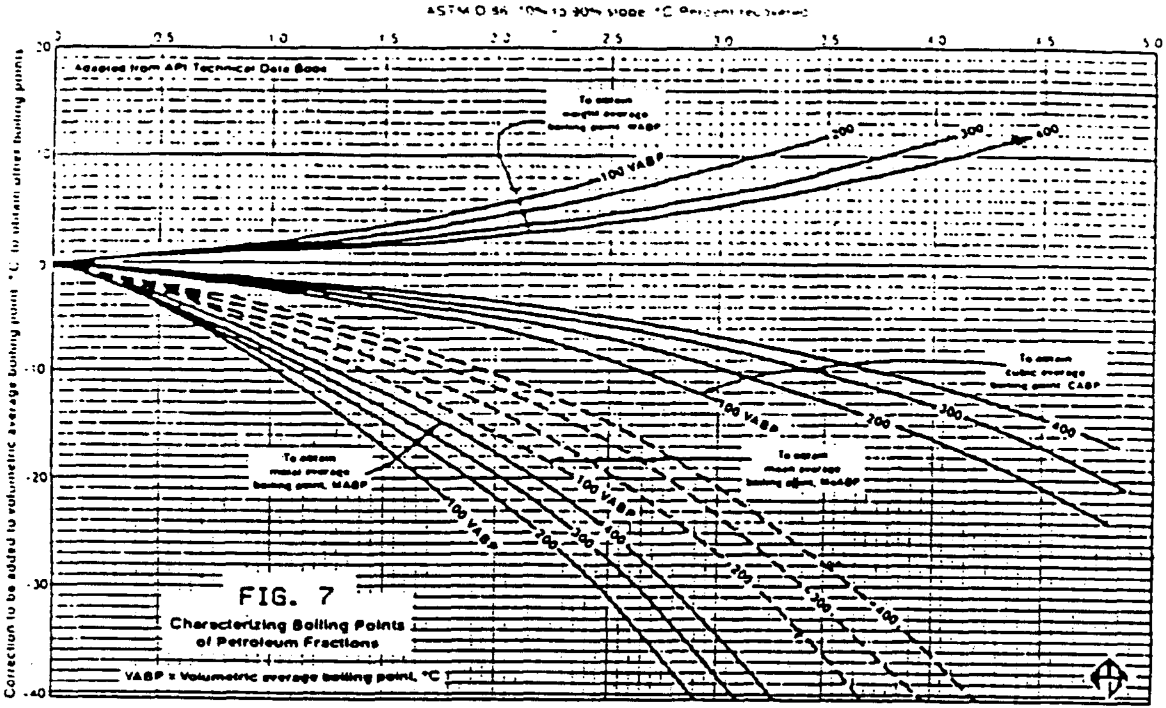


FIG. 7

Characterizing Boiling Points of Petroleum Fractions

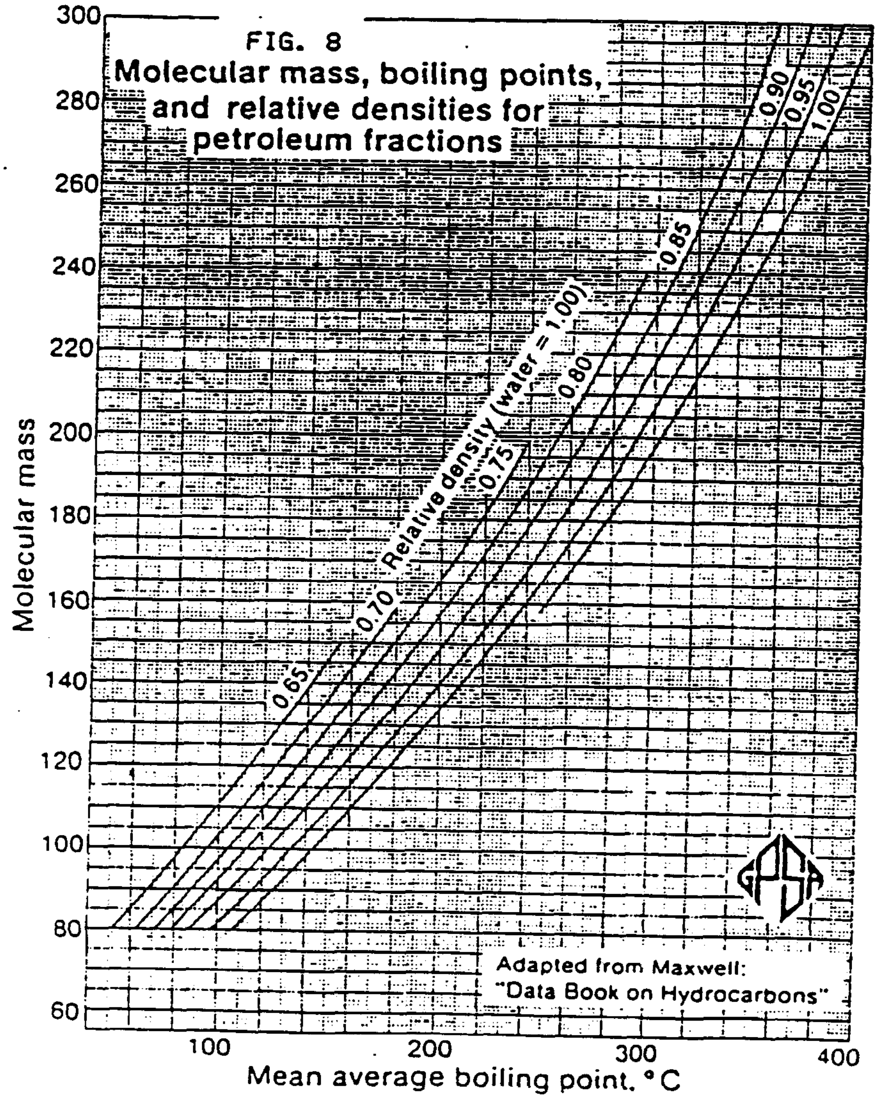


FIG. 8

Molecular mass, boiling points, and relative densities for petroleum fractions

Adapted from Maxwell: "Data Book on Hydrocarbons"

DENSITY

VAPOUR DENSITY

Vapour densities or molar volumes can be calculated from the equation :

$$V = \frac{ZRT}{P}$$

$$\boxed{\rho = \frac{MW.P}{ZRT}}$$

p	psia	bara
T	°R	°K
R	10.73	0.08314
	lbs/ft3	Kg/m3

Specific gravity of a gas $= \frac{MWg}{MWair}$

MWair = 28.967

LIQUID DENSITY

The density of a multi component mixture can be calculated using the summation of the component densities :

$$\rho_m = \sum W_i / \sum \frac{W_i}{\rho_i}$$

Wi = mass component
ρi = density component

liquid densities for hydrocarbon mixtures can be estimated using. Figures 10, 11 in this section.

Approximate relative density of petroleum fractions

FIG. 10

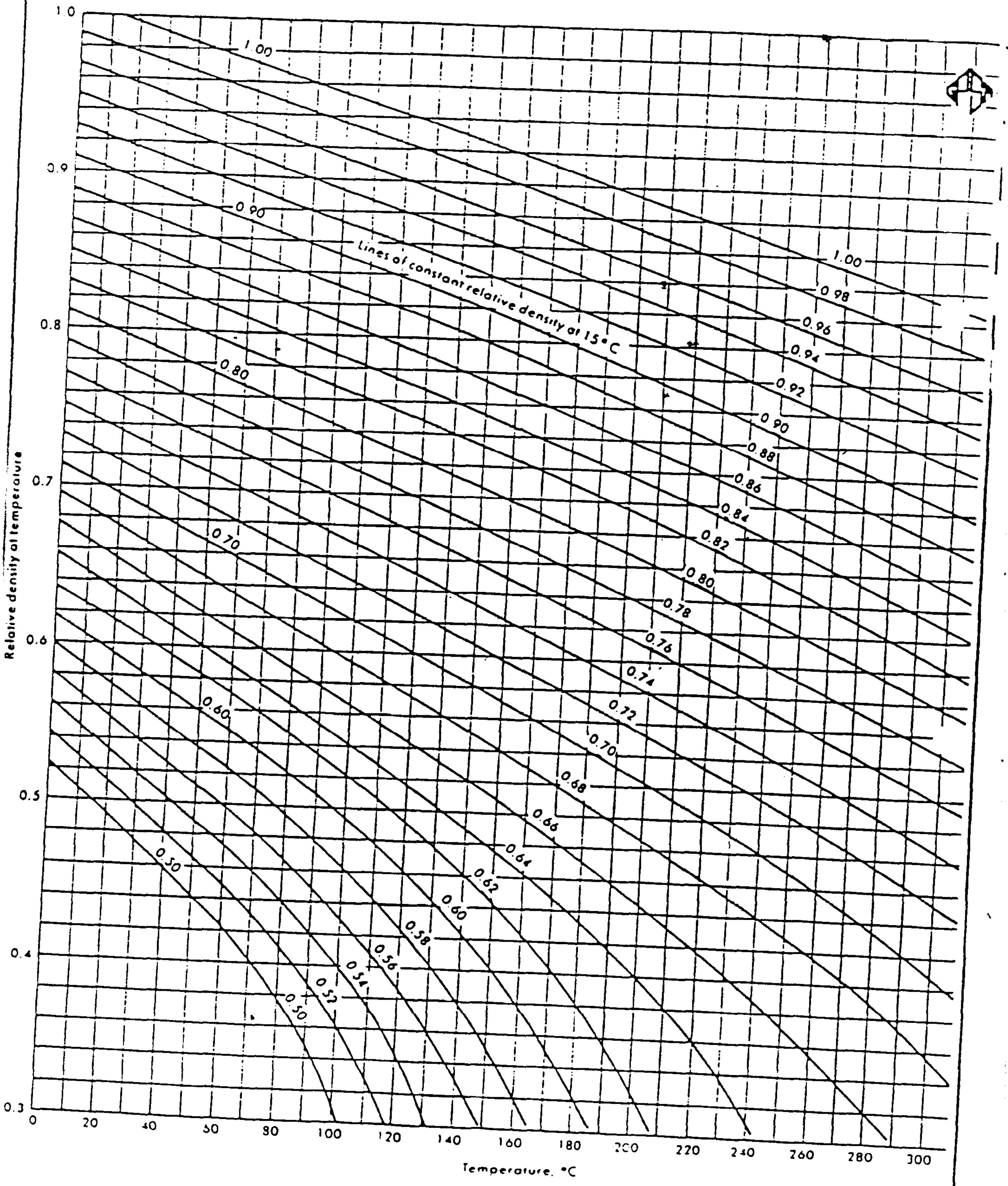
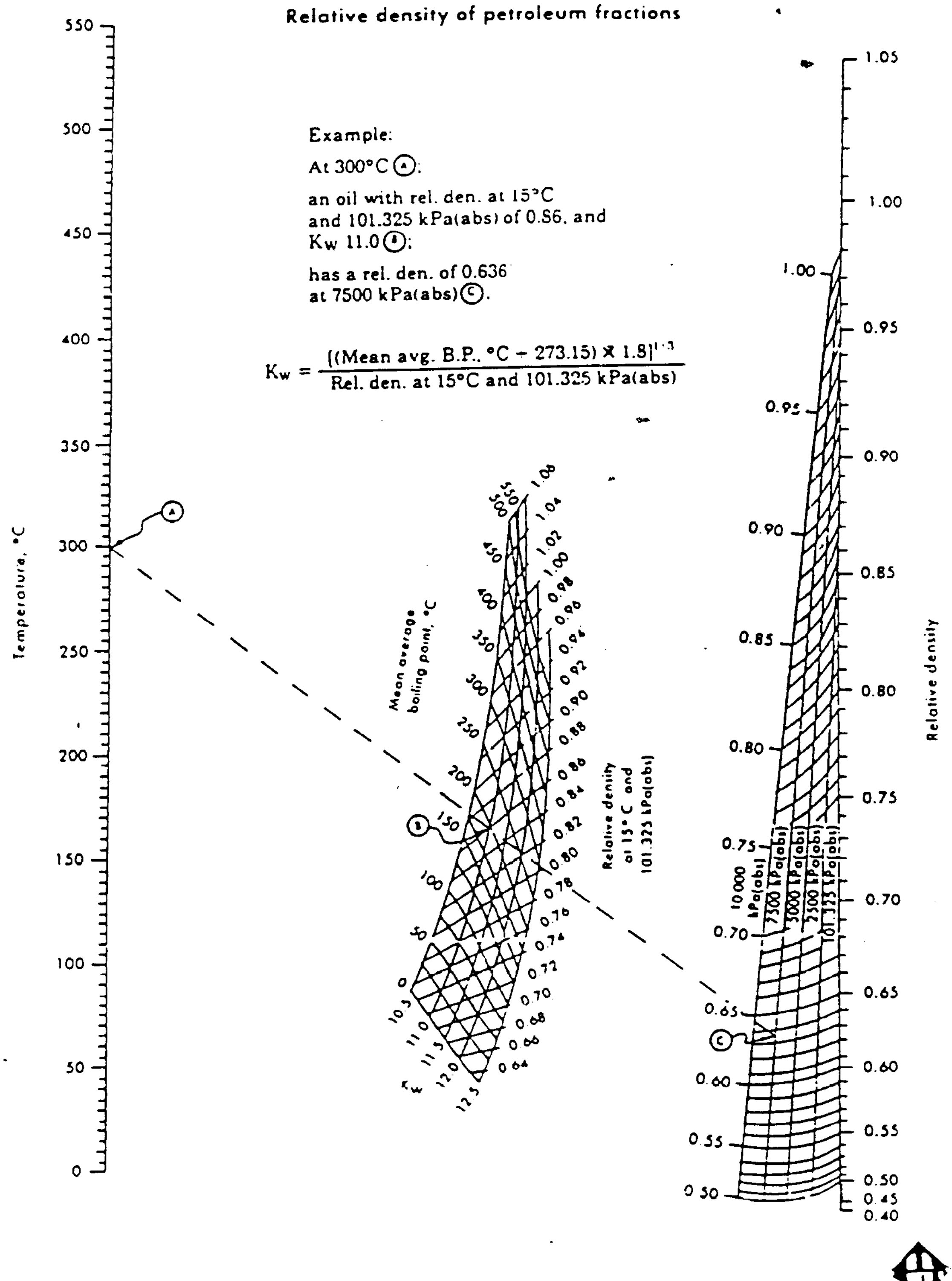


FIG. 11

Relative density of petroleum fractions



Example:
At 300°C Ⓐ:
an oil with rel. den. at 15°C
and 101.325 kPa(abs) of 0.86, and
Kw 11.0 Ⓑ:
has a rel. den. of 0.636
at 7500 kPa(abs) Ⓒ.

$$K_w = \frac{[(\text{Mean avg. B.P., °C} + 273.15) \times 1.8]^{1/3}}{\text{Rel. den. at 15°C and 101.325 kPa(abs)}}$$

Adapted to SI by GPSA from Ritter, Lenoir, and Schweppe, Petroleum Refiner, 1958.

VISCOSITY

UNITS :

Dynamic viscosity : 1 centipose = 0.01 dyne.sec/cm2 = 0.000672 lbm/ft sec

Kinematic viscosity : 1 centistoke = 0.01 cm2/s = $\frac{\text{Dynamic viscosity}}{\text{density}}$

Other quoted units for kinematic viscosity are :

Saybolt universal	Redwood	Engler
Saybolt furol	conversion charts are sited in literature	

VAPOUR VISCOSITY

- Use figure 12 in this section or
- Calculate using : i) $\mu_m = \frac{\sum \mu_i \, y_i \sqrt{Mw_i}}{\sum y_i \sqrt{Mw_i}}$

μ_m = mixture viscosity
μ_i = component viscosity
Mw_i = component mol.wt
y_i = component mol.frac
accuracy ± 5 %

ii) $\mu m = A \exp (B\rho'^{C})$

$$A = \frac{(9.4 + 0.02MW)\,T^{1.5}}{(209 + 19MW + T)\,10^4}$$

$$B = 3.5 + \frac{986}{T} + 0.01MW$$

$$C = 2.4 - 0.2B$$

$$\rho' = \frac{MW.P}{ZRT} \qquad g/cm^3$$

T in °R

$40 < T < 460$ °F

$14.7 < P < 10000$ psia

R = 669.8

LIQUID VISCOSITY

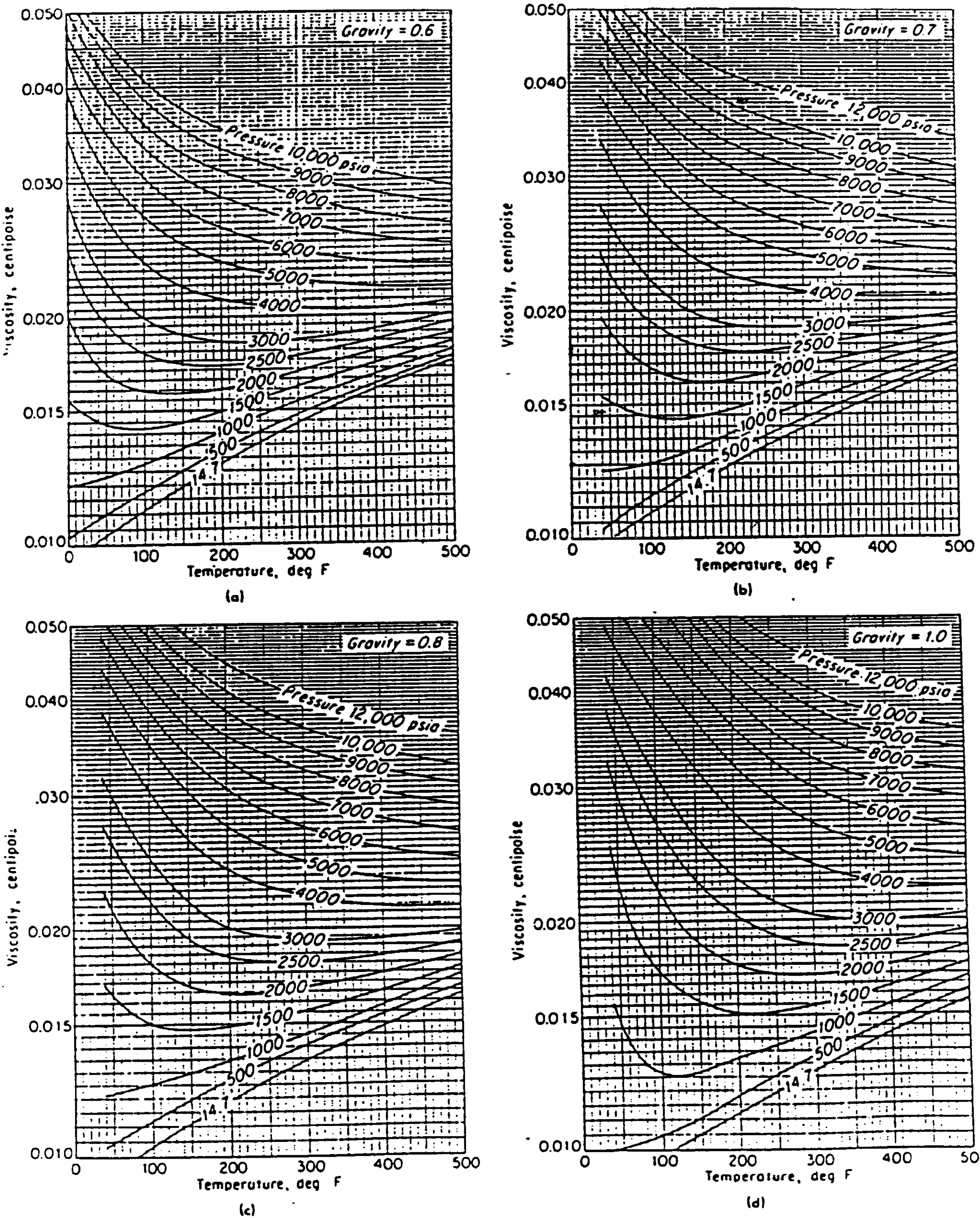
- Use Figure 13 in this section or : -
- Calculate using : i) $\mu_m = \left[\sum X_i \,(\mu_i)^{1/3}\right]^3$ X_i = component mol.frac
- The viscosity of crude oils with an API > 30 ° (sg = 0.88) can be estimated using :

$\log \mu = a - (0.035)(\text{API})$ centipoise

where

°C	a
38	2.05
54	1.83
71	1.55
88	1.30
104	1.08

- Correlations for liquid viscosity possess a general reliability of ± 15 %



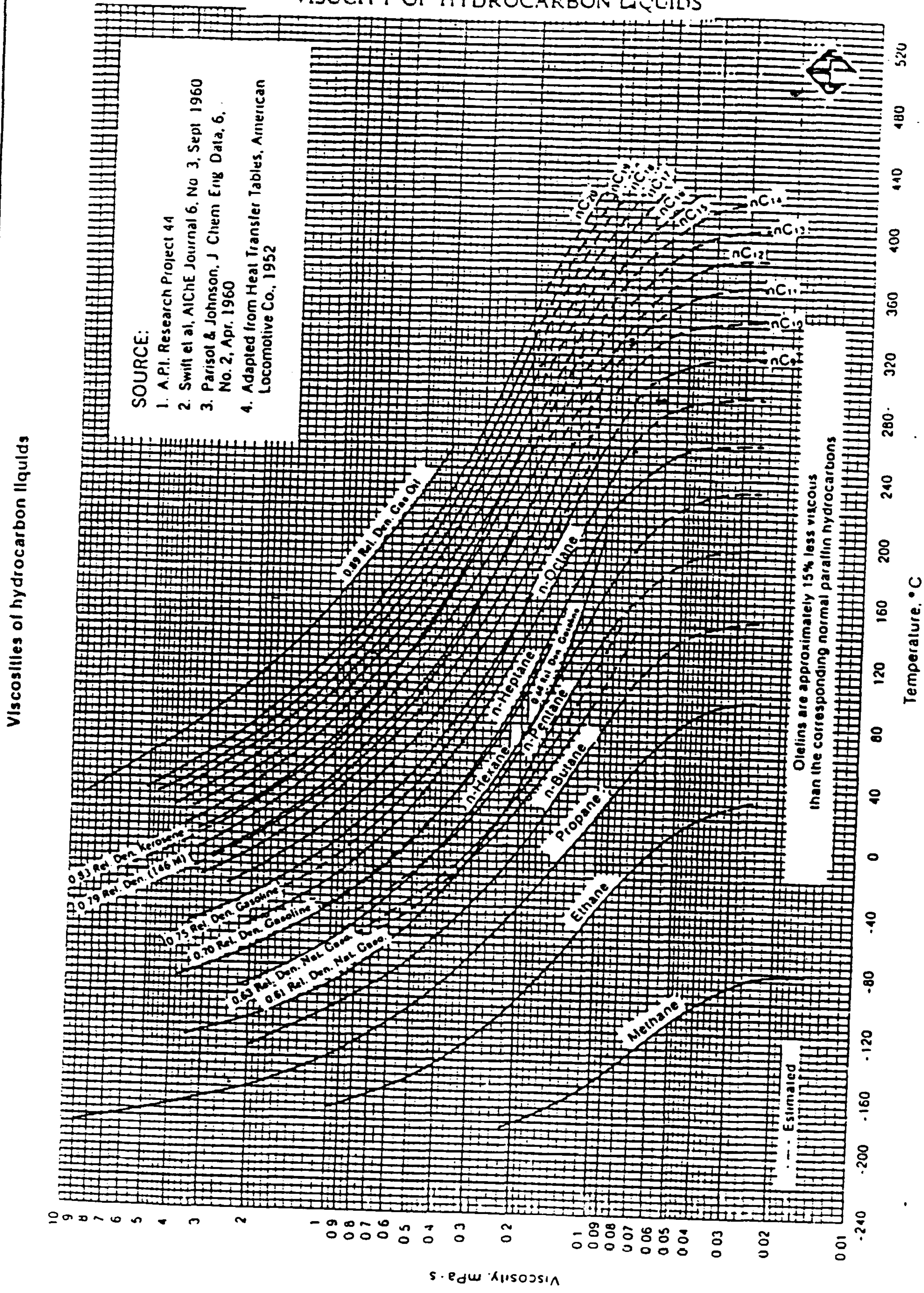
(a) (b) (c) (d)

FIG. 12 Viscosity of natural gases. (a) 0.6 gravity. (b) 0.7 gravity (c) 0.8 gravity (d) 1.0 gravity (2-42.)

VISCOSITY OF NATURAL GASES

FIG. 13

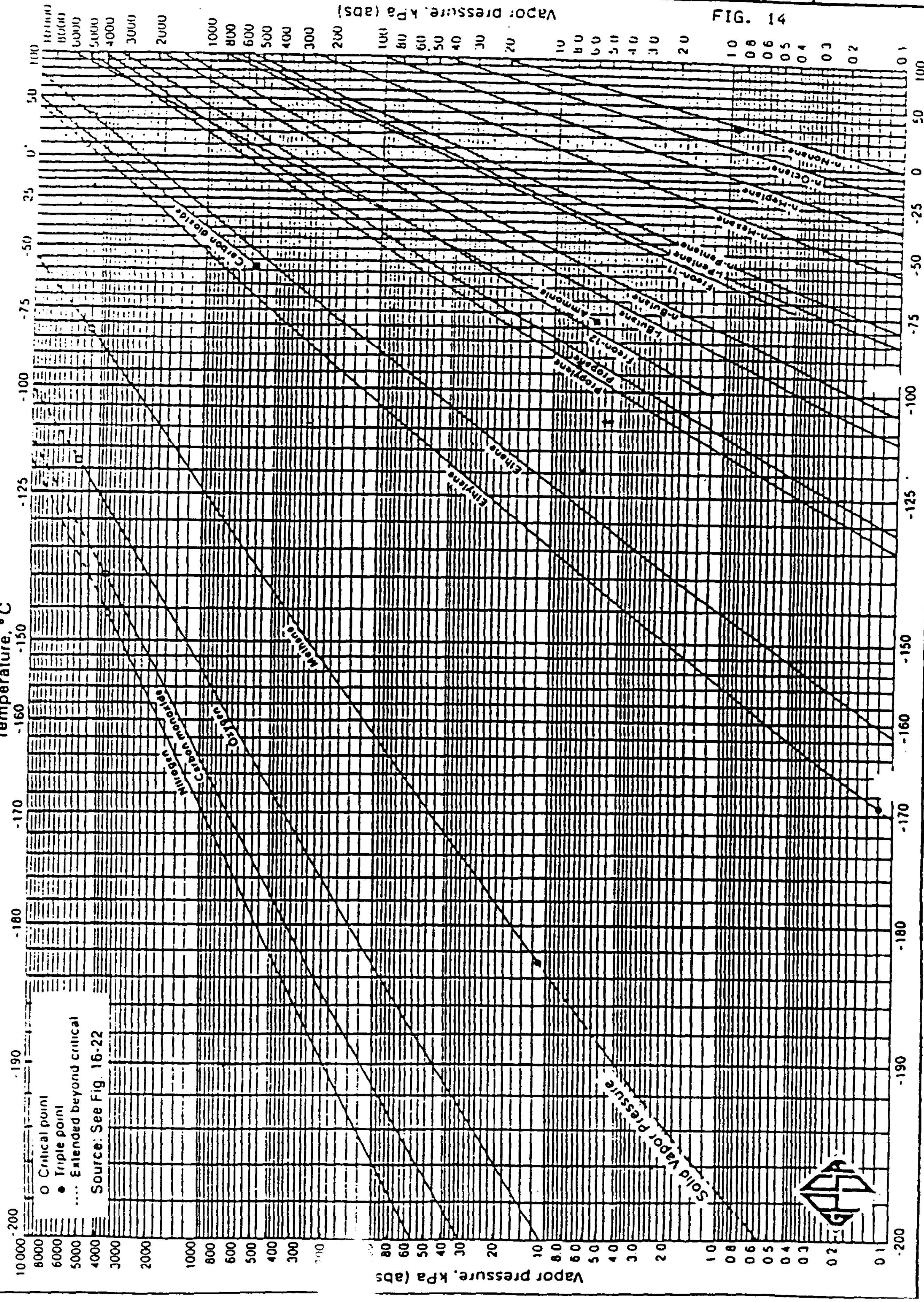
VISOCITY OF HYDROCARBON LIQUIDS



SOURCE:
1. A.P.I. Research Project 44
2. Swift et al, AIChE Journal 6, No. 3, Sept 1960
3. Parisol & Johnson, J Chem Eng Data, 6, No. 2, Apr. 1960
4. Adapted from Heat Transfer Tables, American Locomotive Co., 1952

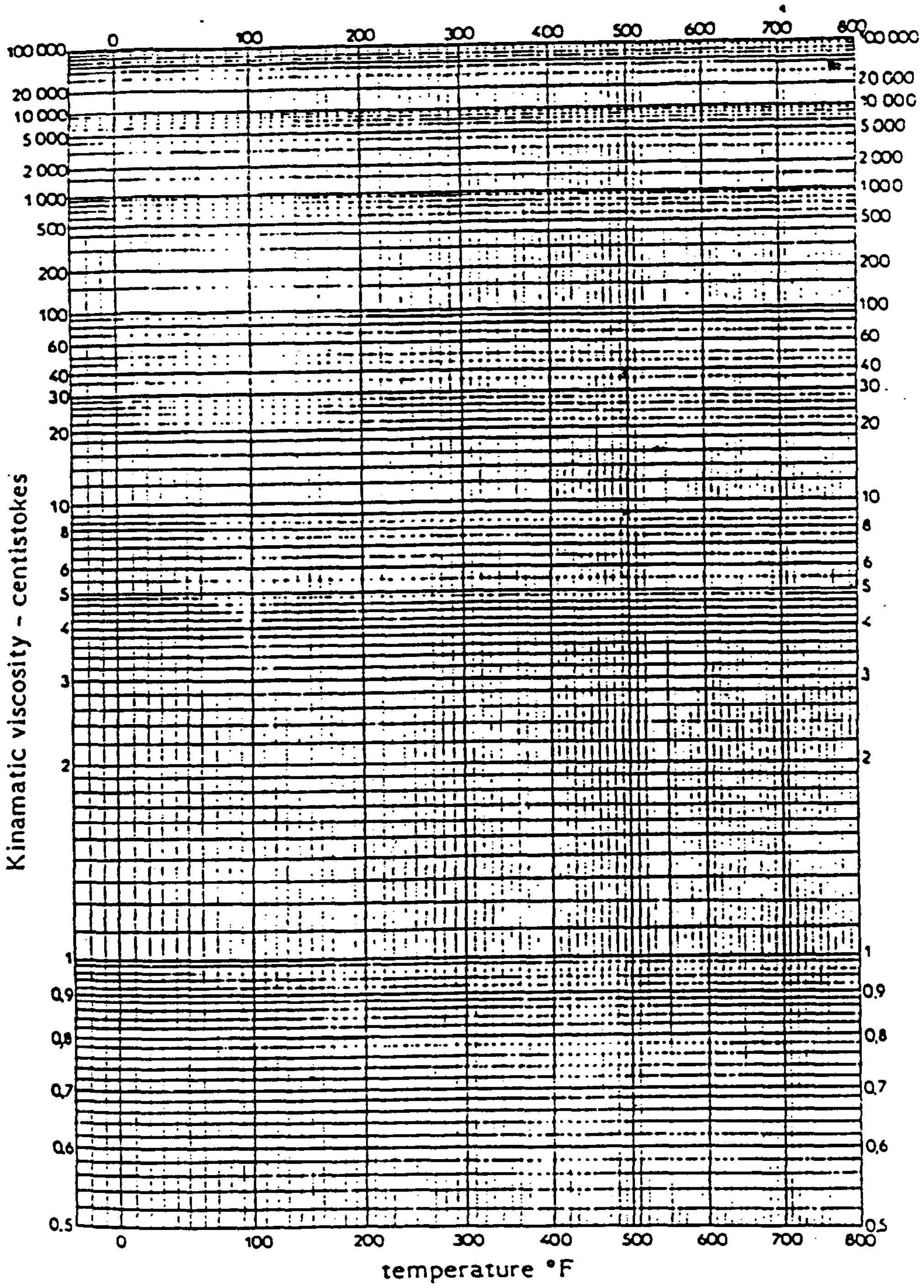
Olefins are approximately 15% less viscous than the corresponding normal paraffin hydrocarbons

FIG. 14



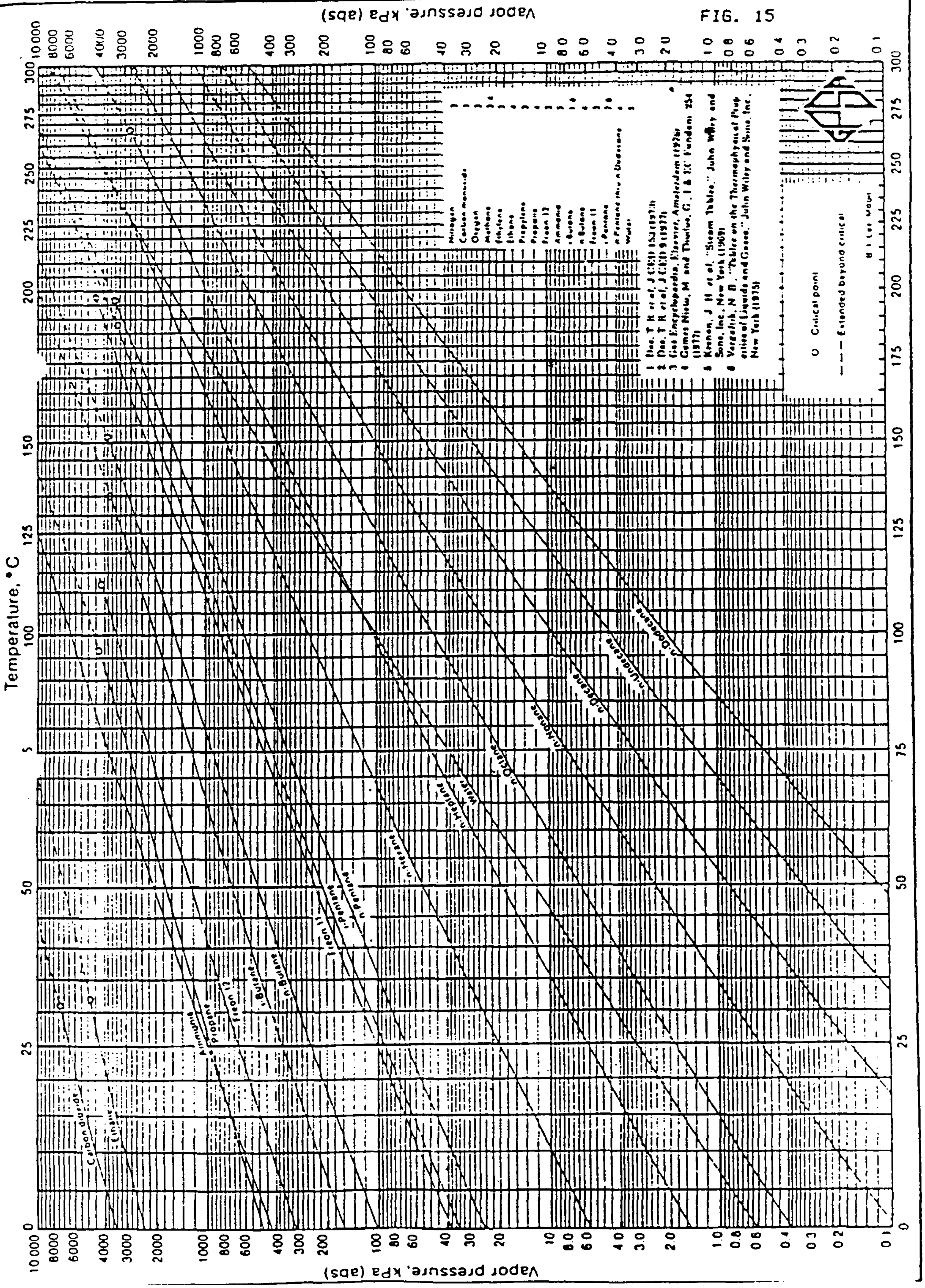
Source: See Fig. 16-22

Low-temperature vapor pressures for light hydrocarbons



ASTM VISCOSITY CHART

FIG. 15


Vapor pressure, kPa (abs)
10 000
8000
6000
4000
3000
2000
1000
800
600
400
300
200
100
80
60
40
30
20
10
8.0
6.0
4.0
3.0
2.0
1.0
0.8
0.6
0.4
0.3
0.2
0.1
Temperature, °C
0
25
50
75
100
125
150
175
200
225
250
275
300
Carbon dioxide
Ethane
Ammonia
Propane
Freon 12
i-Butane
n-Butane
Freon 11
i-Pentane
n-Pentane
n-Hexane
n-Heptane
Water
n-Octane
n-Nonane
n-Decane
n-Undecane
n-Dodecane
Nitrogen
Carbon monoxide
Oxygen
Methane
Ethylene
Ethane
Propylene
Propane
Freon 12
Ammonia
i-Butane
n-Butane
Freon 11
i-Pentane
n-Pentane thru n-Dodecane
Water
1 Das, T R et al, J CED 18 (1973)
2 Das, T R et al, J CED 9 (1977)
3 Gas Encyclopedia, Elsevier, Amsterdam (1976)
4 Gomez Nieto, M and Thodos, G, I & EC Fundam 254 (1977)
5 Keenan, J H et al, "Steam Tables," John Wiley and Sons, Inc., New York (1969)
6 Vargaftik, N B, "Tables on the Thermophysical Properties of Liquids and Gases," John Wiley and Sons, Inc., New York (1975)
o Critical point
— — — Extended beyond critical

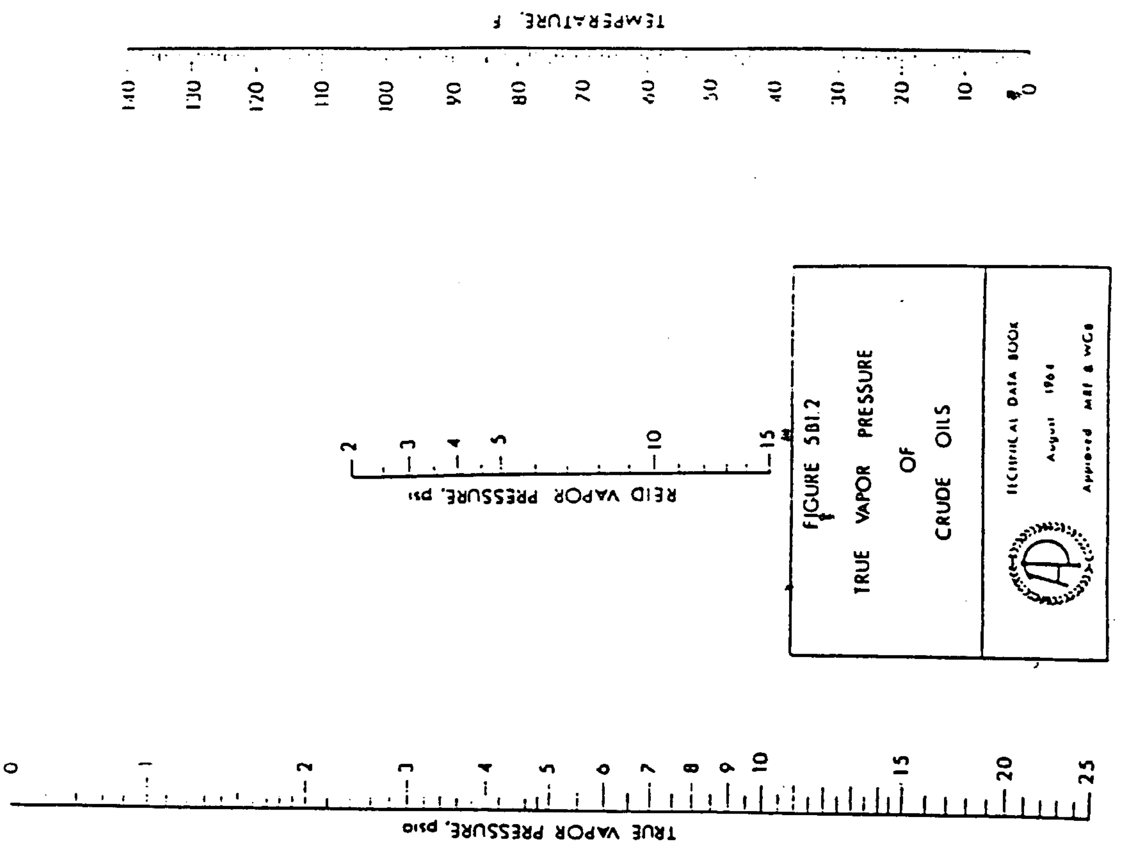


FIGURE 5B1.2
TRUE VAPOR PRESSURE
OF
CRUDE OILS

TECHNICAL DATA BOOK
August 1964
Approved MRI & WGB

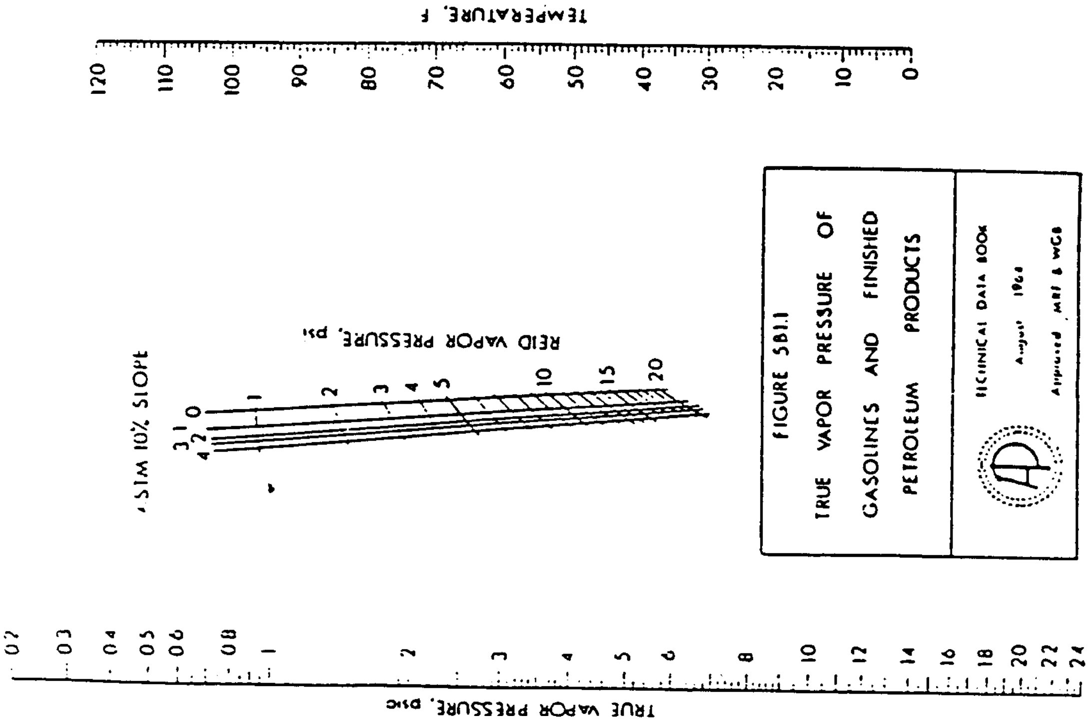
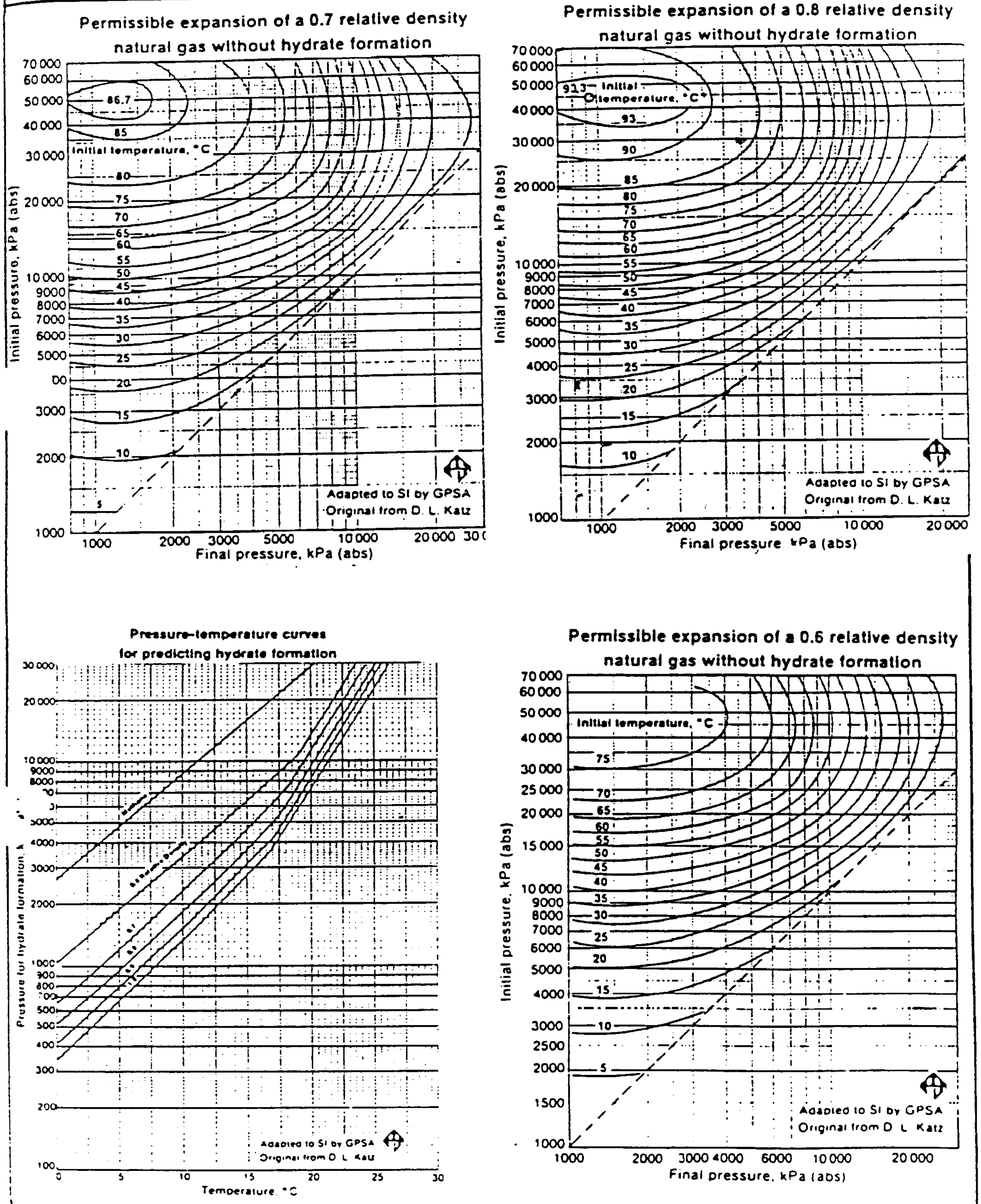


FIGURE 5B1.1
TRUE VAPOR PRESSURE OF
GASOLINES AND FINISHED
PETROLEUM PRODUCTS

TECHNICAL DATA BOOK
August 1964
Approved MRI & WGB

FIG. 16 TRUE VAPOUR PRESSURES OF PETROLEUM PRODUCTS AND CRUDE OIL



Permissible expansion of a 0.7 relative density natural gas without hydrate formation

Permissible expansion of a 0.8 relative density natural gas without hydrate formation

Pressure-temperature curves for predicting hydrate formation

Permissible expansion of a 0.6 relative density natural gas without hydrate formation

Adapted to SI by GPSA
Original from D. L. Katz

FIG. 17 HYDRATE FORMATION

SPECIFIC HEATS

(HEAT CAPACITY)

UNITS : BTU/LB °F   1 BTU/lb °F = 4.19 KJ/Kg °C
KJ/KG °C   1 BTU/lb °F = 1 cal/g °C

VAPOUR MIXTURES

- Use figs 18, 19 in this section
- Cp° is a fuction of temperature and can be calculated using:

$$Cp^\circ = A + BT + CT^2$$

where A, B, C are constants dependant on system composition
and T is in °R (K)
Values of A, B, C are cited in Kern, or Perry.

- Cp° can be corrected for pressure if Pr and Tr are known using Figure
- K = ratio of specific heats $\frac{Cp}{Cv}$ this should also be corrected for pressure if required.

LIQUID MIXTURES

- Use Figure 21 in this section or :
- Calculate using

$$Cpl = 2.96 - 1.34\,G + T\,(0.00620 - 0.002349) \quad \text{KJ/KG °C} \quad (T \text{ in °C})$$

$$Cpl = 0.68 - 0.31\,G + T\,(0.00082 - 0.000319) \quad \text{BTU/LB °F} \quad (T \text{ in °F})$$

G = liquid specific gravity   CAMPBELL
(accuracy ± 5 %)

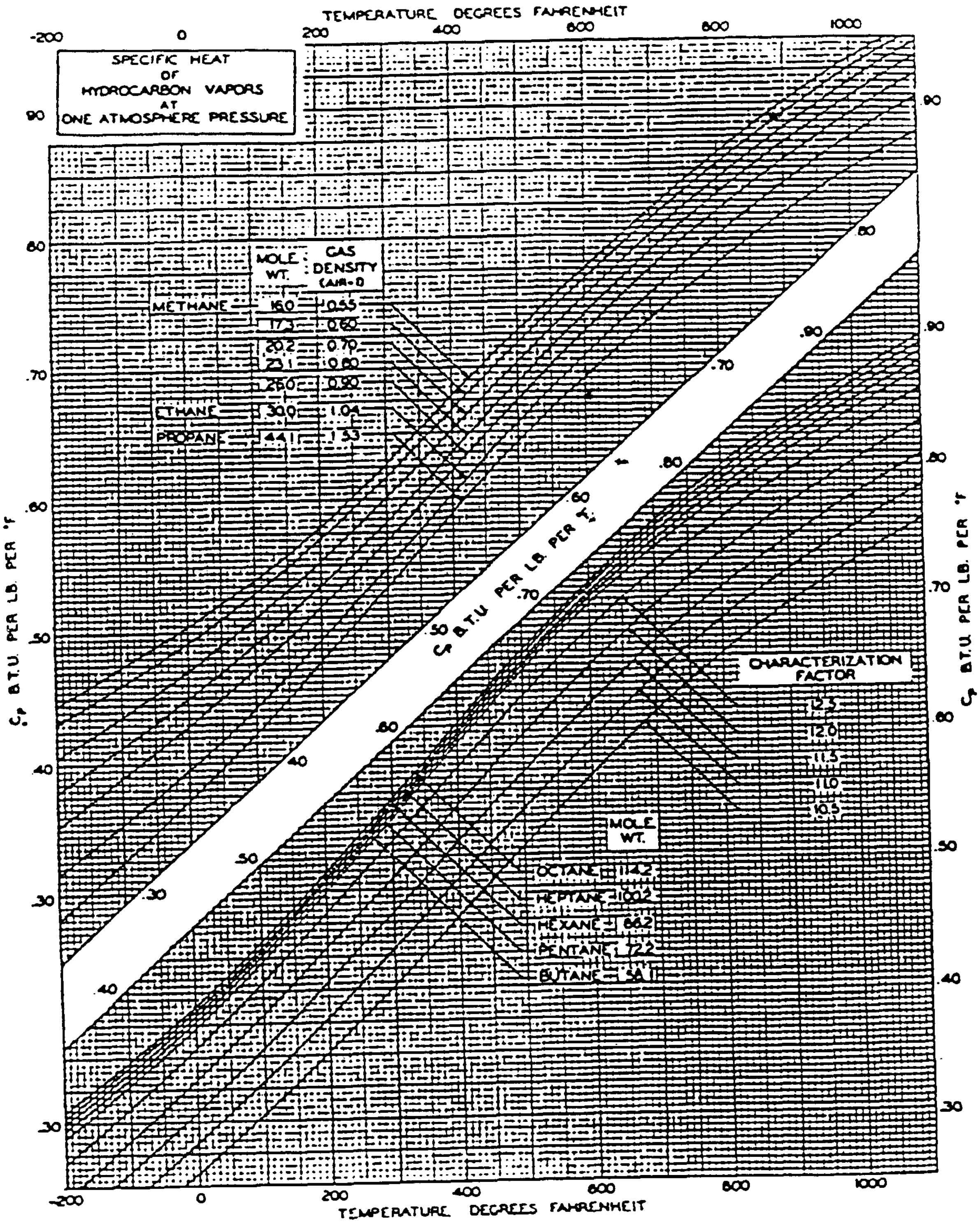


FIG. 18 SPECIFIC HEAT OF HYDROCARBON VAPOURS AT 1 ATM (NOTE UNITS ARE BTU/LB/°F)

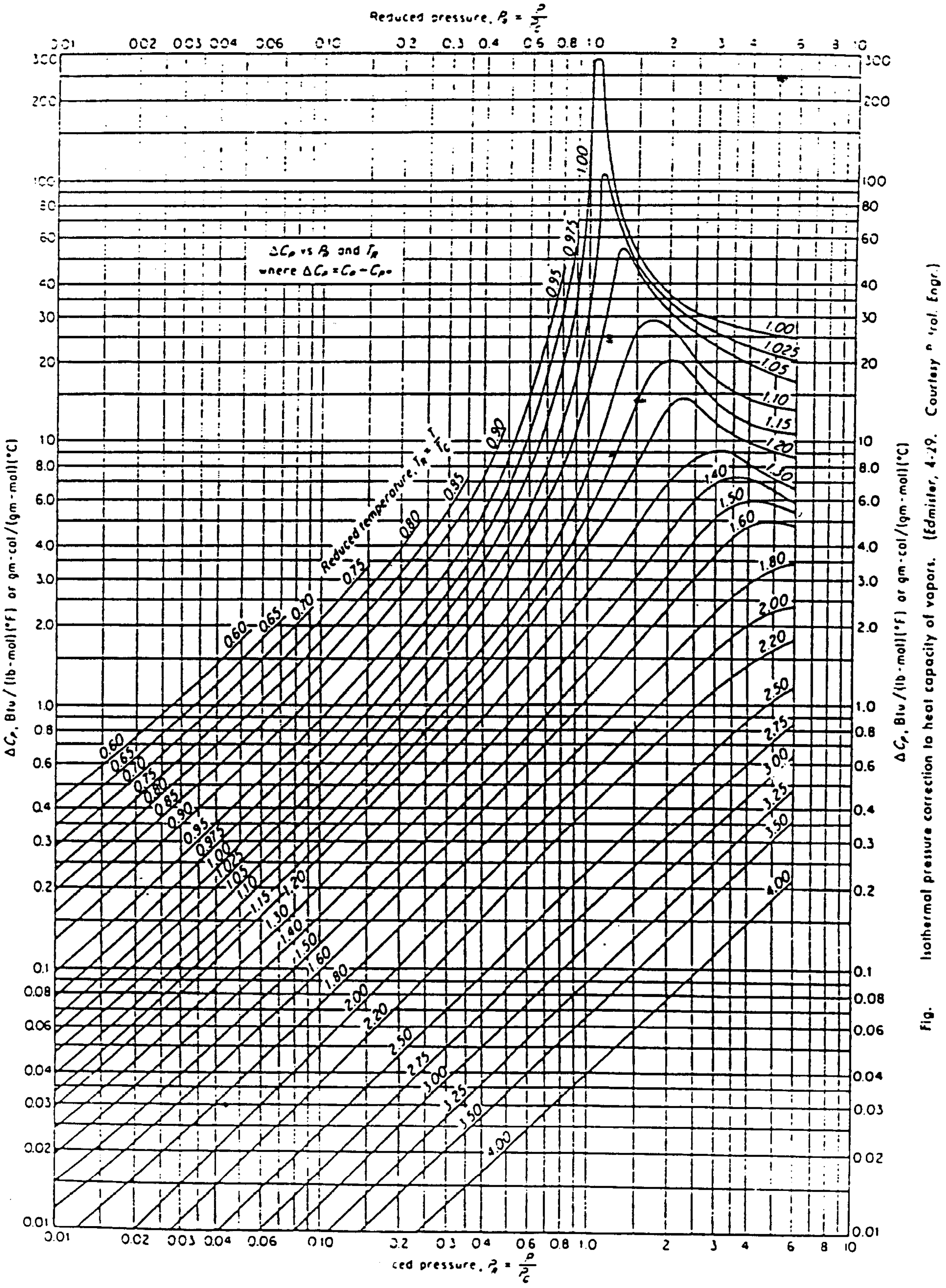


FIG. 19 HEAT CAPACITY CC CTION FACTORS (NOTE UNITS ARE BTU/LB MOLE/°F) (at atmosphe ssure)

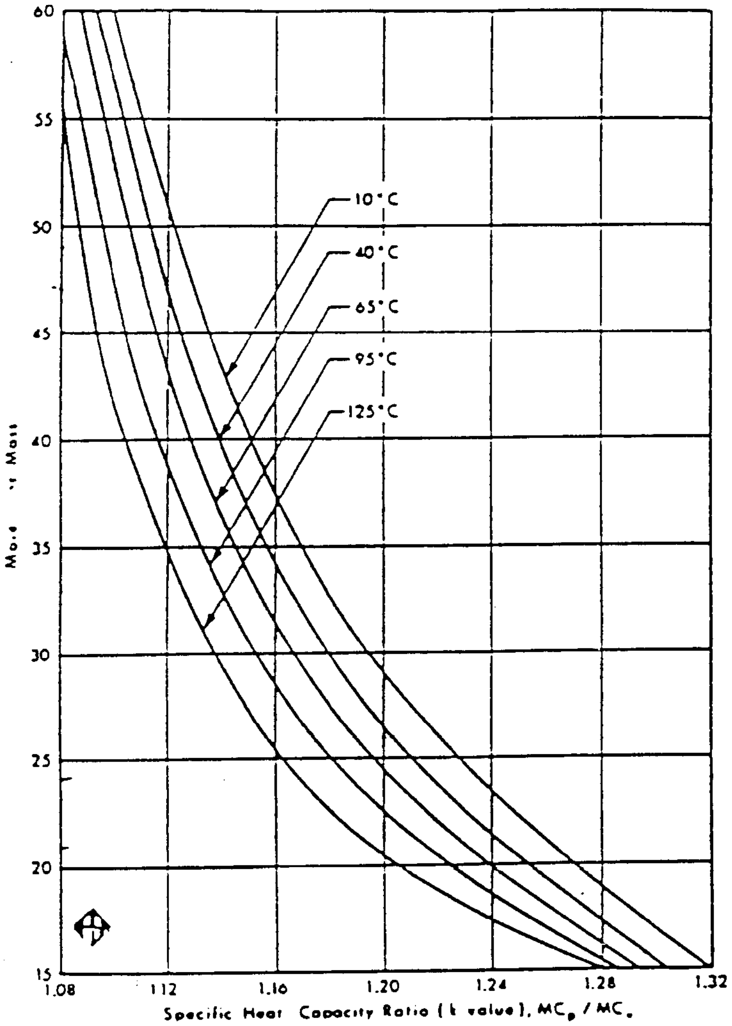
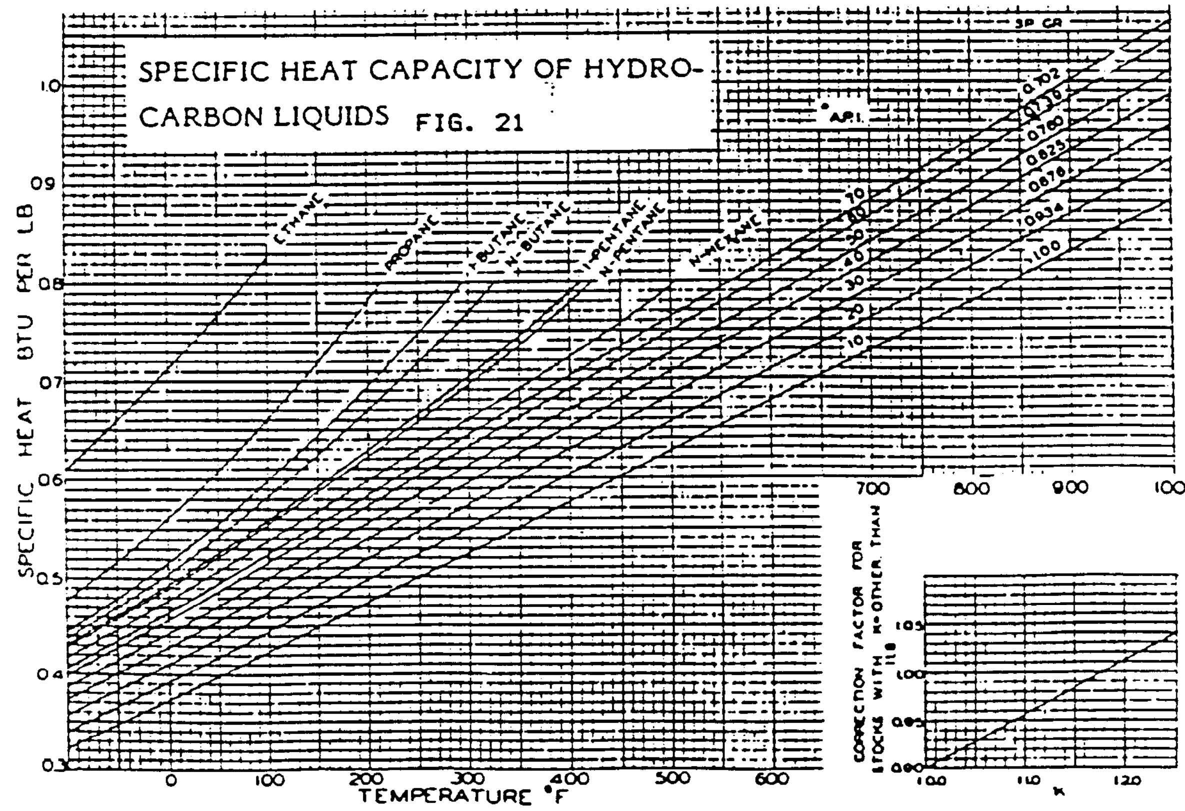


FIG. 20

Approximate specific heat capacity ratios of hydrocarbon gases



SPECIFIC HEAT CAPACITY OF HYDROCARBON LIQUIDS FIG. 21

THERMAL CONDUCTIVITY

k

UNITS : BTU/LB °F   1 BTU/lb °F = 1.488 Kcal/m.h.°C

VAPOUR MIXTURES

- Use figs 22, 23 in this section
- Low pressure thermal conductivities of pure gases and vapours can be estimated using :

$$k = \mu \left(C_p + \frac{2.48}{M} \right)$$

accuracy ± 8 %

k - BTU/hr.ft°F
μ - lb/hr.ft
Cp - BTU/lb °F

LIQUIDS

- Use fig 24 in this section or :
- Liquid hydrocarbon mixtures can be estimated using :

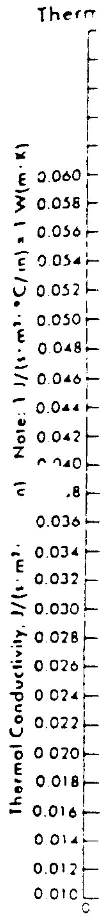
$$k = \frac{0.0677}{sg} \left[1 - 0.0003 (T - 32) \right]$$

accuracy ± 12 %

k - BTU/hr.ft°F
sg - specific gravity .78 < > .95
T - °F 32 < > 392

SOLIDS

- See Perry of Kern for details of metals, earths and building materials.



Thermal Conductivity of Natural Gases at 101.3250 kPa (abs)

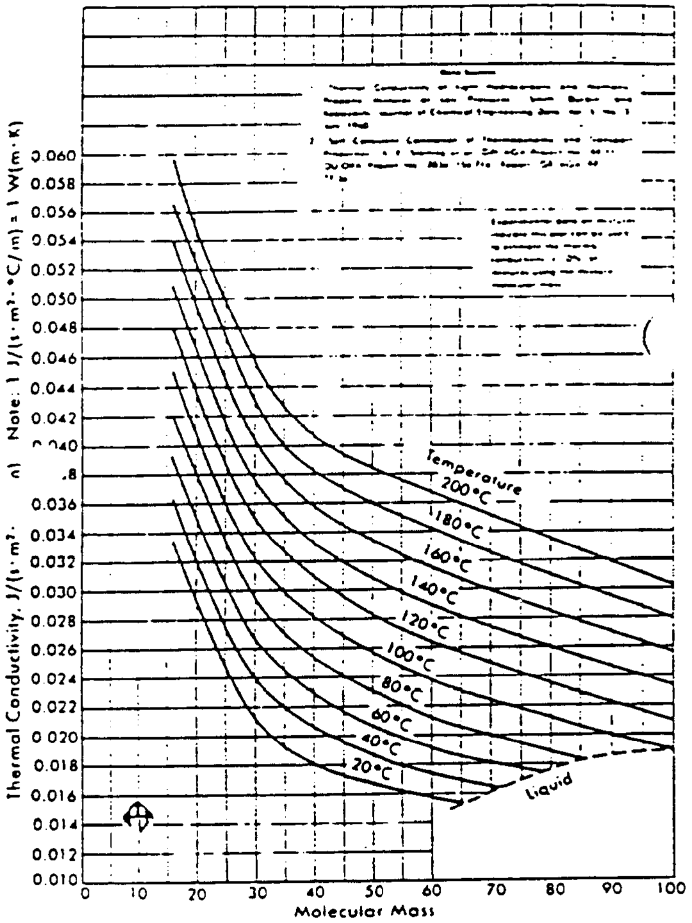


FIG. 22

Thermal conductivity ratio for gases

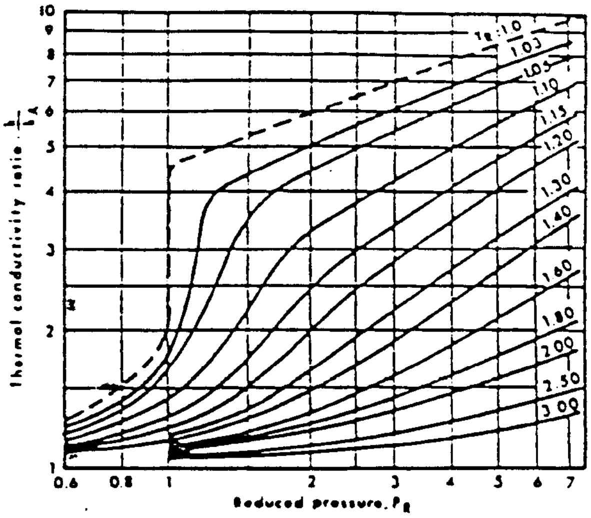
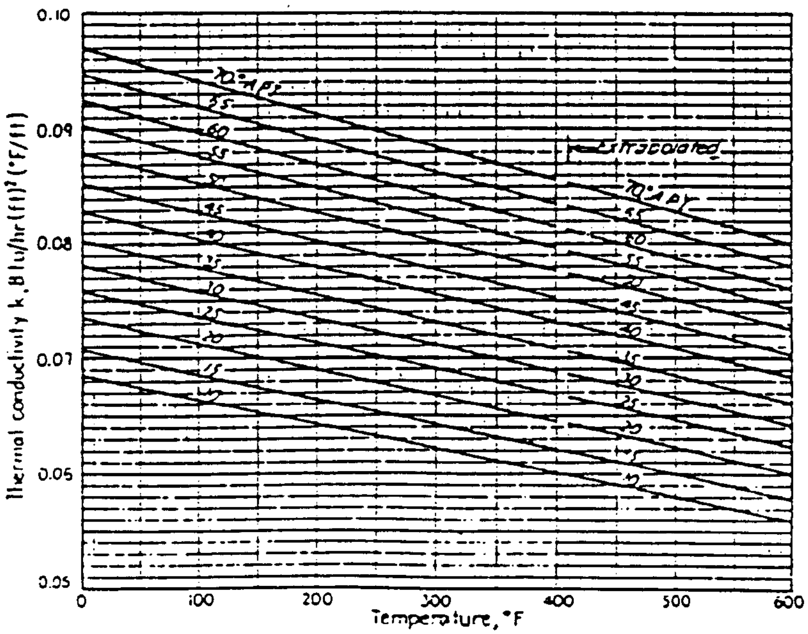


FIG. 23



(Adapted from Natl. Bur. Standards)

THERMAL CONDUCTIVITIES OF HYDRO-
CARBON LIQUIDS

FIG. 24

LATENT HEAT OF VAPOURISATION

UNITS : BTU/LB 1 BTU/lb = 0.5556 Kcal/Kg
Kcal/Kg

- Use figures : 25, 26

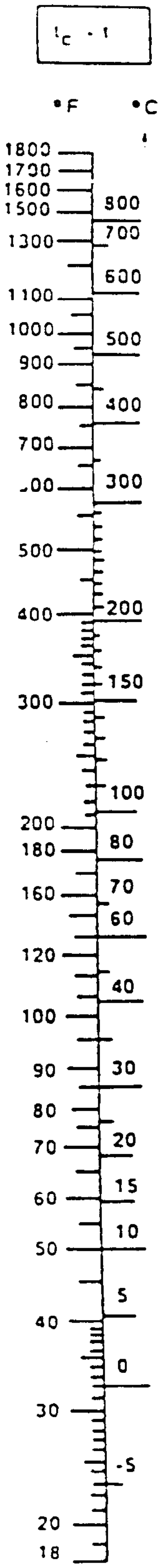
- Estimate using Troutons rule :

$$\underline{\underline{\lambda = 21 . T_b}} \quad \text{cal/gmole}$$

accuracy ± 20 % Tb = boiling point °K

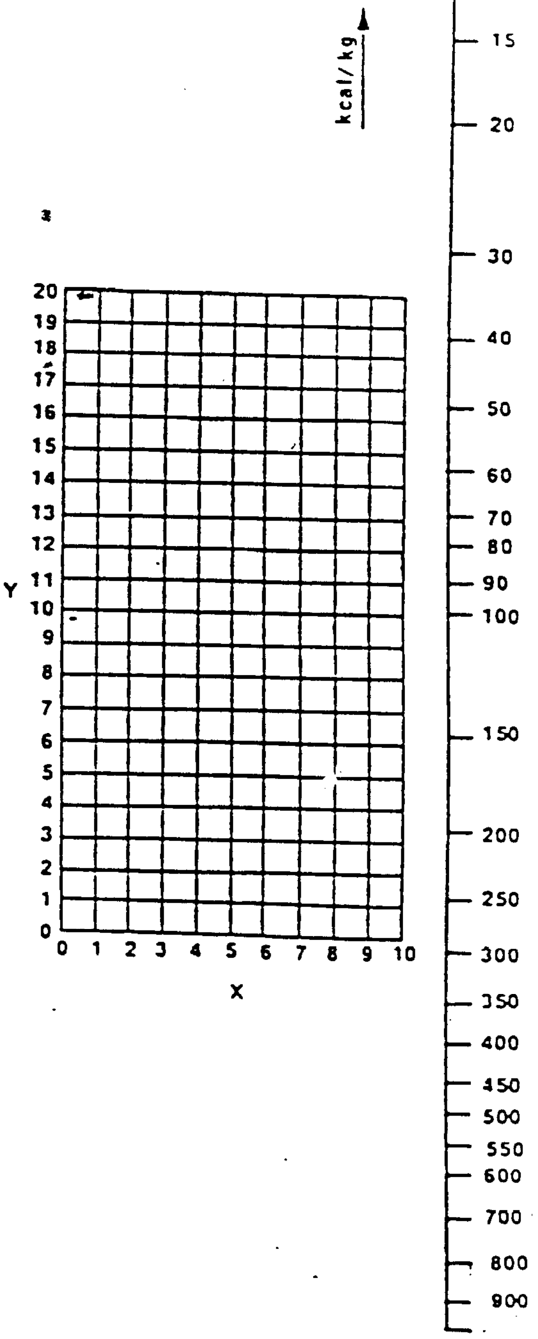
- For relief valve calculations use 50 BTU/lb if actual Lt.ht is not known.

- Detailed estimation methods in Perry : pp 238



LIQUID	t °C	RANGE t_c - t °C	X	Y
Acetic Acid	321	100-200	5.6	11.9
Acetone	235	140-240	7.0	10.3
Ammonia	133	80-200	3.2	3.8
Amyl alcohol (iso)	307	200-300	6.0	9.4
Benzene	283	10-300	3.6	12.5
Butane (-n)	152	40- 70	2.6	11.6
Butane		70-200	3.6	11.7
Butane (-iso)	134	75-175	3.4	12.1
Butyl alcohol (-n)	297	170-300	2.0	9.8
Butyl alcohol (-iso)	265	150-200	1.7	9.7
Butyl alcohol		200-270	6.9	7.7
Butyl alcohol (.sec)	265	170-270	6.6	8.3
Butyl alcohol (.tert)	235	100-200	3.9	9.5
Carbon dioxide	33	10-100	3.3	11.1
Carbon disulfide	272	140-275	3.5	13.7
Carbon tetrachloride	283	10-300	3.6	17.3
Chlorine	143	100-200	1.5	14.5
Chloroform	268	175-265	3.7	15.7
Dichloroethylene (-cis)	242	200-300	9.4	13.3
Dimethyl amine	166	125-200	4.3	8.8
Diphenyl	528	10- 32	2.2	15.2
Diphenyl		32-150	3.8	15.2
Diphenyl		150-400	0.8	12.8
Diphenyl oxide	511	50-340	3.1	15.5
Diphenyl oxide		340-500	6.2	14.5
Ethane	480	10-130	4.0	9.8
Ethyl alcohol	243	10-140	3.1	7.0
Ethyl alcohol		140-250	4.7	6.3
Ethyl amine	183	130-230	3.9	9.0
Ethyl chloride	188	150-230	4.1	12.2
Ethylene	10	10- 50	3.0	9.3
Ethylene		50-125	4.0	0.6
Ethyl ether	194	15-130	3.1	12.7
Ethyl ether		130-240	1.8	12.7
Freon 11 (CCl-F)	198	70-250	3.6	17.3
Freon 12 (CCl-F)	111	60-150	3.9	17.2
Freon 21 (CHCl-F)	163	20-220	3.3	15.4
Freon 22 (CHCl)	96	50-160	4.0	15.1
Freon 113 (CCl-FCClF)	214	40-250	3.5	13.7
Freon 114 (CCl-FCClF)	145	45-200	3.5	13.7
Heptane (-n)	267	10-270	3.4	13.5
Hexane (-n)	236	55-240	3.4	13.2
Methane	47	10- 90	5.2	8.3
Methyl alcohol	240	20-140	3.3	5.3
Methyl acohol		140-240	3.8	4.7
Methyl amine	157	100-200	4.1	6.5
Methyl chloride	143	16-110	2.6	11.1
Methyl chloride		110-120	5.2	11.2
Methyl formate	214	150-250	1.9	11.3
Methylene chloride	216	150-250	1.0	13.7
Nitrous oxide	36	6- 25	1.2	9.2
Nitrous oxide		25-125	5.8	12.3
Octane (-n)	296	16-300	3.0	13.6
Pentane (-n)	197	15-250	3.3	12.7
Pentane (-iso)	188	10-200	3.2	12.7
Propane	56	15-252	4.3	11.0
Propyl alcohol (-n)	264	25-270	2.1	8.8
Propyl alcohol (-iso)	236	150-250	3.5	6.3
Pyridine	344	230-350	2.3	12.5
Sulfur dioride	158	100-200	2.0	12.3
Toluene	321	100-300	1.5	13.7
Trichloroethylene	271	180-305	6.0	15.8
Water	375	10-355	3.0	1.0

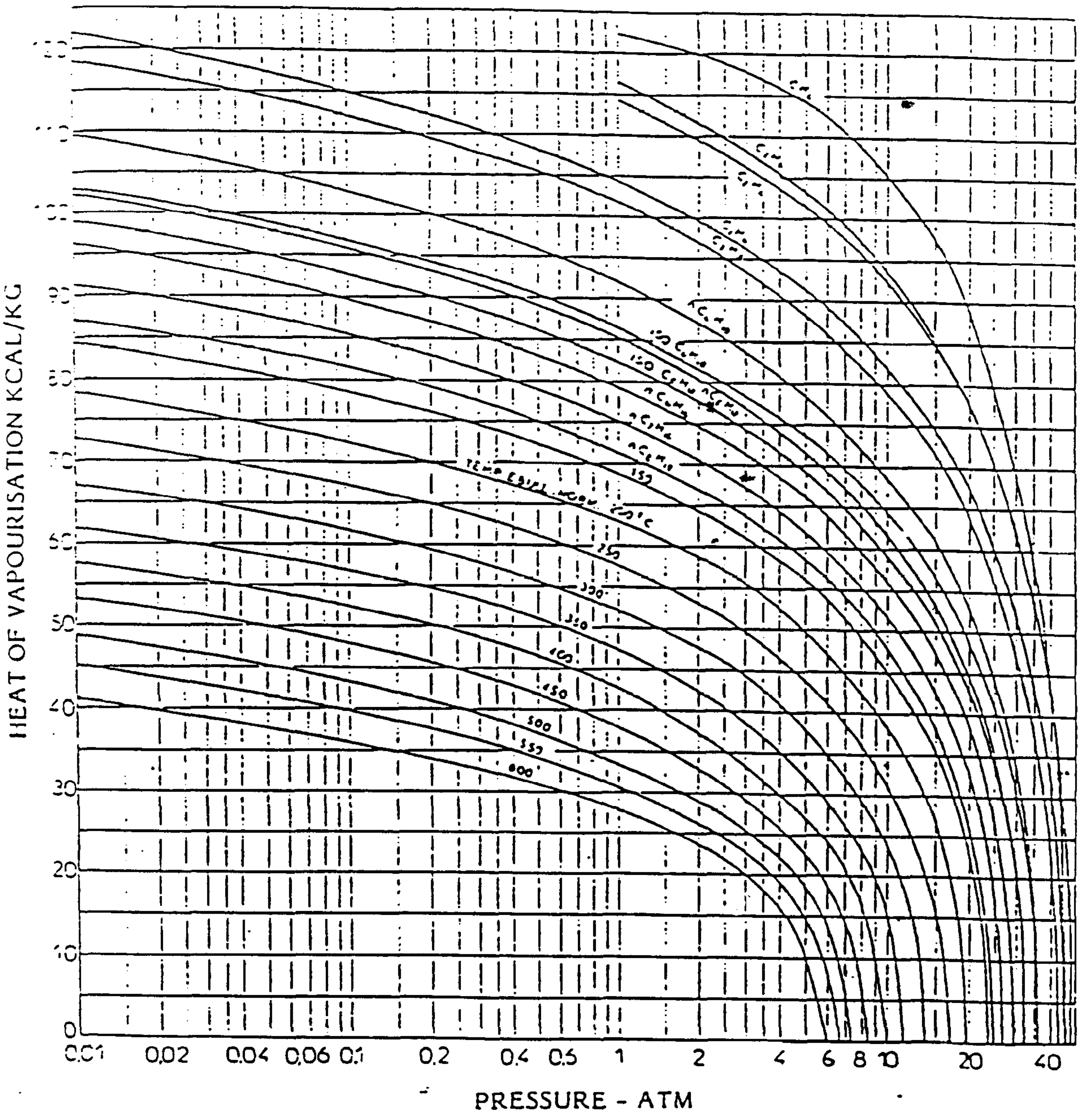
PRECISION : 10%



Exemple : Pour l'eau à 100°C, $t_c - t = 275°$

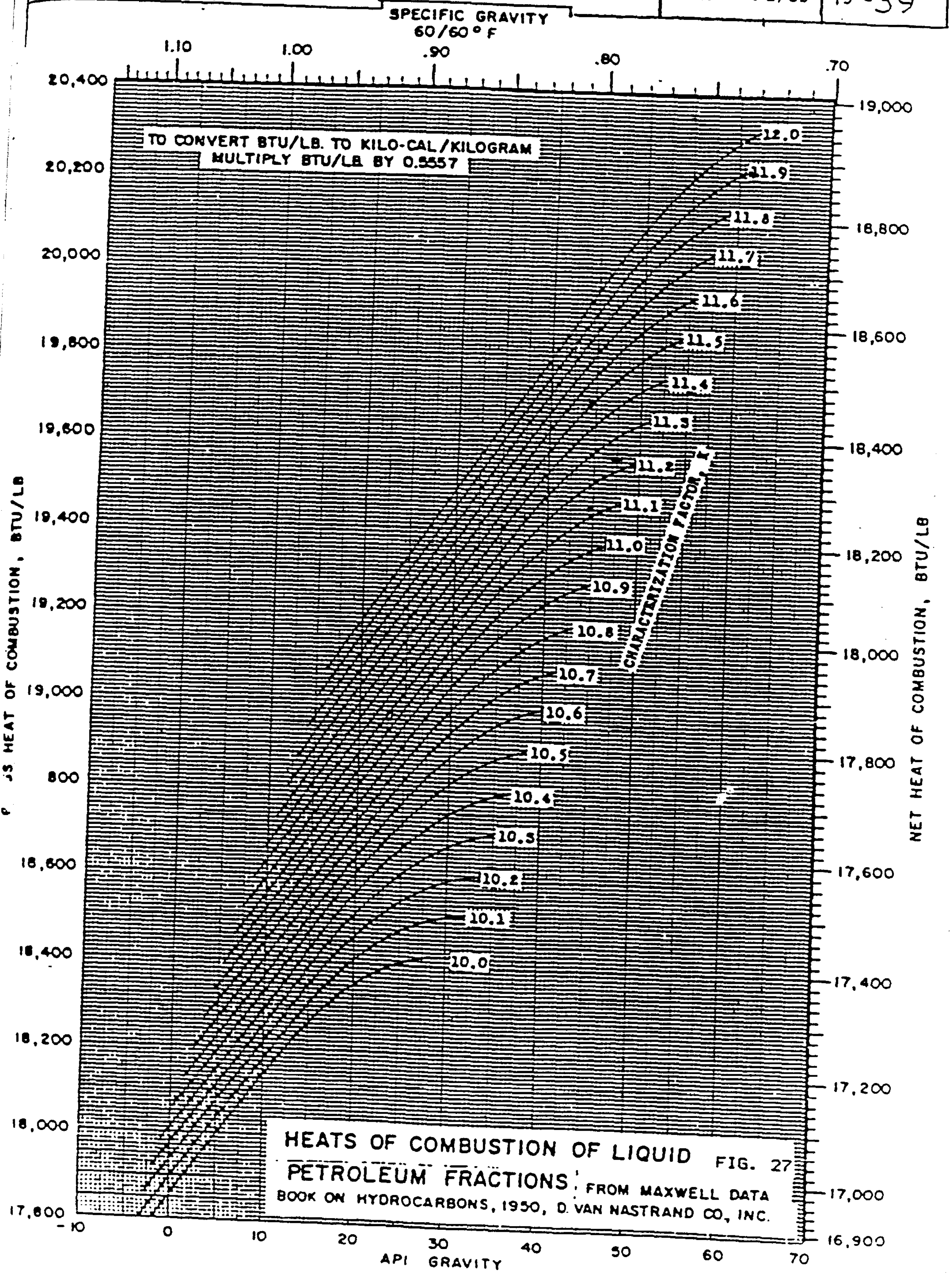
—— d_{H20} 538 kcal/ kg

LATENT HEATS OF VAPORIZATION OF VARIOUS LIQUIDS FIG. 25



LATENT HEAT OF VAPOURISATION OF HYDROCARBONS

FIG. 26



TO CONVERT BTU/LB. TO KILO-CAL/KILOGRAM MULTIPLY BTU/LB BY 0.5557

HEATS OF COMBUSTION OF LIQUID PETROLEUM FRACTIONS FIG. 27

FROM MAXWELL DATA BOOK ON HYDROCARBONS, 1950, D. VAN NASTRAND CO., INC.

SURFACE TENSIONS

UNITS : Dynes/cm 1 dyne/cm = 10^{-3} N/m
N/m

- For surface tensions of paraffins use fig. 28

- To estimate surface tensions for hydrocarbon liquids/gas use

$$\sigma = \left[\frac{P}{MW} \cdot \frac{\rho_L - \rho_v}{62.4}\right]^4$$

source : baker
accuracy : ± 10 %

P = Parachor = 18.07 + 2.946 MW for paraffins with MW < 100
= 278 + 2.35 (MW - 100) for paraffins with MW > 100

ρl = liquid density lb/ft^3
ρv = vapour density lb/ft^3

- For oil-gas mixtures can also use

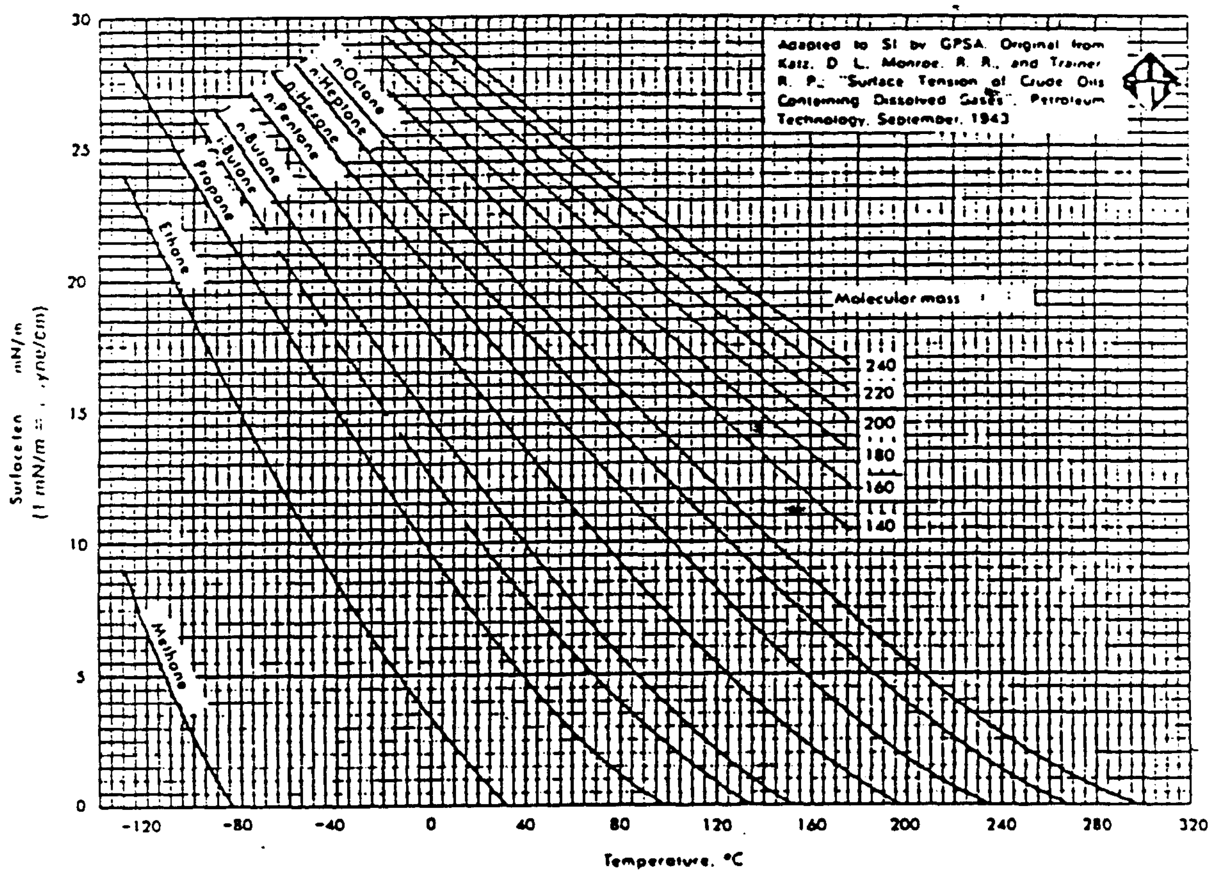
$$\sigma = (42.4 - 0.047\,T - 0.267\,API) \times \exp(-0.0007\,P)$$

T = temperature in °F
P = pressure in psia

source: Beggs + Brill

Surface tension of paraffin hydrocarbons

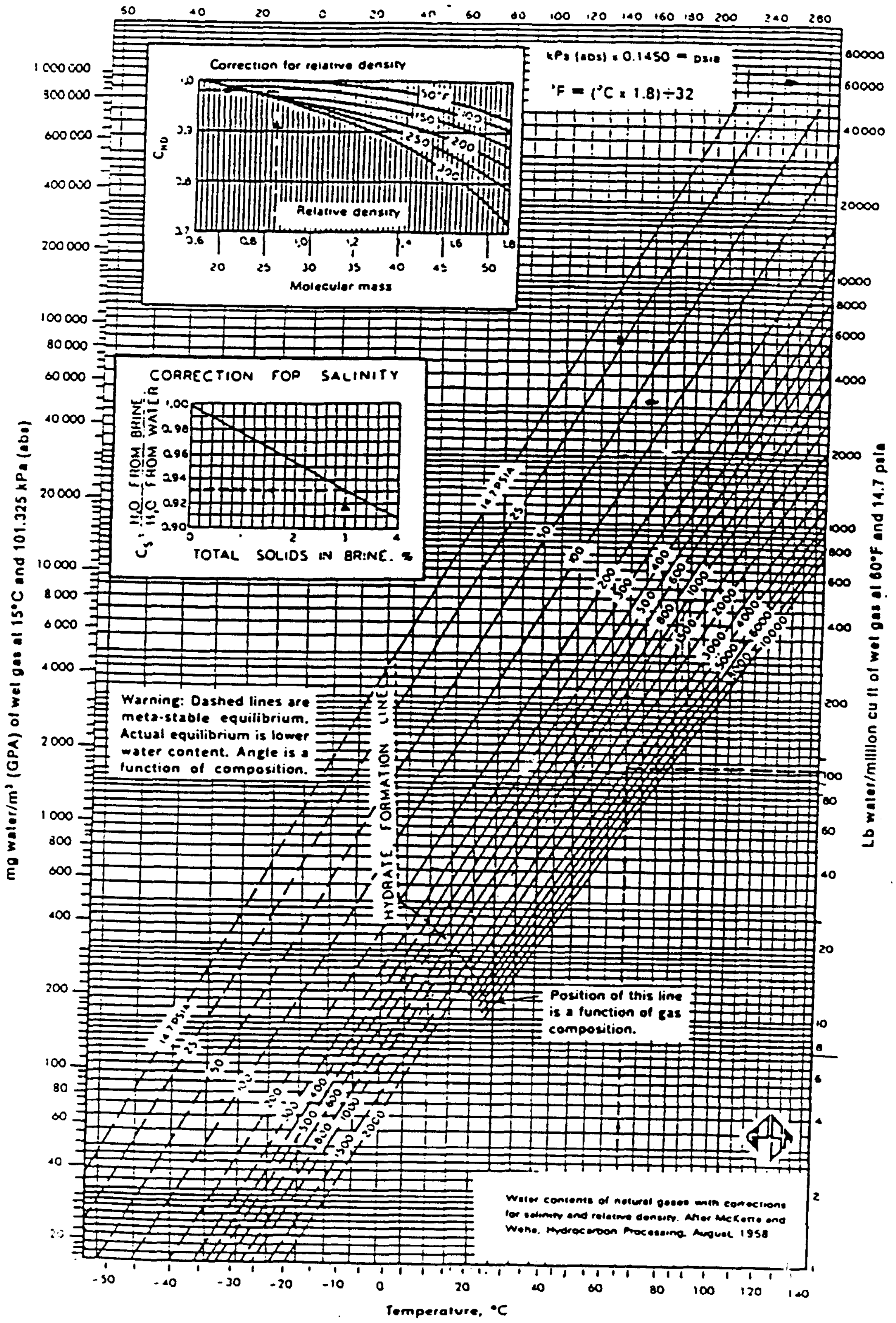
FIG. 28



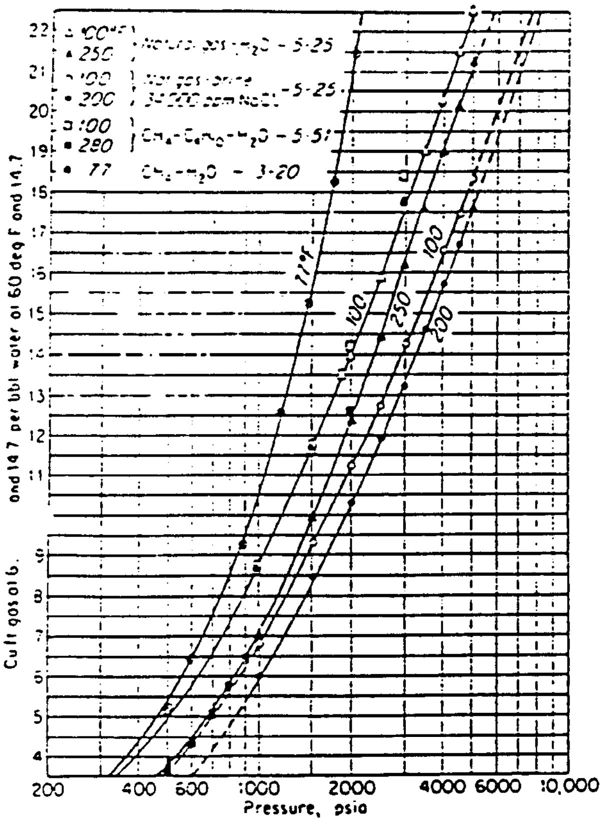
Adapted to SI by GPSA. Original from Katz, D. L., Monroe, R. R., and Trainer R. P., "Surface Tension of Crude Oils Containing Dissolved Gases", Petroleum Technology, September, 1943

Dew point of natural gas

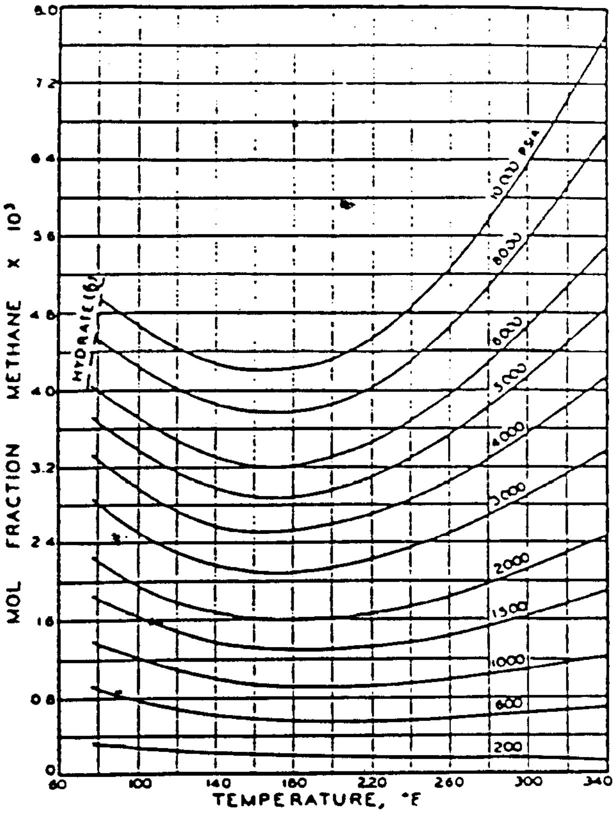
FIG. 29



Water contents of natural gases with corrections for salinity and relative density. After McKetta and Wehe, Hydrocarbon Processing, August 1958



Solubility of natural gases in water and brine. FIG. 30



Solubility of methane in water.

FIG. 31

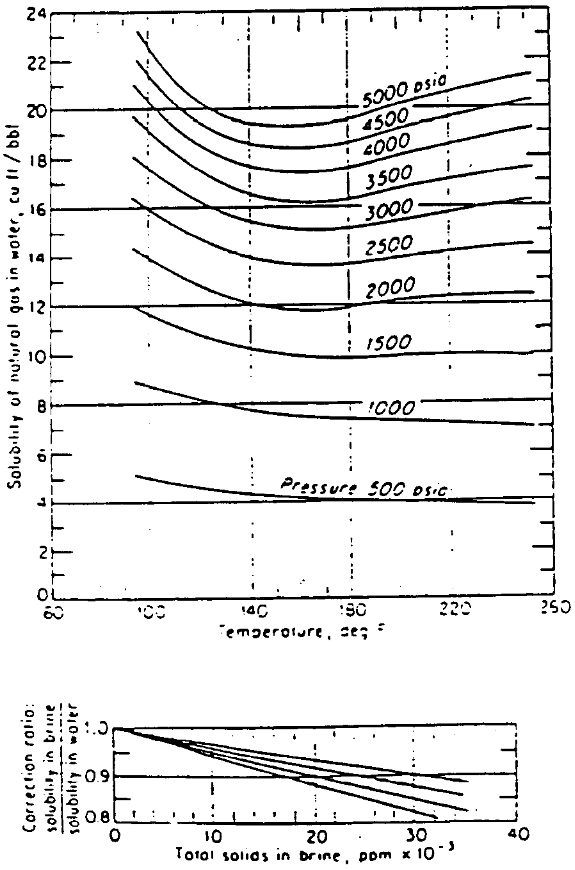
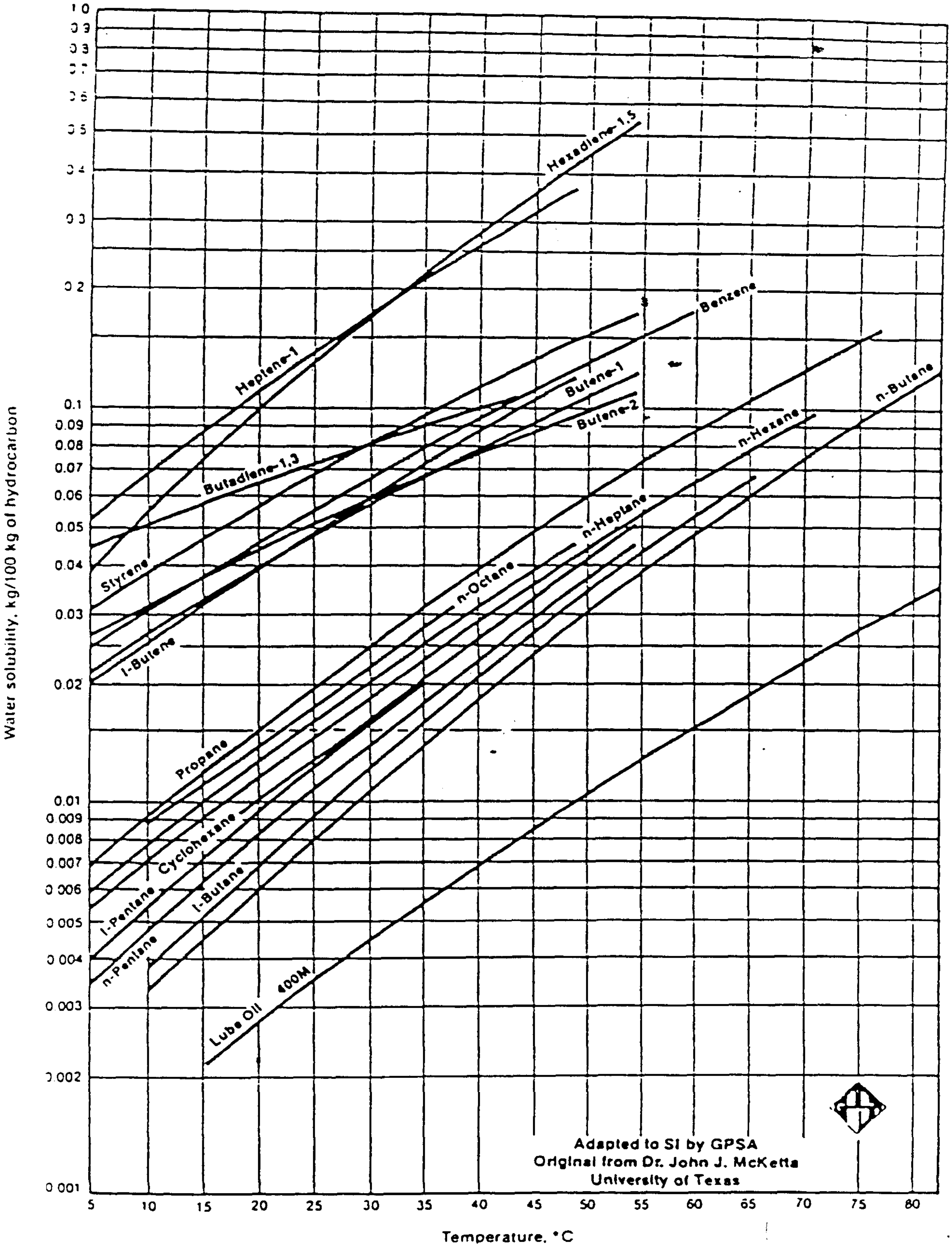


FIG. 32 , Solubility of natural gas in water.

SOLUBILITY OF NATURAL GAS IN WATER

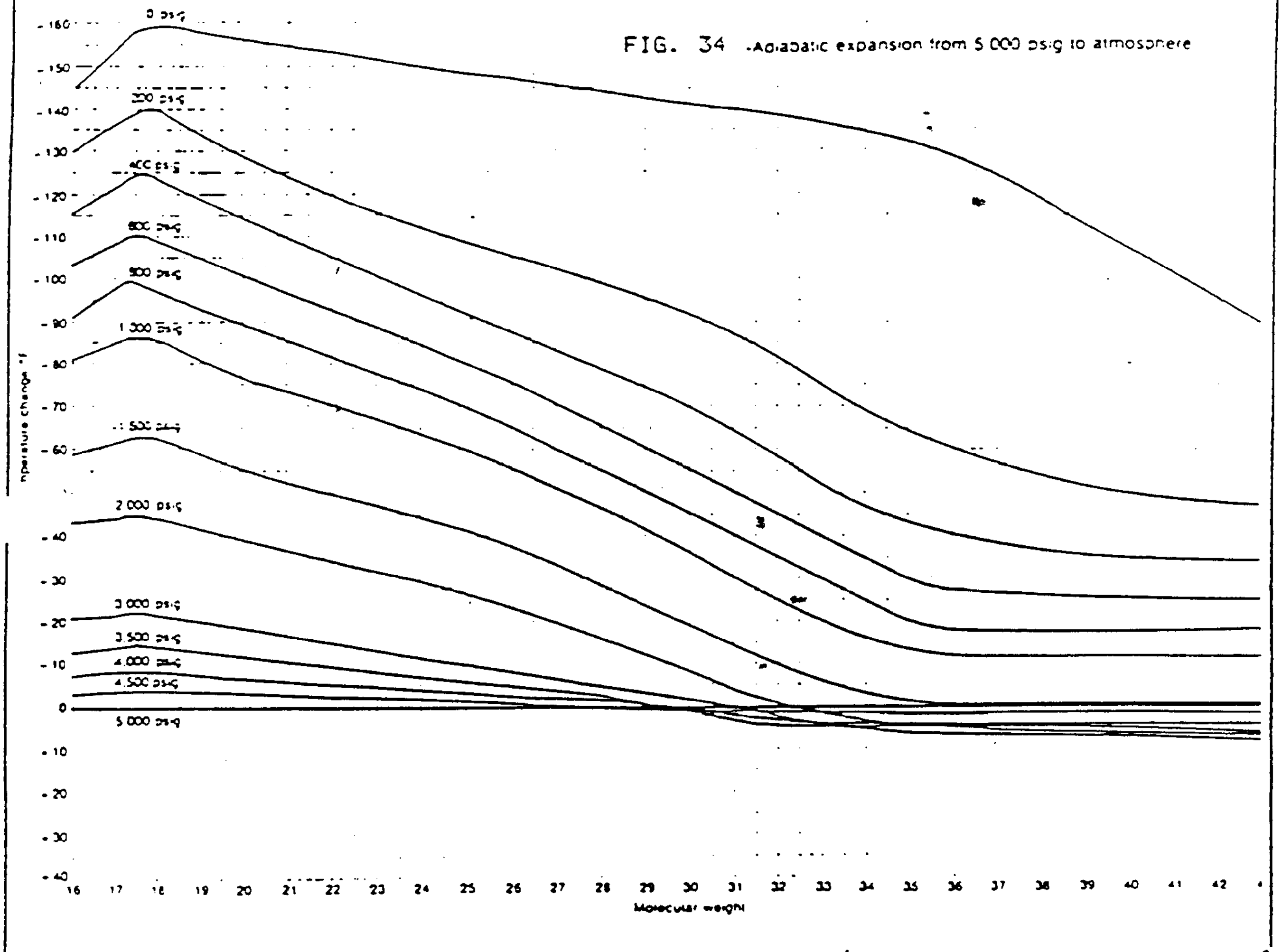
FIG. 33

Solubility of water in hydrocarbons



Adapted to SI by GPSA
Original from Dr. John J. McKetta
University of Texas

FIG. 34 -Adiabatic expansion from 5 000 psig to atmosphere



TEMPERATURE DROPS FOR EXPANDING GAS

FIG. 35 -Adiabatic expansion from 10.000 to 5.000 psig.

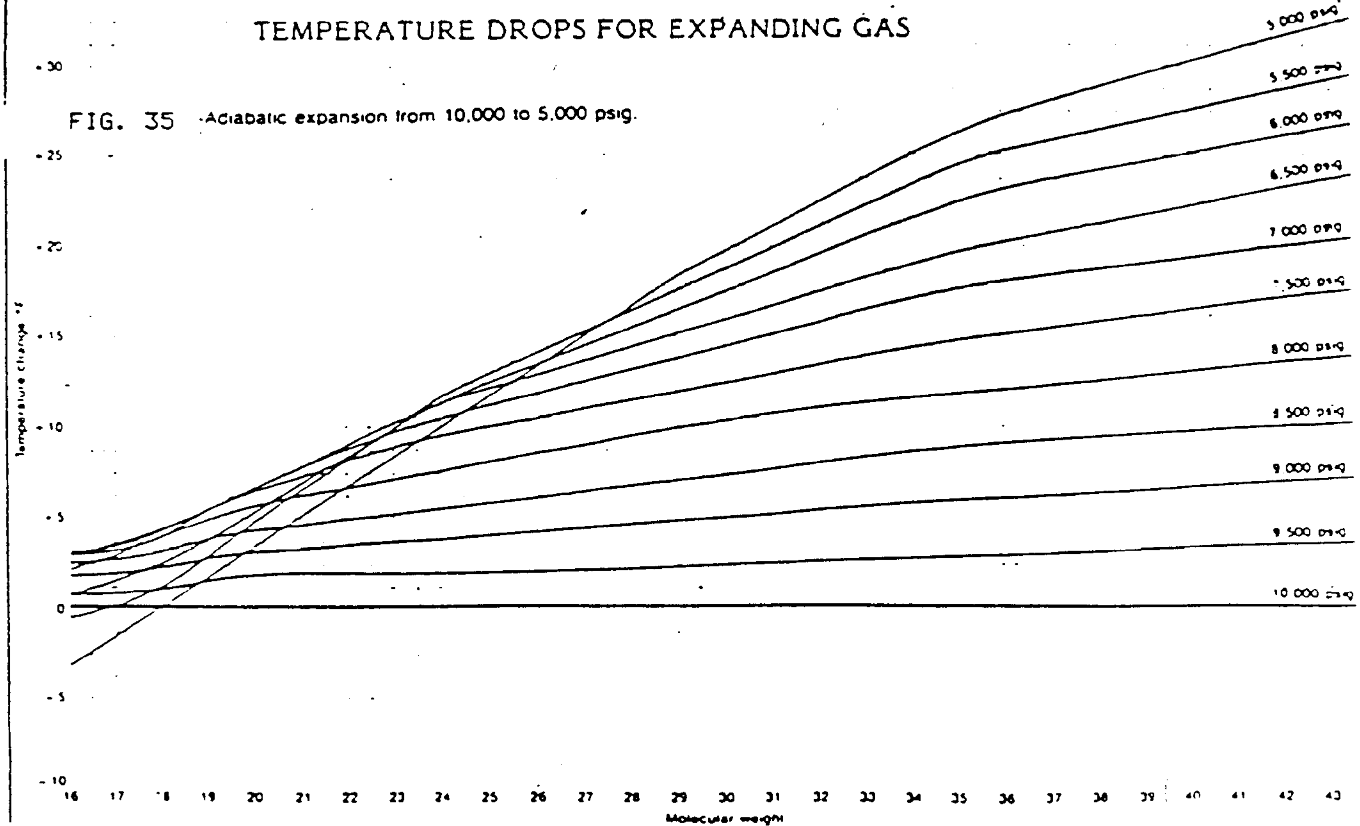


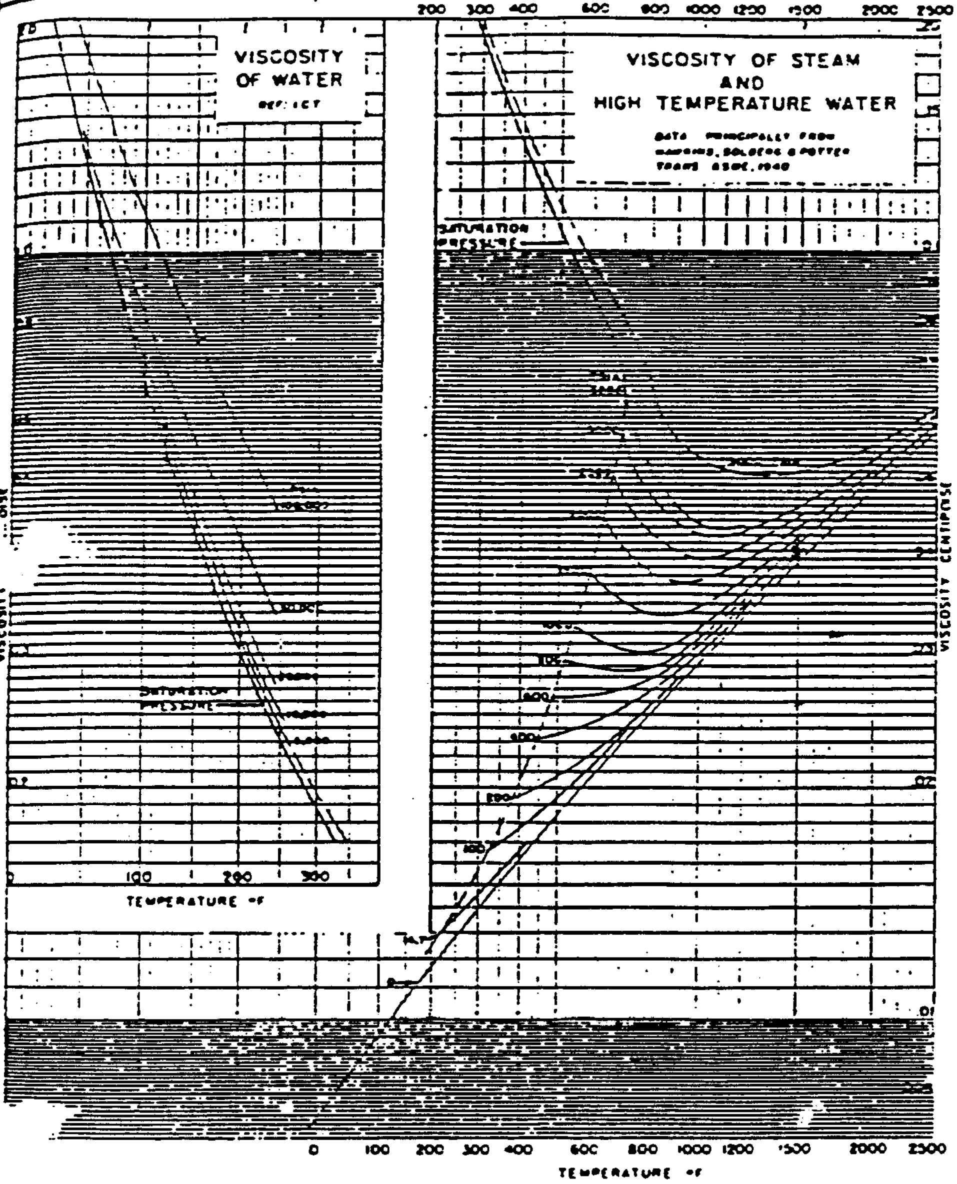
FIG. 36

Physical properties of gas treating chemicals

	Monoethanolamine	Diethanolamine	Triethanolamine	Diglycolamine®	Diisopropanolamine	Selexol®	Propylene Carbonate
Formula	$HOC_2H_4NH_2$	$(HOC_2H_4)_2NH$	$(HOC_2H_4)_3N$	$H(OC_2H_4)_2NH_2$	$(HOC_3H_6)_2NH$	Dimethylether of polyethylene glycol	$C_3H_6CO_3$
Molecular mass	61.08	105.14	148.19	105.14	133.19	280	102.09
Boiling point @ 101.3 kPa (abs), °C	170.5	269	360 (Decomposes)	221	249	270	242
Freezing point, °C	10.5	28	22.4	-12.5	42	-28.9	49.2
Critical constants							
Pressure, kPa (abs)	5985	3273	2448	3772	3770	---	—
Temperature, °C	350	442	514	403	109	---	—
Density @ 20°C, kg/m³	1018	1095	1124	1058 @ 15.6°C	999 @ 30°C	1031 @ 25°C	1205.7
Relative density, 20°C/20°C	1.0179	1.0919 (30/20°C)	1.1258	1.0572	0.989 @ 45/20°C	--	1.203
Specific heat capacity, kJ/(kg·°C) @ 15.6°C	2.546 @ 20°C	2.512	2.931	2.391	2.889 @ 30°C	2.052 @ 5°C	1.403
Thermal conductivity, J/(s·m²·°C/m) @ 20°C	0.256	0.220	—	0.209	—	0.190 @ 25°C	0.208 @ (?)°C
Latent heat of vaporization, kJ/kg at kPa (abs)	826 @ 101.3 kPa	670 @ 9.73 kPa	535 @ 101.3 kPa	510 @ 101.3 kPa	430 @ (?) kPa	---	484 @ 101.3 kPa
Heat of reaction, kJ/kg of acid gas							
H_2S			-930	-1568	—	-442 @ 25°C	...
CO_2			-1465	-1977	—	-372 @ 25°C	
Viscosity, mPa·s	24.1 @ 20°C	350 @ 20°C (90 mass % solution)	1013 @ 20°C (95 mass % solution)	40 @ 15.6°C	198 @ 45°C 86 @ 54°C	5.8 @ 25°C	10.8 @ 25°C 4.5 @ 0°C 2.5 @ 25°C 0.87 @ 100°C
Refractive index, N_d, 20°C	1.4539	1.4776	1.4852	1.4598	1.4542 @ 45°C	---	1.4209
Flash point, COC, °C	93	138	185	127	124	151	132

	Ethylene Glycol	Diethylene Glycol	Triethylene Glycol	Tetraethylene Glycol	Sulfolane®	Methanol
Formula	HOC_2H_4OH	$HO(C_2H_4O)_2H$	$HO(C_2H_4O)_3H$	$HO(C_2H_4O)_4H$	$C_4H_8SO_2$	CH_3OH
Molecular mass	62.07	106.12	150.17	194.32	120.17	32.04
Boiling point @ 101.3 kPa (abs), °C	197	245	287	327	285	64.5
Freezing point, °C	-13	-8	-7.2	-6.2	27.6	-97.7
Critical constants						
Pressure, kPa (abs)	7696	4661	3304	2594	5290	7956
Temperature, °C	372	408	442	474	545	240
Density @ 20°C, kg/m³	1113	1116	1123	1246	1265	790
Relative density, 20°C/20°C	1.1155	1.1184	1.1254	1.248	1.268	0.7917
Specific heat capacity, kJ/(kg·°C) @ 15.6°C	2.311	2.240	2.060	2.177	1.507 @ 30°C	2.470 (5°-10°C)
Thermal conductivity, J/(s·m²·°C/m) @ 20°C	0.287 @ 15.6°C	0.249 @ 15.6°C	0.241 @ 15.6°C	—	0.197 @ 37.8°C	0[illegible]15
Latent heat of vaporization, kJ/kg at kPa (abs)	800 @ 101.3 kPa	540 @ 101.3 kPa	416 @ 101.3 kPa	374 @ 101.3 kPa	525 @ 100°C	1103 @ 101.3 kPa
Viscosity, mPa·s	20.9 @ 20°C	35.7 @ 20°C	47.9 @ 20°C	60.0 @ 20°C	10.3 @ 30°C 6.1 @ 50°C 2.5 @ 100°C 1.4 @ 150°C 0.97 @ 200°C	0.6 @ 20°C
Refractive index, N_d, 20°C	1.4316	1.4472	1.4559	1.4595	1.481 @ 30°C	1.3286
Flash point, COC, °C	116	143	166	17[illegible]	177	14

NOTE: 1 mPa·s = 1 centipoise



PHYSICAL PROPERTIES OF WATER

Heat Capacity of Water*

Temperature, °C.	Heat capacity, constant pressure, 1 atm cal/g °C
0	1.00803
1	1.00717
2	1.00636
3	1.00564
4	1.00495
5	1.00433
6	1.00378
7	1.00325
8	1.00277
9	1.00234
10	1.00194
11	1.00158
12	1.00124
13	1.00093
14	1.00067
15	1.00041
16	1.00019
17	0.99998
18	.99978
19	.99962
20	.99947
21	.99933
22	.99921
23	.99912
24	.99902
25	.99892
26	.99885
27	.99878
28	.99873
29	.99869
30	.99866
31	.99864
32	.99861
33	.99861
34	.99859
35	.99859
36	.99861
37	.99861
38	.99864
39	.99866
40	.99869
41	.99871
42	.99876
43	.99880
44	.99883
45	.99890
46	.99895
47	.99900
48	.99907
49	.99912
50	0.99919
51	.99926
52	.99935
53	.99943
54	.99950
55	.99959
56	.99969
57	.99978
58	.99988
59	.99998
60	1.00007
61	1.00019
62	1.00029
63	1.00041
64	1.00053
65	1.00065
66	1.00079
67	1.00091
68	1.00105
69	1.00117
70	1.00131
71	1.00146
72	1.00160
73	1.00177
74	1.00191
75	1.00208
76	1.00225
77	1.00241
78	1.00258
79	1.00277
80	1.00294
81	1.00313
82	1.00332
83	1.00351
84	1.00373
85	1.00392
86	1.00414
87	1.00435
88	1.00457
89	1.00480
90	1.00502
91	1.00526
92	1.00550
93	1.00574
94	1.00600

Thermal Conductivity of Water and Steam*
(mwatts/(m x °K.)

Pressure bars	Temp., °C										
	0	50	100	150	200	250	300	400	500	600	700
1	569	641	24.6	28.7	33.2	38.2	43.4	54.9	67.4	80.6	94.3
50	573	647	684	690	666	619	52.5	61.2	72.0	85.0	96.6
100	577	651	668	691	672	625	545	66.4	77.8	89.8	103
150	581	655	691	696	676	633	559	82.2	84.3	95.1	108
200	585	659	695	700	681	639	571	107	92.6	101	113
250	589	662	698	703	685	646	582	157	103	107	119
300	592	666	701	706	689	652	592	264	116	114	124
350	596	669	704	710	693	657	601	351	132	122	129
400	599	672	707	713	697	662	609	390	131	130	135
450	603	675	710	716	701	667	618	416	160	139	142
500	606	678	713	720	704	671	622	436	208	149	149

*Extracted from Appendix, Table J, of Official Report of meeting of the Thermal Conductivity panel, 6th International Conference on the Properties of Steam, Paris, 1964. For unit equivalence, see Table 3-266.

PHYSICAL PROPERTIES OF AIR

Thermal Conductivity of Air*

Thermal conductivity k, Btu/ft hr °F

Pressure, lb/sq in abs	−250	−200	−150	−100	−50	0	50	100	150	200	250
0	0.615	0.763	0.903	1.042	1.175	1.306	1.433	1.555	1.670	1.782	1.888
14.7	0.621	0.766	0.905	1.045	1.181	1.311	1.436	1.557	1.671	1.782	1.888
50	0.629	0.774	0.914	1.053	1.185	1.315	1.443	1.562	1.672	1.783	1.889
100	0.647	0.787	0.927	1.065	1.194	1.325	1.452	1.569	1.675	1.784	1.890
150	0.670	0.805	0.941	1.077	1.210	1.335	1.461	1.576	1.679	1.785	1.891
200	0.733	0.824	0.958	1.090	1.223	1.350	1.470	1.584	1.685	1.788	1.893
250		0.849	0.977	1.107	1.238	1.363	1.480	1.593	1.691	1.794	1.896
300		0.882	0.999	1.126	1.254	1.377	1.491	1.602	1.699	1.801	1.899
400		0.967	1.035	1.156	1.283	1.405	1.512	1.620	1.714	1.813	1.91
500		1.125	1.065	1.195	1.319	1.434	1.533	1.636	1.730	1.824	1.92
600			1.15	1.25	1.38	1.46	1.56	1.66	1.75	1.84	1.93
700			1.20	1.30	1.40	1.50	1.58	1.68	1.77	1.86	1.94
800			1.27	1.35	1.45	1.53	1.60	1.70	1.78	1.89	1.96
900			1.35	1.41	1.50	1.56	1.63	1.72	1.80	1.89	1.97
1000			1.44	1.48	1.53	1.59	1.65	1.74	1.82	1.91	1.98

*Compiled by P. E. Liley. To obtain actual thermal-conductivity values divide the table values by 100. For tables in SI units from 90° to 1300°K., 1 to 1000 bars, see Vasserman, Kazavchinskii, and Rabinovich, "Thermophysical Properties of Air and Air Components," Moscow, 1966, and NBS-NSF trans. TT 70-50095, 1971. This source discusses present-day accuracy.

Viscosity of Air*

Viscosity, lb/(ft)(hr) × 10⁻²

Pressure, lb/sq in abs	−100	−50	0	50	100	150	200	250
200	3.27	3.64	3.95	4.29	4.57	4.78	5.12	5.45
400	3.39	3.73	4.06	4.36	4.63	4.86	5.19	5.51
600	3.54	3.83	4.14	4.43	4.69	4.94	5.25	5.56
800	3.72	3.95	4.22	4.50	4.76	5.02	5.31	5.61
1,000	3.90	4.07	4.31	4.58	4.84	5.10	5.38	5.67
1,200	4.06	4.20	4.42	4.66	4.92	5.16	5.44	5.72
1,400	4.26	4.35	4.54	4.77	5.00	5.24	5.50	5.77
1,600	4.47	4.53	4.66	4.87	5.06	5.31	5.57	5.84
1,800	4.70	4.73	4.83	5.00	5.17	5.39	5.63	5.90
2,000	5.10	4.95	4.97	5.10	5.27	5.47	5.70	5.97
2,500	6.05	5.52	5.36	5.36	5.51	5.68	5.87	6.07
3,000	6.82	6.14	5.77	5.70	5.76	5.91	6.06	6.19
3,500	7.62	6.76	6.25	6.06	6.08	6.12	6.25	6.43
4,000	8.35	7.34	6.65	6.42	6.36	6.42	6.43	6.63
4,500	9.10	7.91	7.09	6.76	6.66	6.69	6.71	6.82
5,000	9.86	8.49	7.55	7.16	6.99	6.99	6.97	7.02
6,000	11.33	9.66	8.39	7.90	7.66	7.52	7.43	7.60
7,000	12.83	10.76	9.17	8.61	8.26	8.03	7.92	8.10
8,000	14.56	11.94	10.16	9.42	8.89	8.56	8.39	8.52
9,000	16.09	12.94	11.05	10.11	9.46	9.07	8.83	8.90
10,000	17.70	14.03	11.85	10.79	10.10	9.65	9.37	9.17

*Compiled by P. E. Liley. For tables in SI units from 90° to 1300°K., 1 to 1000 bars, see Vasserman, Kazavchinskii, and Rabinovich, "Thermophysical Properties of Air and Air Components," Moscow, 1966, and NBS-NSF trans. TT 70-50095, 1971. This source contains a discussion of present accuracy.

Specific Heat of Air at High Pressures

Cal./g °C.

Temp., °C.	Atmospheres 1	10	20	40	70	100
100	0.237	0.239	0.240	0.245	0.250	0.254
0	.238	.242	.247	.251	.277	.298
−50	.238	.246	.257	.279	.332	.412
−100	.239	.259	.285	.370	.846	
−150	.240	.311	.505			

For the specific heats of other materials as a function of the pressure, see "International Critical Tables," vol. 5, pp. 82–83. See Figs. 3-11 and 3-14.

COMPOSITION OF THE ATMOSPHERE

The composition of dry air is remarkably constant all over the globe and throughout the entire troposphere. The proportions *by volume* of the various components are given below (after A. F. Paneth, 1939, 1952).

Substance	% by volume	Substance	% by volume
N_2	78·09	CH_4	$2{\cdot}0 \times 10^{-4}$
O_2	20·95	Kr	1×10^{-4}
Ar	0·93	H_2	5×10^{-5}
*CO_2	0·03	N_2O	5×10^{-5}
Ne	$1{\cdot}8 \times 10^{-3}$	Xe	9×10^{-6}
He	$5{\cdot}2 \times 10^{-4}$	Rn	6×10^{-18}

*This varies somewhat near towns and industrial areas.