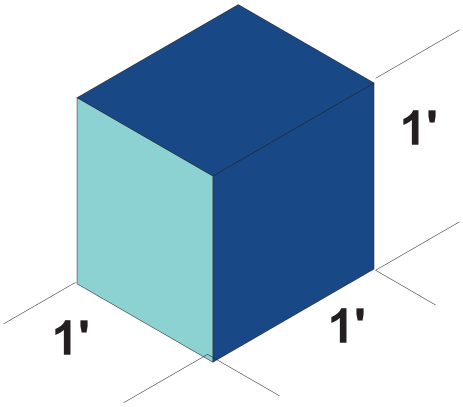
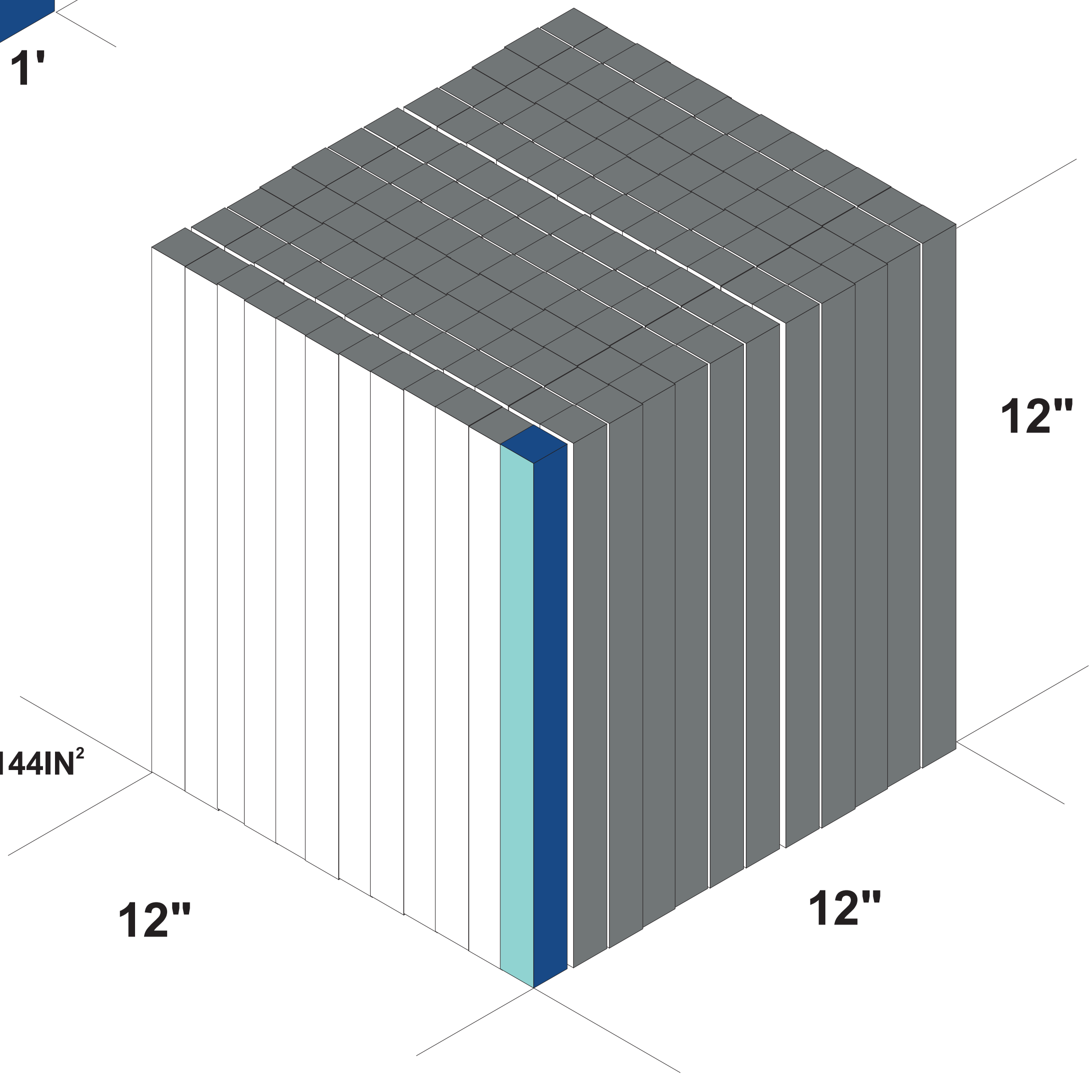
PRESSURE - WHAT A CONCEPT!

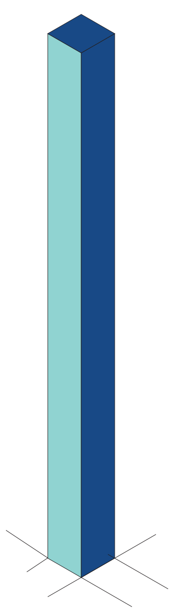


Fresh Water = 62.4 #/Ft^3
1FT x 1FT X 1 FT = 1FT^3 = 62.4 #'S



BASE AREA=12"X12"=144IN^2

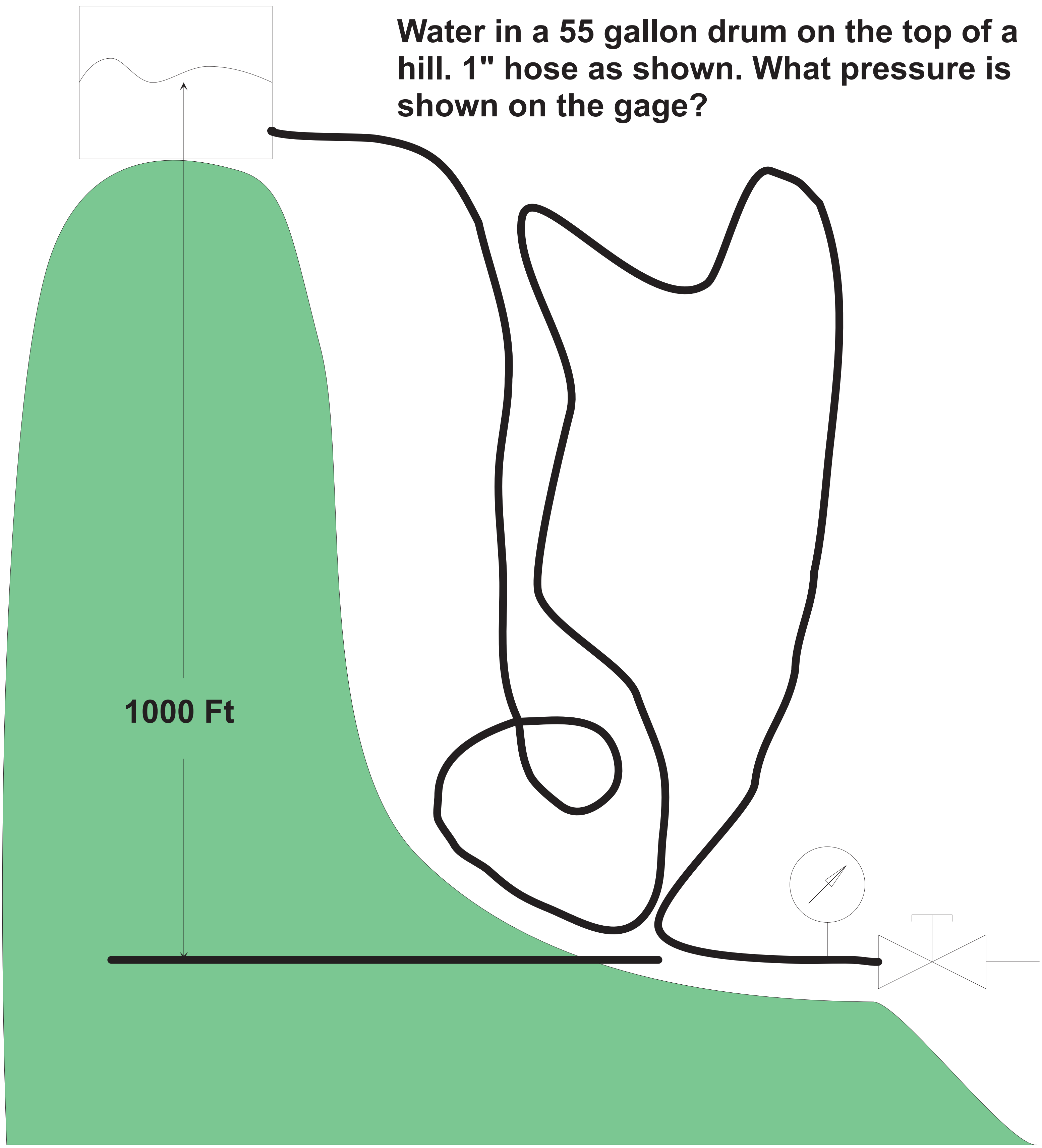
PRESSURE HEAD
62.4lbs/144In^2=.433lbs/ft
EVERY FT OF HEIGHT WATER
GENERATES A PRESSURE OF .443#/In^2

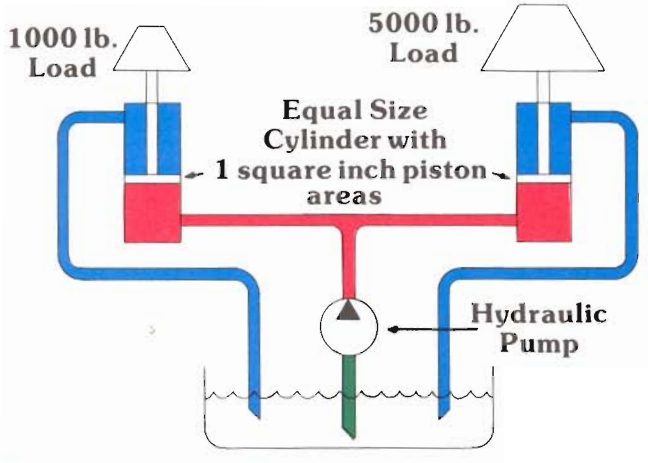


.433PSI/FT   **P=.433XhxSg.**

Static Pressure

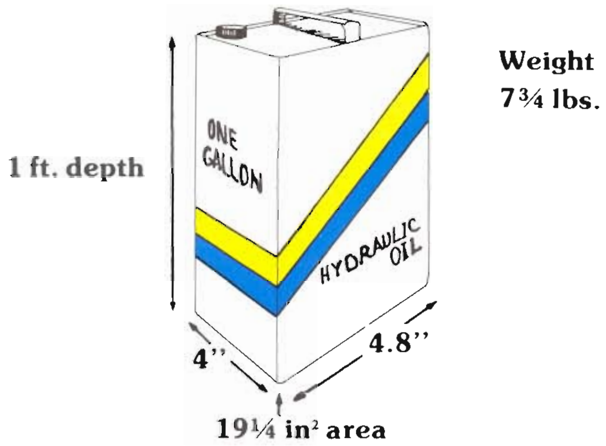
Water in a 55 gallon drum on the top of a hill. 1" hose as shown. What pressure is shown on the gage?





Fluid takes the path of least resistance.

HEAD PRESSURE



Static head pressure is a force over an area created by the weight of the fluid itself. If we were to weigh a one gallon volume of a typical hydraulic oil, we would find that it weighs approximately 7¾ lbs. Likewise, a container which holds one gallon of fluid at a one foot depth, has a bottom with a surface area of 19¼ square inches (approximately 4.8" x 4"). Consequently, we have a pressure on the bottom of the container of 7¾ lbs. over 19¼ square inches or:

$$\frac{\textbf{7.75 lbs.}}{\textbf{19.25 sq ins}} = \textbf{0.4 PSI per ft}$$

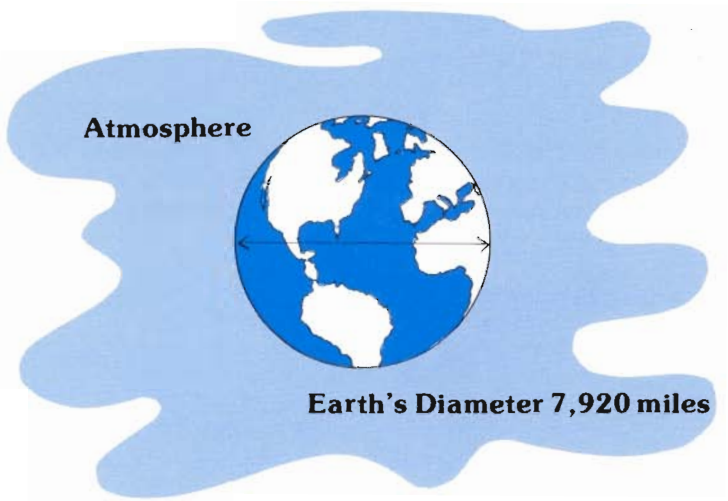
It does not matter how big the bottom of the container is, we are only concerned, that for every one foot of depth, *oil creates a static head pressure of 0.4 PSI.*

The weight of oil in a large reservoir which holds oil at a 10 foot depth creates a pressure in the fluid at the bottom of the reservoir of 4 PSI:

$$\textbf{10}\,\cancel{\textbf{ft.}}\ \textbf{depth x .4}\ \frac{\textbf{psi}}{\cancel{\textbf{ft}}} = \textbf{4 PSI}$$

ATMOSPHERIC PRESSURE AND VACUUMS

As with oil, air also has weight. It is a well known fact that atmospheric pressure at sea level averages 14.7 pounds per square inch. Therefore if we were able to weigh our earth's atmosphere, and then divide this weight by the surface area of the earth, we would find that the pressure due to the weight of air above us is equal to 14.7 PSI. In working backwards we find some interesting trivia:

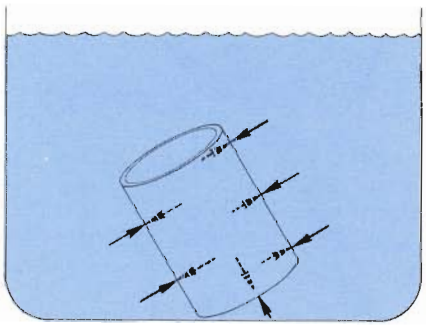


The earth has a diameter of 7920 miles or a radius of 3960 miles. Consequently, its surface area is:

$A = 4\pi r^2 = 197{,}060{,}790$ square miles
or 791,098,480,000,000,000 sq. inches

Our Atmosphere weighs $= 7.91 \times 10^{17}\ \cancel{\text{sq. ins.}} \times 14.5 \dfrac{\text{lbs.}}{\cancel{\text{sq. ins.}}} =$

11,470,927,000,000,000,000 lbs. OR 5.73×10^{15} tons.



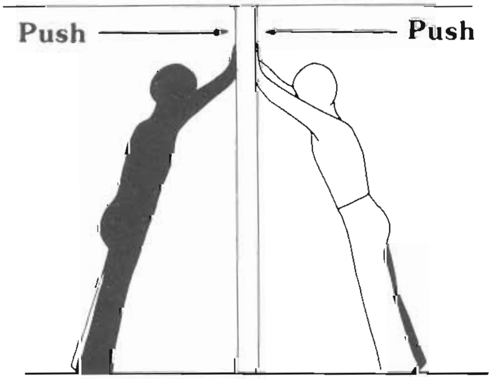
Force due to head pressure is normally equal in all directions.

As with any object immersed in a fluid (either air or oil), we live in an *atmosphere* which exerts 14.7 pounds on every square inch of our body. Because we breathe, the pressure is also exposed internally so we don't feel the effect of this pressure. We do, however, feel the effects of static head pressure when we dive into the deep end of a swimming pool. This happens because we can't breath in the higher pressure water to equalize internal and external forces. Under normal conditions, the forces created by head pressure are *equal in all directions*, and cancel the effects of each other.

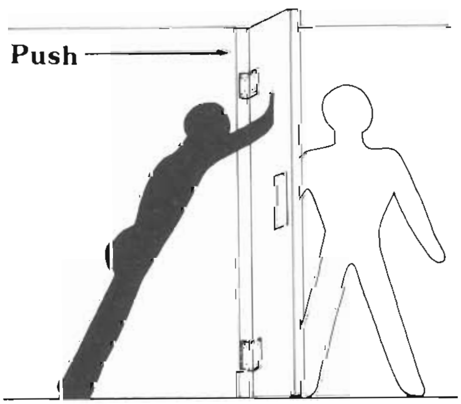
VACUUM

A perfect vacuum is a volume which is totally evacuated. In a perfect vacuum all air molecules are removed. Although a perfect vacuum is virtually impossible to achieve, partial vacuums can be used so that the static head pressure of our atmosphere can exert a force to do work.

In comparison with a mechanical system, a vacuum and atmospheric pressure work together much like two people pushing on opposite sides of a door:



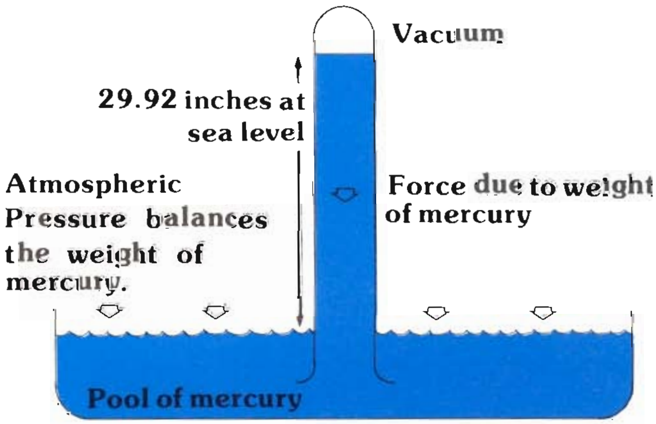
Equal force on both sides of the door cancel the effects of each other, and the door cannot move.



When one person stops pushing the door moves in his direction.

BAROMETER

A barometer is a device used to measure the effects of the static head pressure of our atmosphere. Barometers measure the *absolute pressure* of the atmosphere and are usually calibrated in inches of mercury. A mercury barometer can be made by immersing a long test tube in a pool of liquid mercury. When all air is expelled, the tube is turned vertically with the open end still suspended in the pool. In doing this, the mercury drops in the tube, leaving a near perfect vacuum in the top of the tube. The mercury is simply the media separating the atmospheric pressure from the vacuum. The gauge is read by measuring how high the ambient pressure pushes the mercury up the tube. Hence, the calibration in inches of mercury. Water could also be used in place of mercury. However, 29.92 inches of mercury is equivalent to 34 feet of water which would require an extremely long test tube.



ABSOLUTE AND GAUGE PRESSURE

There are two basic methods of measuring pressure. Namely those readings which take into account the atmospheric pressure we live in and secondly, readings which ignore atmospheric pressure and start the scale at zero when the ambient pressure is actually 14.7 PSI.

Absolute pressure readings use a vacuum as their zero base. Atmospheric pressure on this scale would be 14.7 PSI or 30 in-hg (29.92 in-hg).

Gauge pressure readings are always 14.7 PSI lower than absolute pressure, since a standard pressure gauge will read 0 PSI at sea level.

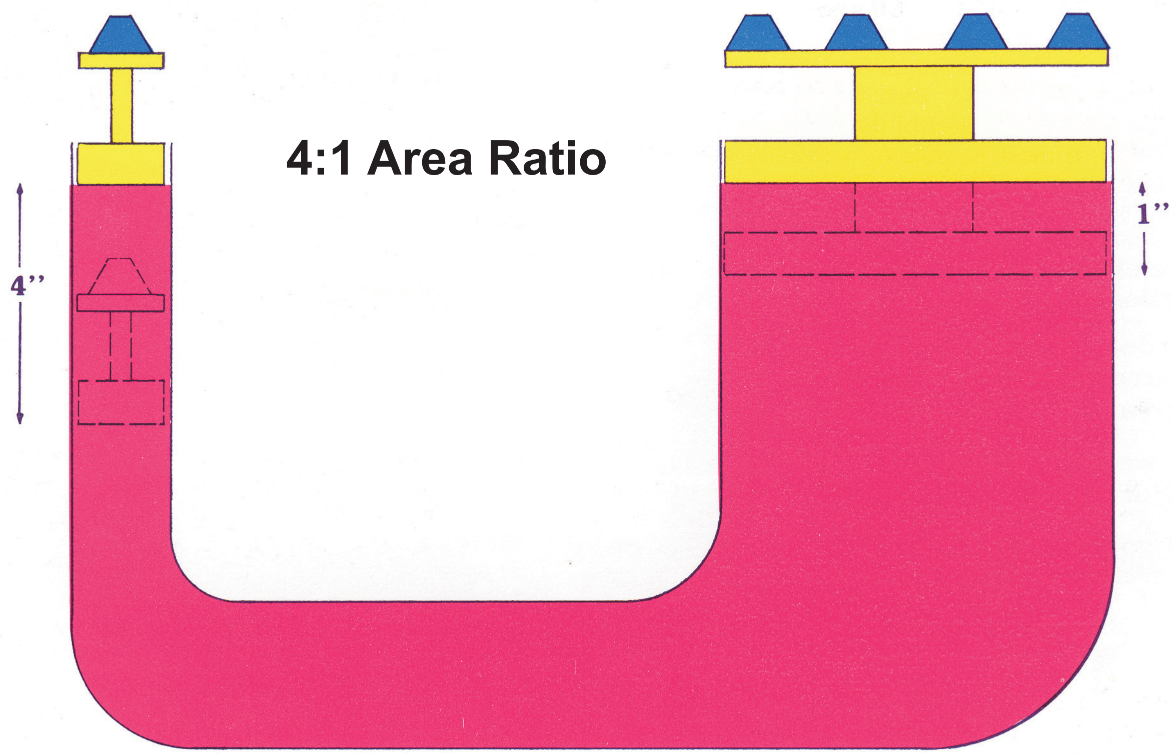
Therefore:

Gauge Pressure + 14.7 PSI = Absolute Pressure

Force Multiplier

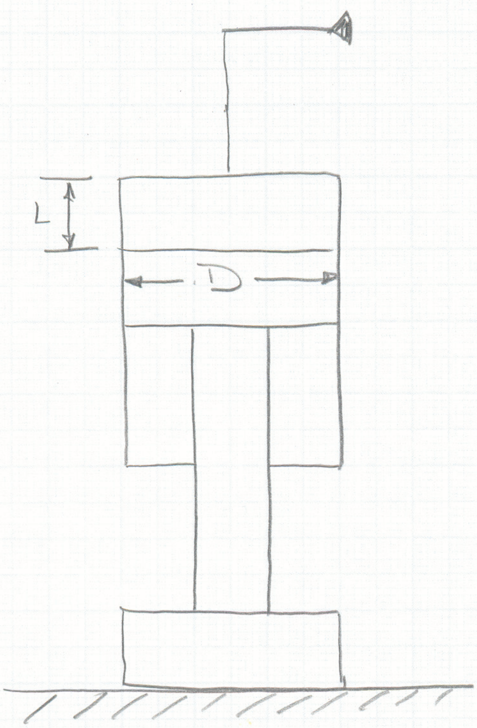
F=PXA

Where:
F= force
P=pressure
A=Area



We gain a force advantage (multiplier) at the cost of stroke (distance). This is always the 'area ratio'.

Oil Compression



$$V_C = \frac{(V_T)(\Delta P)}{\beta}$$

WHERE: V_C = COMPRESSED VOLUME

V_T = TOTAL VOLUME UNDER PRESSURE

ΔP = CHANGE IN PRESSURE

β = BULK MODULUS = 200,000 – 300,000 PSI.

V_T = (CYL AREA) x (STROKE) + (PIPE AREA) x (L)

RULE OF THUMB

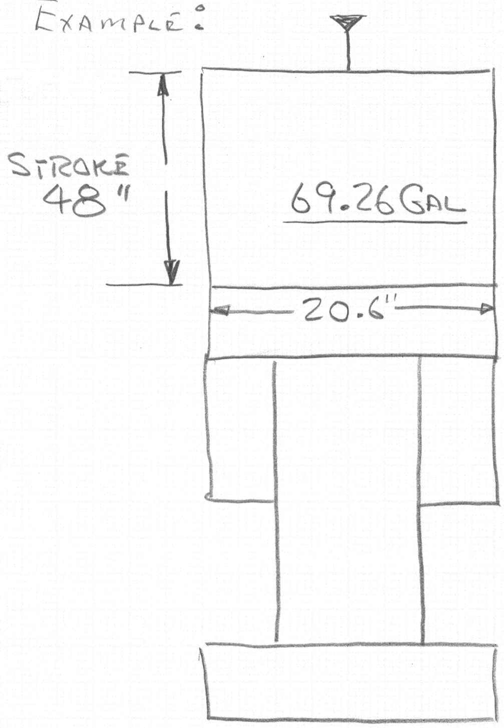
* OIL IS COMPRESSED 1/2% FOR EVERY 1,000 PSI CHANGE IN PRESSURE.

Oil is compressible - About 1/2% of it's total volume subjected to pressure is compressed for every 1,000 psi increase in pressure.

Oil Compression

So why is oil compression / oil decompression important? Look at the following example.
500 Ton Press with a single cylinder ... Forget about the pipes and block volume for the time being

EXAMPLE:



500 TON PRESS
- 3000 PSI. MAX PRESSURE
- 20.6" DIA => 333.33 IN^2

$$V_C = \frac{(V_T)(\Delta P)}{\beta} =$$

V_T = CYL VOLUME = AREA X STROKE

$$A = \frac{\pi D^2}{4} = \frac{(3.14)(20.6^2)}{4} = 333.33\ IN^2$$

STROKE = 48"

$$V_C = (A)(S) = (333.33\ IN^2)(48\ IN) =$$

$$V_C = 15{,}998.00\ IN^3$$

$$V_C = (15{,}998\ \cancel{IN^3})\left(\frac{1\ GAL}{231\ \cancel{IN^3}}\right) = 69.26\ GAL.$$

∴ THERE IS 69.26 GALLONS OF OIL UNDER PRESSURE.

ΔP => THE ΔP OR "CHANGE IN PRESSURE" IS 3000 PSI. TO BUILD UP TO 500 TONS

β => THE "BULK MODULUS" OF OIL IS ABOUT 250,000 ... BUT THE PRESS FRAME BENDS & STRETCHES. WE HAVE FOUND THE PRESS "MODULUS" IS 200,000 PSI.

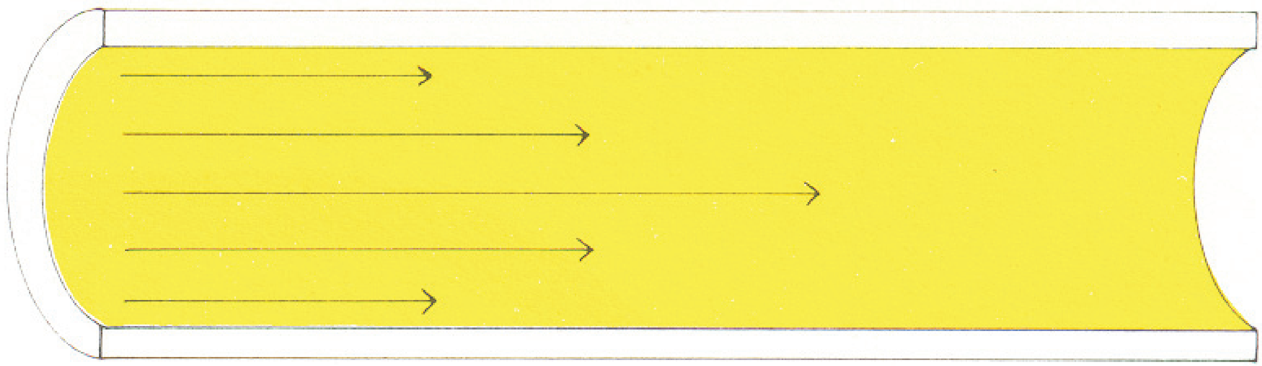
$$\therefore\ V_C = \frac{(V_T)(\Delta P)}{\beta} = \frac{(69.26\ GAL)(3000\ PSI)}{200{,}000\ PSI} = \underline{1.04\ GAL}$$

SO... THIS MEANS THAT TO BUILD UP TO 3000 PSI, YOU MUST <u>ADD</u> 1.04 GALLONS OF OIL TO A FULL CYLINDER. AT 150 GPM = 2.5 GAL/SEC =>

$$TIME = \frac{1.04\ GAL}{2.5\ GAL/SEC} = \underline{\underline{.42\ SEC}}$$

NOW THAT THE OIL IS COMPRESSED, WE HAVE TO DECOMPRESS IT OVER SOME TIME ... ABOUT .2 SEC, ... FLOW RATE AT 312 G.P.M.

FLOW

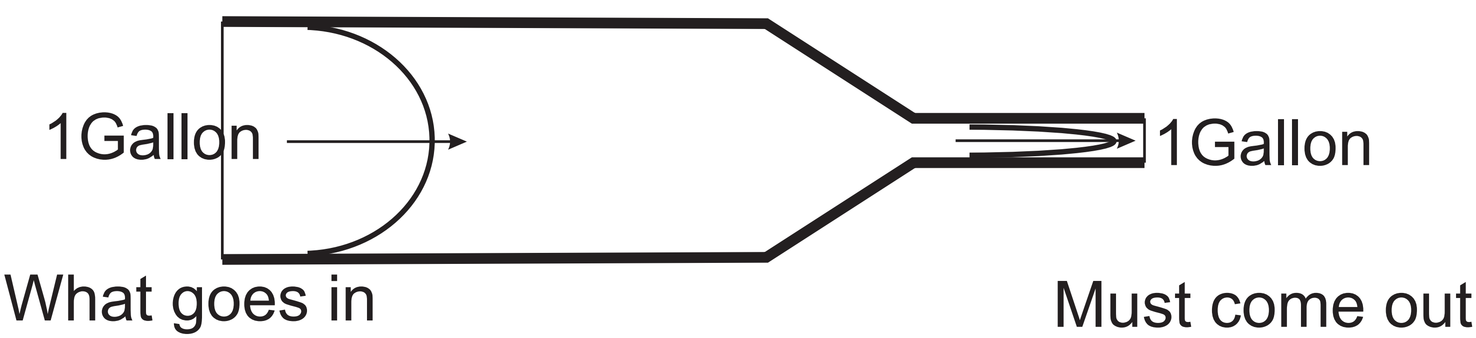


FLOW= VOLUME / TIME or Gallons / Minute or In^3/Min

Q=AXV / 231 **where: A=Area In2**
V=Velocity In / in
Q=Gallons per Minute

Velocity= (.3208 x Q) / Area

Area = (3.14) X Dia2) / 4



Since the flow rate remains the same on both ends, the velocity must increase.

The increase in velocity is proportional to the area difference

Laminar vs. Turbulent Flow

Nature of Flow in Pipe — Laminar and Turbulent

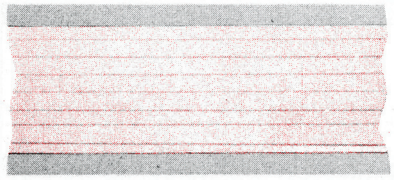


Figure 1-1
Laminar Flow
Actual photograph of colored filaments being carried along undisturbed by a stream of water.

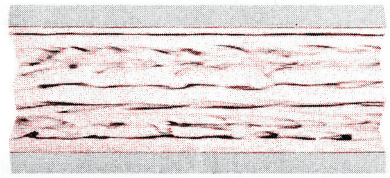


Figure 1-2
Flow in Critical Zone, Between Laminar and Transition Zones
At the critical velocity, the filaments begin to break up, indicating flow is becoming turbulent.

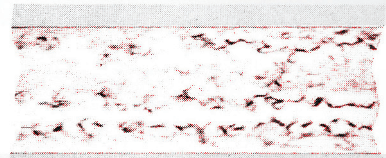


Figure 1-3
Turbulent Flow
This illustration shows the turbulence in the stream completely dispersing the colored filaments a short distance downstream from the point of injection.

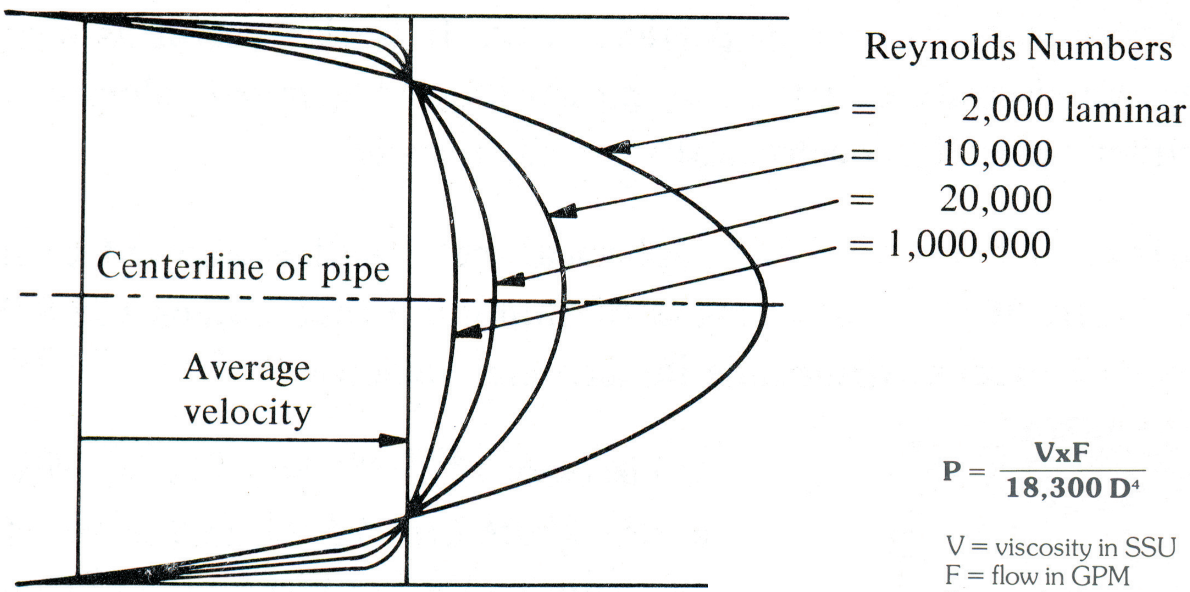


Figure 6.8. *Velocity distributions*

$$P = \frac{V x F}{18{,}300\, D^4}$$

V = viscosity in SSU
F = flow in GPM
D = inside diameter of pipe in inches
P = pressure drop per foot in PSI

If the flow rate in GPM and the desired oil velocity in feet per second (feet/sec.) are known the following formula can be used to calculate the *inside area* required:

$$\text{Area} = \frac{\text{GPM x .3208}}{\text{velocity}}$$

Because available energy levels in different portions of the circuit vary, we recommend the following velocities to keep the energy loss to a small percentage of the total energy available:

Suction Lines	**2 to 4 feet/sec.**
Return Lines	**10 to 15 feet/sec.**
Working Lines (500 - 3000 PSI)	**15 to 20 feet/sec.**
Working Lines (3000 - 5000 PSI)	**15 to 30 feet/sec.**

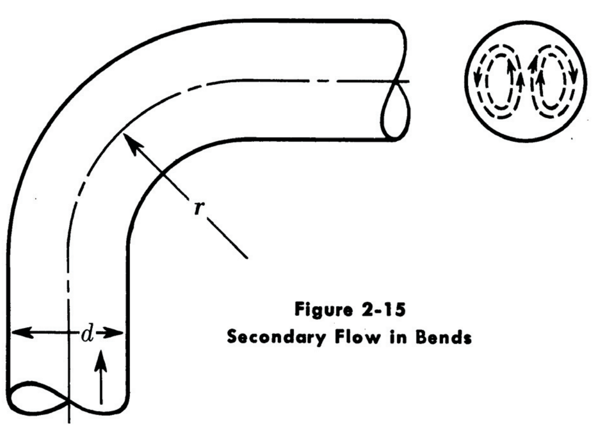
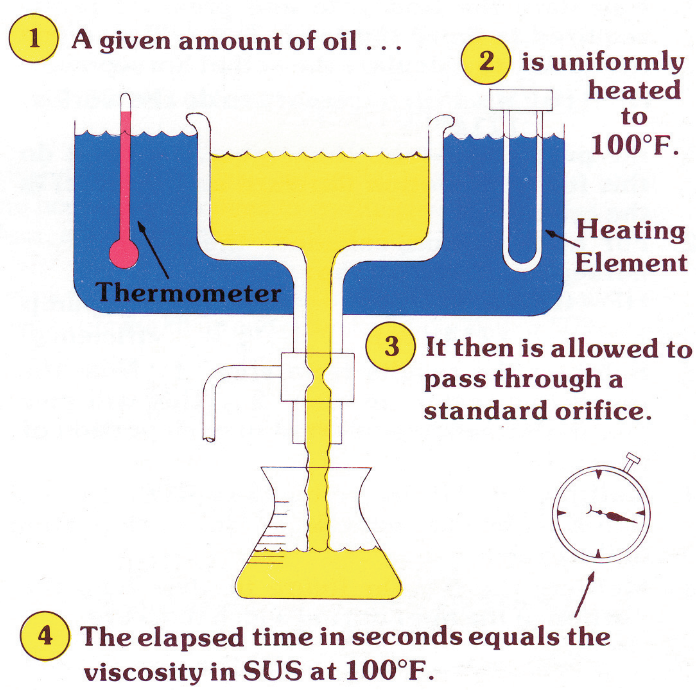


Figure 2-15
Secondary Flow in Bends

VISCOSITY

RELATIVE VISCOSITY IS MEASURED IN SUS

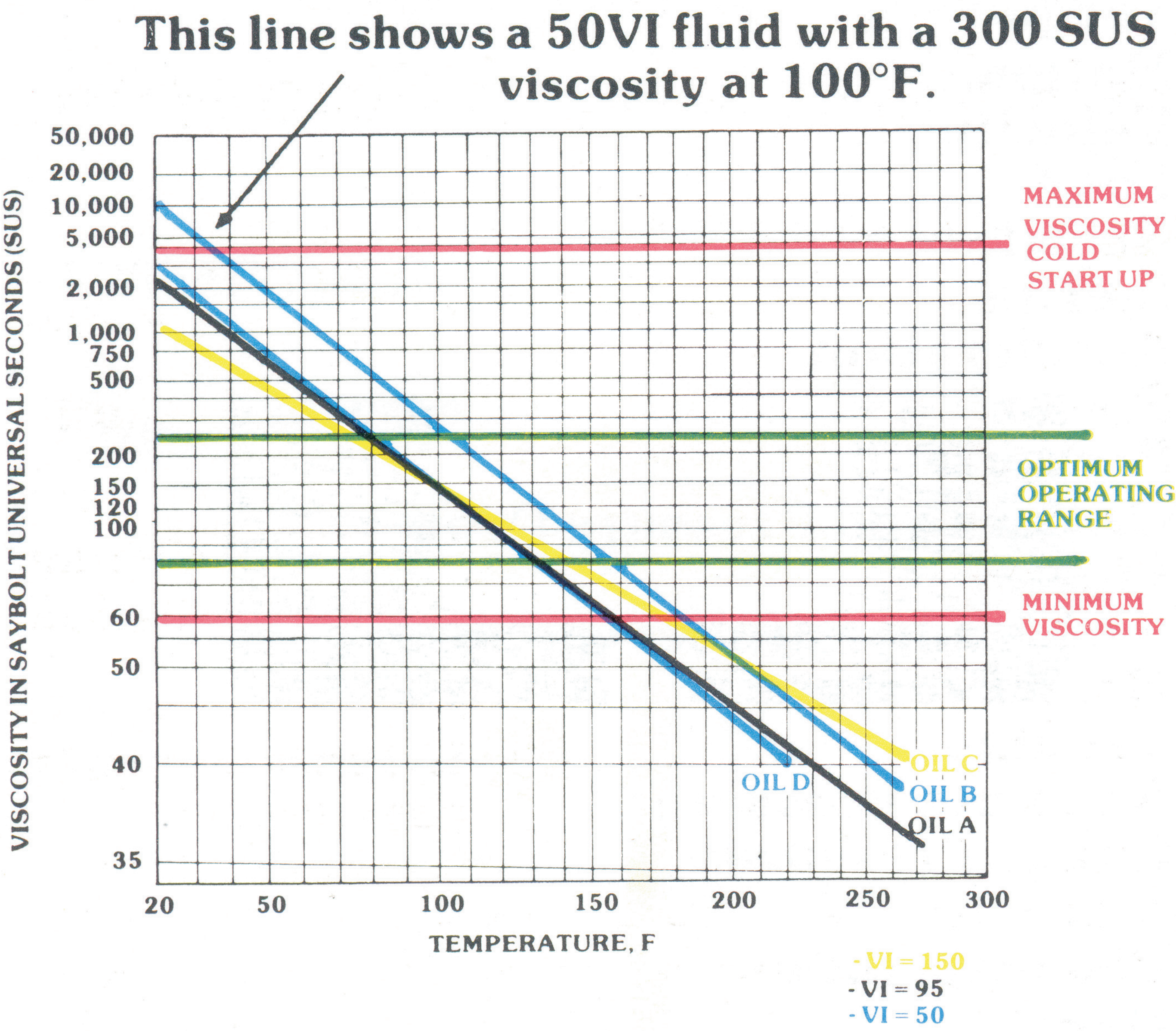


Temperature, (°F)	Specific Weight, γ (lb/ft^3)	Density, ρ ($slugs/ft^3$)	Viscosity, μ ($lb\ sec/ft^2$) $10^5\mu =$	Kinematic Viscosity, ν (ft^2/sec) $10^5\nu =$	Surface Tension, σ (lb/ft) $100\sigma =$	Vapor Pressure Head, p_v/γ (ft)	Bulk Modulus of Elasticity, β ($lb/in.^2$) $10^{-3}\beta =$
32	62.42	1.940	3.746	1.931	0.518	0.20	293
40	62.43	1.940	3.229	1.664	0.514	0.28	294
50	62.41	1.940	2.735	1.410	0.509	0.41	305
60	62.37	1.938	2.359	1.217	0.504	0.59	311
70	62.30	1.936	2.050	1.059	0.500	0.84	320
80	62.22	1.934	1.799	0.930	0.492	1.17	322
90	62.11	1.931	1.595	0.826	0.486	1.61	323
100	62.00	1.927	1.424	0.739	0.480	2.19	327
110	61.86	1.923	1.284	0.667	0.473	2.95	331
120	61.71	1.918	1.168	0.609	0.465	3.91	333
130	61.55	1.913	1.069	0.558	0.460	5.13	334
140	61.38	1.908	0.981	0.514	0.454	6.67	330
150	61.20	1.902	0.905	0.476	0.447	8.58	328
160	61.00	1.896	0.838	0.442	0.441	10.95	326
170	60.80	1.890	0.780	0.413	0.433	13.83	322
180	60.58	1.883	0.726	0.385	0.426	17.33	318
190	60.36	1.876	0.678	0.362	0.419	21.55	313
200	60.12	1.868	0.637	0.341	0.412	26.59	308
212	59.83	1.860	0.593	0.319	0.404	33.90	300

† Reproduced with permission from *Fluid Mechanics* by V. L. Streeter, McGraw-Hill Book Company, Inc., New York, 1962, p. 533.

VISCOSITY INDEX

Viscosity Index - The amount Viscosity changes with temperature is defined by a factor called Viscosity Index. The higher the number the less the oil viscosity will change with temperature.



Viscosity Rating Systems

Kinematic Viscosity expresses total resistance to fluid flow including internal fluid friction plus effect of mass or weight of the fluid. It is measured in several systems, with equivalent values shown in the chart compared to SUS ratings in the first column. All these systems are based on the time for a quantity of fluid to flow through a standard orifice under specified conditions. In the U.S. the Saybolt Universal Second (SUS) rating is most often used. It is derived from English units. The Centistoke is the standard for international fluid power. It is derived from metric units. **(1 Stoke = 100 Centistokes)**.

Absolute Viscosity is an expression only of the internal fluid friction without taking into account the effect of the mass or weight of the fluid. A statement of absolute viscosity must also include a statement of the specific gravity of the fluid. The international standard unit for absolute viscosity is the Poise or Centipoise **(1 Poise = 100 Centipoise)**. It is derived from metric units. In the English system the unit is the Reyn. Centipoise viscosities in the last column of the chart are for any fluid, including standard hydraulic oil, which has a specific gravity of 0.9. The Centipoise is related to the Centistoke. Any value of kinematic viscosity in Centistokes can be converted to absolute viscosity in Centipoise by multiplying Centistokes times the specific gravity. Thus water, with specific gravity of 1.0, has the same kinematic and absolute viscosity ratings.

While absolute viscosity is important in scientific processes, it is of little value in fluid power because viscosity effects such as pump cavitation, pressure losses in valving and piping are produced not only by internal fluid friction but by the weight (specific gravity) of the fluid as well. Thus, we express viscosity in kinematic SUS values almost entirely, in the U.S.A.

Kinematic Viscosities						Absolute Viscosity
SUS Saybolt Universal Seconds	SSF Saybolt Seconds Furol	Centistokes	Redwood No. 1 Standard Seconds*	Ford No. 3 Seconds	Engler Specific Degrees**	Centipoises at 0.9 Specific Gravity†
10,000	1000	2200	9000	875	290	1980
9,000	900	1950	8100	788	266	1755
8,000	800	1700	7200	700	236	1530
7,000	700	1500	6300	613	207	1350
6,000	600	1300	5400	525	177	1170
5,000	500	1050	4500	438	148	945
4,000	400	850	3600	350	118	765
3,000	300	630	2700	263	89	567
2000	200	420	1800	175	59	378
1500	150	315	1350	131	44	284
1000	100	220	900	87.5	30	198
900	90	195	810	78.5	27	175
800	80	170	720	70.0	24	153
700	70	150	630	61.3	21	135
600	60	130	540	52.5	18	117
500	50	110	450	43.8	15	99
400	40	87	360	35.0	12	78.3
300	33	65	270	26.3	8.9	58.5
200	24	43	180	17.5	5.9	38.7
100	15	20.8	90	8.8	3.0	18.7
90	——	18.3	81	7.9	——	16.5
80	——	15.8	72	7.0	——	14.2
70	——	13.3	63	6.1	——	12.0
60	——	10.5	54	5.3	——	9.5
55	——	8.9	50	4.8	——	8.0
50	——	7.5	45	4.4	——	6.8
45	——	5.9	41	3.9	——	5.3
40	——	4.3	36	3.5	——	3.9
35	——	2.7	32	3.1	——	2.4

*For Redwood No. 2 Admiralty Seconds viscosity, divide values in this column by 10.

**For Engler viscosity values in seconds, multiply values in this column by 50.

†Absolute viscosity in Centipoises is related to the specific gravity of the fluid. Values in this column are for hydraulic oil of 0.9 specific gravity. For fluids with other values of specific gravity, Centipoise viscosity is found by multiplying values in **Centistokes** column by specific gravity of fluid.

SUS Viscosity Variation with Temperature

Oil Type	Oil Temperature 0°F	20°F	40°F	60°F	80°F	90°F	100°F	110°F	120°F	130°F	140°F	150°F	160°F	180°F	200°F
750 SUS Hvd.Oil	— — —	35,000	10,000	3500	1500	1050	750	550	400	310	240	195	155	110	82
500 SUS Hyd.Oil	55,000	15,000	5000	2000	925	650	500	360	280	220	175	140	115	86	68
300 SUS Hyd.Oil	22,000	6500	2500	1100	550	400	300	230	185	150	122	110	89	69	58
225 SUS Hyd.Oil	12,000	3800	1500	720	380	285	225	165	140	115	100	85	75	61	52
150 SUS Hyd.Oil	5000	1900	850	430	240	190	150	140	100	87	76	68	62	53	47
100 SUS Hyd.Oil	2200	900	440	240	150	120	100	85	74	66	60	55	51	46	43
90 SUS Hyd.Oil	1700	700	360	200	130	105	90	77	68	61	56	52	49	44	42
SAE 10	10,000	3200	1250	600	310	240	180	145	140	100	84	74	66	55	48
SAE 20	27,000	7500	2800	1150	550	400	300	230	180	145	115	99	84	66	55
SAE30	90,000	20,000	6200	2300	1000	680	500	360	280	215	170	135	110	80	64
SAE 40	— — —	45,000	12,000	4000	1600	1200	750	550	400	300	240	185	150	105	77
SAE 50	— — —	— — —	30,000	9000	3200	2000	1400	950	680	500	370	280	210	140	99
Pydraul F9	— — —	19,000	4300	1250	500	325	230	157	125	100	81	70	62	51	45
Pydraul 312	55,000	12,000	3600	1300	600	420	310	230	175	140	110	93	80	63	54
Pydraul 625			60,000	9000	2000	1200	630	400	260	175	125	98	79	58	49
Skydrol 500A	420	240	150	100	75	67	61	56	52	49	47	45	43	41	— — —
Skydrol 7000	1900	650	300	165	105	88	77	68	61	56	52	49	46	42	40
MIL-H-5606A	650	360	220	142	100	88	78	70	64	59	55	52	49	45	42
MIL-L-7808	625	310	175	110	78	69	62	56	52	48	46	43	42	39	
Houghto-Safe 620	4800	2000	980	500	300	240	190	155	130	110	95	83	75	62	54
Kerosene	160	110	80	65	55	52	49	47	45	43	41	40	39		
Diesel Fuel	75	54	45	39	— — —	— — —	— — —	— — —	— — —	— — —	— — —	— — —	— — —	— — —	— — —
JP4 Fuel	47	43	40	— — —	— — —	— — —	— — —	— — —	— — —	— — —	— — —	— — —	— — —	— — —	— — —
SAE 140 Gear Oil	— — —	— — —	55,000	18,000	6500	4000	2700	1900	1350	1000	750	550	420	260	175

Identification of the petroleum base hydraulic oils in the first 7 lines of this chart is by their SUS viscosity ratings at 100°F. This data should be approximately correct for any brand of petroleum oil. The values in the chart were taken from the standard SAE viscosity chart.

SAE ratings usually cover a range of viscosities, and the approximate SUS range covered by each SAE number is shown to the right.

SUS Range Covered By SAE Oil Ratings

SAE No.	SUS Range @ 100°F	SAE No.	SUS Range @ 100°F	SAE No.	SUS Range @ 100°F
5W	30-140	20	170-370	50	810-1300
10W	140-210	30	370-560	60	1300-1600
20W	210-500	40	560-810	70	1600-2100

Oil Pressure Loss Through Pipes

Table 1 shows the pressure loss per 100 feet of Schedule 40 pipe. It is for standard hydraulic oil of 0.9 specific gravity and 220 SUS viscosity. For other specific gravities and viscosities, see information at the bottom of this page.

Table 1. Pressure Loss Per 100 Feet of Schedule 40 Pipe

GPM	Pipe Size*	Pres** Drop	Flow Vel†	GPM	Pipe Size*	Pres** Drop	Flow Vel†	GPM	Pipe Size*	Pres** Drop	Flow Vel†
10	3/8	185	17	35	1/2	249	36	70	3/4	205	42
	1/2	73	11		3/4	83	21		1	63	26
	3/4	24	6.0		1	32	13		1¼	21	15
	1	9.0	3.7		1¼	11	7.5		1½	11	11
	1¼	3.0	2.2		1½	5.7	5.5		2	4.2	6.7
15	1/2	109	16	40	3/4	95	24	80	1	75	31
	3/4	36	9.0		1	36	15		1¼	24	17
	1	14	5.6		1¼	12	8.6		1½	13	13
	1¼	4.5	3.2		1½	6.5	6.3		2	4.8	7.7
	1½	2.4	2.4		2	2.4	3.8		2½	2.3	5.4
20	1/2	146	21	45	3/4	106	27	90	1	80	33
	3/4	47	12		1	41	17		1¼	27	19
	1	18	7.4		1¼	14	9.7		1½	15	4
	1¼	6.0	4.3		1½	4.4	7.1		2	5.4	8.6
	1½	3.2			2	2.7	4.3		2½	2.6	6.0
25	1/2	180	26	50	3/4	122	31	100	1	2	38
	3/4	59	15		1	46	19		1¼	30	22
	1	23	9.3		1¼	15	11		1½	16	16
	1¼	7.6	5.4		1½	8.1	7.9		2	6.0	9.6
	1½	4.0	3.9		2	3.0	4.8		2½	2.9	6.7
30	1/2	214	31	60	3/4	142	36	125	1	114	47
	3/4	71	18		1	53	22		1¼	38	27
	1	27	11		1¼	18	13		1½	20	20
	1¼	9.0	6.4		1½	9.8	9.5		2	7.5	12
	1½	4.8	4.7		2	3.6	5.7		2½	9.8	8.4

Table 2. Conversion Factors for Tubing

For pressure loss per 100 feet of tubing, find tubing I.D. in table below. The next larger NPT size is shown in Column 2. Refer back to Table 1 for pressure loss for this pipe size. Multiply times factor in Column 3 of **Table 2**;

EXAMPLE: For 50 GPM flow through a tube with 1.310 I.D., Column 2 shows 1¼″ NPT to be the next larger pipe size. From **Table 1**, pressure loss is 15 PSI for 1¼″ pipe. Multiply this times the factor from column 3 of **Table 2**: 15 PSI × 1.11 = 16.65 (or 17) PSI pressure loss per 100 feet.

For other schedules of pipe or for hose, flow loss will be in proportion to the inside area of pipe compared to one of the pipe sizes in **Table 1**.

Tube I. D.	Use NPT	Mult. by	Tube I. D.	Use NPT	Mult. by	Tube I. D.	Use NPT	Mult. by
0.334	3/8	2.18	.760	3/4	1.17	1.134	1¼	1.48
0.356	3/8	1.92	.782	3/4	1.11	1.152	1¼	1.43
0.370	3/8	1.77	.810	3/4	1.03	1.260	1¼	1.20
0.384	3/8	1.65	.834	1	1.58	1.282	1¼	1.16
0.402	3/8	1.50	.856	1	1.50	1.310	1¼	1.11
0.416	3/8	1.40	.870	1	1.45	1.334	1¼	1.07
0.430	3/8	1.31	.884	1	1.41	1.356	1¼	1.04
0.532	1/2	1.37	.902	1	1.35	1.370	1¼	1.01
0.506	1/2	1.23	1.010	1	1.08	1.732	2	1.42
0.584	1/2	1.13	1.032	1	1.03	1.760	2	1.38
0.606	1/2	1.08	1.060	1¼	1.69	1.782	2	1.34
0.620	1/2	1.01	1.084	1¼	1.62	1.810	2	1.30
0.634	3/4	1.69	1.106	1¼	1.56	1.834	2	1.27
0.652	3/4	1.60	1.120	1¼	1.52	1.856	2	1.24

For Flows Not Shown: Pressure loss increases approximately in proportion to the increase in flow or flow velocity.

Adjusting for Other Viscosities: Pressure loss through a pipe is directly proportional to fluid viscosity (on SUS of 100 and above). A 440 SUS fluid would have approximately twice the pressure loss shown in the tables.

Adjusting for Specific Gravity: Pressure loss is directly proportional to specific gravity. Water/oil emulsions will have 7% higher, water/glycol fluids will have 14%, and phosphate ester fluids will have 22% higher pressure loss than calculated from the tables.

*Schedule 40 pipe. **PSI loss per 100 feet. †Oil flow velocity, ft/second.

Oil Flow Capacity of Pipes

Schedule 40 (Standard Weight) Pipe

Pipe NPT	2 Ft/Sec. GPM	4 Ft/Sec. GPM	10 Ft/Sec. GPM	15 Ft/Sec. GPM	20 Ft/Sec. GPM	30 Ft/Sec. GPM
1/8	0.35	0.71	1.77	2.66	3.54	5.31
1/4	0.65	1.30	3.24	4.87	6.49	9.73
3/8	1.19	2.38	5.95	8.93	11.9	17.85
1/2	1.89	3.79	9.47	14.21	18.94	28.41
3/4	3.32	6.65	16.62	24.93	33.25	49.87
1	5.39	10.78	26.94	40.41	53.88	80.82
1¼	9.32	18.65	46.62	69.93	93.25	139.87
1½	12.69	25.38	63.46	95.19	126.92	190.38
2	20.92	41.84	104.60	156.90	209.20	313.79
2½	29.85	59.70	149.24	223.86	298.48	447.72
3	46.09	92.18	230.44	345.66	460.88	691.31

Schedule 80 (Extra Strong Weight) Pipe

Pipe NPT	2 Ft/Sec. GPM	4 Ft/Sec. GPM	10 Ft/Sec. GPM	15 Ft/Sec. GPM	20 Ft/Sec. GPM	30 Ft/Sec. GPM
1/8	0.23	0.45	1.13	1.70	2.26	3.40
1/4	0.45	0.89	2.23	3.35	4.47	6.70
3/8	0.88	1.75	4.38	6.57	8.76	13.14
1/2	1.46	2.92	7.30	10.95	14.60	21.90
3/4	2.70	5.39	13.48	20.22	26.96	40.44
1	4.48	8.97	22.42	33.63	44.84	67.26
1¼	8.00	15.99	39.99	59.98	79.97	119.96
1½	11.02	22.03	55.08	82.63	110.17	165.25
2	18.41	36.82	92.04	138.07	184.09	276.13
2½	26.42	52.84	132.11	198.17	264.22	396.33
3	41.18	82.36	205.89	308.84	411.78	617.67

Schedule 160 Pipe

Pipe NPT	2 Ft/Sec. GPM	4 Ft/Sec. GPM	10 Ft/Sec. GPM	15 Ft/Sec. GPM	20 Ft/Sec. GPM	30 Ft/Sec. GPM
1/2	1.05	2.11	5.27	7.91	10.54	15.81
3/4	1.83	3.67	9.17	13.75	18.34	27.51
1	3.25	6.50	16.26	24.39	32.52	48.78
1¼	6.59	13.18	32.94	49.41	65.89	98.83
1½	9.43	18.87	47.17	70.75	94.33	141.50
2	13.93	27.87	69.67	104.51	139.35	209.02
2½	22.11	44.22	110.55	165.83	221.10	331.65
3	33.71	67.43	168.57	252.85	337.13	505.70

Pipe size should be selected on the basis of oil flow velocity. Undersizing results in a high pressure and power loss and system overheating. Oversizing reduces pressure and power losses but may be unnecessarily expensive to plumb.

Pump Suction Lines ...

Schedule 40 pipe should be used and a size chosen which will keep oil velocity within the range of 2 to 4 feet per second.

Oil Return Lines ...

Schedule 40 pipe should be used and a size chosen which will keep oil velocity within the range of 10 to 15 feet per second.

Medium Pressure Lines ...

In those lines carrying 500 to 2000 PSI, flow velocity should be kept at 15 to 20 feet per second. Use Schedule 80 or 160 pipe, or use steel tubing as listed on the next page.

High Pressure Lines ...

Flow velocity may be allowed up to 30 feet per second in lines carrying 3000 to 5000 PSI. Normally, steel tubing is used, but the tables may be used for finding pipe size, then tubing should be selected with the same inside area.

Oil Flow Through Orifices

These charts show PSI pressure drops to be expected in hydraulic oil when flowing through sharp edged orifices. **Caution!** Calculated pressure drops are only approximate because factors such as specific gravity, viscosity, shape of orifice, and plumbing ahead of and following the orifice may cause variations. It is best to make the orifice slightly undersize to start, then to gradually enlarge it while measuring actual pressure drop.

By making the orifice as sharp edged as possible, it becomes less sensitive to oil temperature changes (which affect oil viscosity).

Specific gravity of the fluid significantly influences the pressure drop, which increases approximately as the square of the increase of specific gravity. The charts were calculated for oil with a gravity of 0.9, a close approximation for hydraulic oil. Using other fluids, a multiplying factor must be applied to chart values. For example, to find the pressure drop of water, which has a gravity of 1.00, find the multiplier as follows:

$$(1.00)^2 \div (0.9)^2 = 1.00 \div 0.81 = 1.23 \text{ Multiplying Factor}$$

Therefore, multiply all chart values by 1.23 when calculating for water flow.

These charts were calculated from information supplied by Double A Products Co. The constant, 23.5, shown in the formula below was developed experimentally by measuring pressure drops across average orifices. Values not shown may be calculated from the same basic formula used in calculating the chart:

$$\text{Pressure Drop } (\Delta P) = [\text{GPM} \div (23.5 \times A)]^2$$

Pressure Drop Across Orifices from 3/64″ to 3/16″

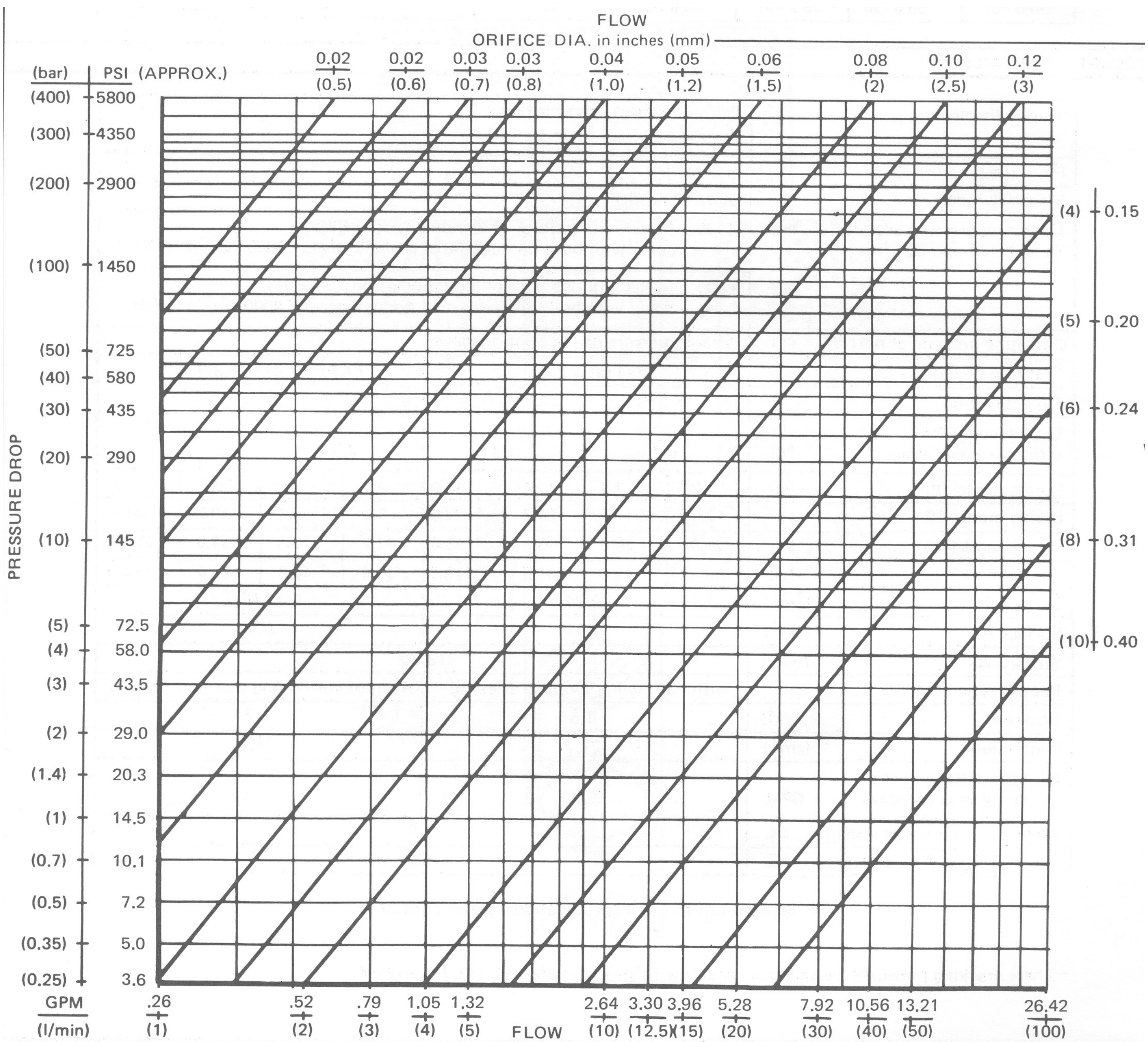
Figures in the body of these charts are PSI pressure drops to be expected in a flow of hydraulic oil across sharp edged orifices of various diameters.

	Orifice Diameters in Inches									
GPM	3/64	1/16	5/64	3/32	7/64	1/8	9/64	5/32	11/64	3/16
3	5445	1730	710	340	185	110	68	44	30	21
4	——	3075	1260	608	328	192	120	79	54	38
5	——	4803	1970	950	513	300	188	123	84	59
7½	——	——	4430	2140	1155	677	422	277	189	134
10	——	——	——	3800	2050	1205	750	493	336	238
12½	——	——	——	——	3205	1880	1175	770	526	371
15	——	——	——	——	4615	2705	1690	1110	757	534
15½	——	——	——	——	——	3685	2300	1510	1030	727
20	——	——	——	——	——	4810	3005	1970	1345	950
22½	——	——	——	——	——	——	3800	2495	1705	1205
25	Chart Values are in PSI						4690	3080	2100	1485
27½	——	——	——	——	——	——	——	3725	2545	1795
30	——	——	——	——	——	——	——	4435	3025	2140
35	——	——	——	——	——	——	——	——	4120	2910
40	——	——	——	——	——	——	——	——	——	3800

Pressure Drop Across Orifices from 13/64″ to 1/2″

	Orifice Diameters in Inches									
GPM	13/64	7/32	15/64	1/4	4/32	5/16	11/32	3/8	7/16	1/2
3	16	12	——	——	——	——	——	——	——	——
4	28	21	16	12	Chart Values are in PSI					
5	43	32	25	19	12	——	——	——	——	——
7½	97	72	55	42	26	17	12	——	——	——
10	172	128	98	75	47	31	21	15	——	——
12½	270	200	153	117	73	48	33	23	13	——
15	388	288	220	169	106	69	47	33	18	11
17½	528	393	300	230	144	94	64	45	25	14
20	690	513	392	301	188	123	84	59	32	19
22½	873	649	496	380	237	156	106	75	41	24
25	1075	800	612	470	293	192	131	93	50	29
27½	1305	970	741	568	355	233	159	112	61	36
30	1550	1155	880	675	420	277	189	134	72	42
35	2115	1570	1200	920	575	377	258	182	98	58
40	2760	2050	1570	1200	751	492	336	237	128	75

Engineering Data
Flow Thru Orifice



Work / Power

HYDRAULIC HORSEPOWER:

a) $\mathbf{HP_T} = \dfrac{\mathbf{GPM \times PSI}}{\mathbf{1714}}$

HP_T = Theoretical HP

b) Input to pump:

$$\mathbf{HP_{in}} = \frac{\mathbf{GPM \times PSI}}{\mathbf{1714 \times (e_t)}}$$

e_t = overall pump efficiency

c) output of Hydraulic Motor:

$$\mathbf{HP_{out}} = \frac{\mathbf{GPM \times PSI \times (e_t)}}{\mathbf{1714}}$$

e_t = overall motor efficiency

HEAT GENERATION:

BTU/hr = 1.5 x GPM x PSI

PSI=Pressure loss which does not produce work

HEAT RADIATION OF A HYDRAULIC RESERVOIR:

BTU/hr = 2.54 (Av) (△T)

BTU/hr	=	Heat radiated
Av	=	Vertical tank area in contact with oil
△T	=	Desired oil temp minus ambient air temperature in degrees Fahrenheit

ESTIMATING IMMERSION HEATERS:

$$\mathbf{KW} = \frac{\mathbf{V \times \Delta T}}{\mathbf{800\ T}}$$

V	=	Tank capacity gallons
△T	=	(desired - ambient) temperature in degrees Fahrenheit
T	=	Time in hours
KW	=	Input heat required.

POWER:

1 HP = 1.014 metric HP
1 HP = .7457 KW
1 HP =42.4 BTU/min
1 HP = 2545 BTU/Hr.
1 HP = 550 ft - lb/sec

ELECTRICAL FORMULAS

To Find	Alternating Current	
	Single-Phase	Three-Phase
Amperes when horsepower is known	$\frac{HP \times 746}{E \times Eff \times pf}$	$\frac{HP \times 746}{1.73 \times E \times Eff \times pf}$
Amperes when kilowatts are known	$\frac{Kw \times 1000}{E \times pf}$	$\frac{Kw \times 1000}{1.73 \times E \times pf}$
Amperes when kva are known	$\frac{Kva \times 1000}{E}$	$\frac{Kva \times 1000}{1.73 \times E}$
Kilowatts	$\frac{I \times E \times pf}{1000}$	$\frac{1.73 \times I \times E \times pf}{1000}$
Kva	$\frac{I \times E}{1000}$	$\frac{1.73 \times I \times E}{1000}$
Horsepower = (Output)	$\frac{I \times E \times Eff \times pf}{746}$	$\frac{1.73 \times I \times E \times Eff \times pf}{746}$

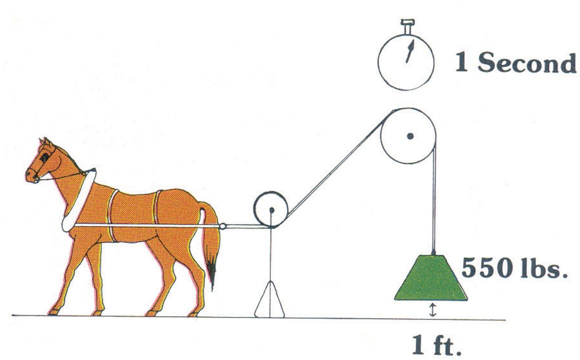
I = Amperes; E = Volts; Eff = Efficiency;
pf = Power factor; Kva = Kilovolt-amperes;
Kw = Kilowatts.

Power is defined as the rate of doing work. To better describe this term we will use the example we cited earlier. Assuming the book weighs 1 pound and we lift it 3 feet off the table we have done 3 ft.-lbs. of work. It does not matter if we lift it fast (1 second) or slow (1 hour), we always do the same amount of work. It does, however, take more power to lift the book in a lesser amount of time. Consequently, the units of power are defined as the amount of work (ft.-lbs.) per unit time (seconds) or:

$$\mathbf{POWER} = \frac{\mathbf{ft\text{-}lbs.}}{\mathbf{sec.}}$$

The common method of measuring power is known as *horsepower*. Horsepower is defined as the amount of weight (lbs.) a horse could lift one *foot* in one *second*. By experiment it was found that the average horse could lift 550 lbs. one foot in one second, consequently:

$$\mathbf{1\ Horsepower} = \frac{\mathbf{550\ ft.\text{-}lbs.}}{\mathbf{sec.}}$$



Power To Heat

HEAT:

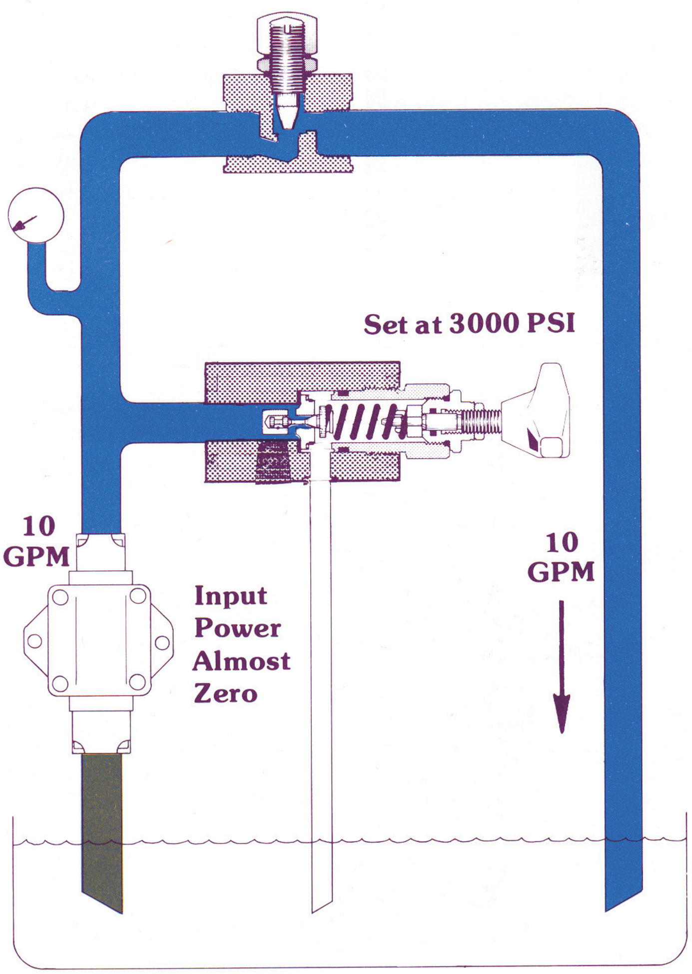
1 BTU = 778 ft - lbs.

HP & HEAT RELATIONSHIPS:

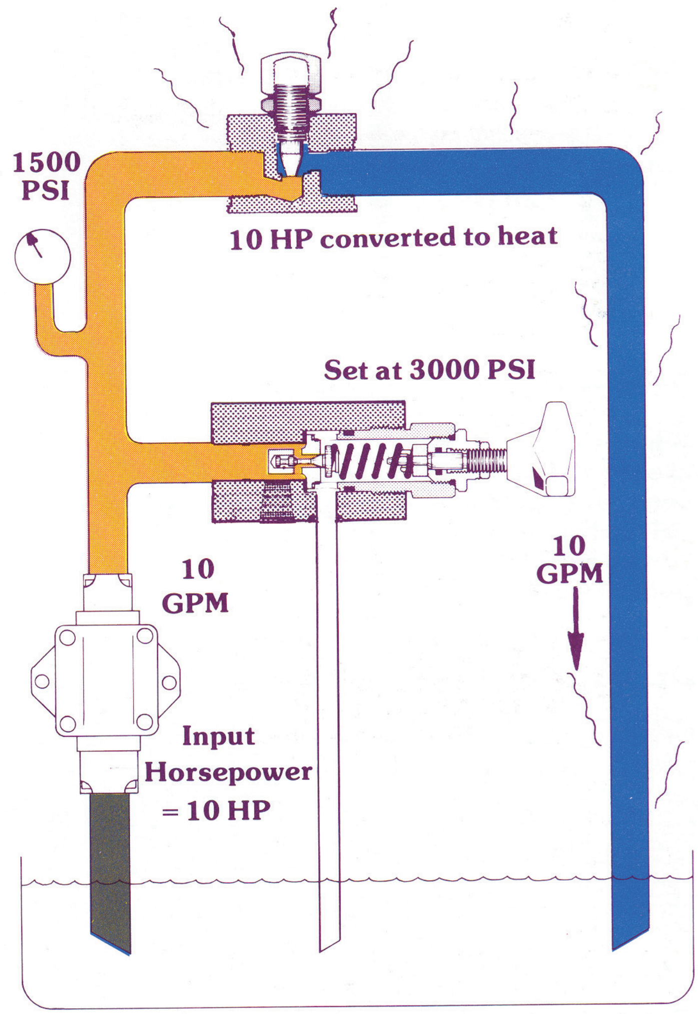
$$1\ \text{HP} = \frac{42.4\ \text{BTU}}{\text{min}} = \frac{2545\ \text{BTU}}{\text{hr.}}$$

$$= 746\ \text{Watts} = .746\ \text{KW}.$$

$$Hp = \frac{(P) \times (Q)}{1714}$$



When the restriction is wide open there is virtually no resistance to flow, therefore, no pressure.



As the needle valve is closed, pressure builds up in the system and energy is converted to heat (since no useful work is being done). All the flow is still across the restriction since pressure isn't high enough to open the relief.

How Much Power ...

Pressure Washers

NorthStar™

High Performance Pressure Washers

High Performance Pressure Washers

NorthStar™ high performance pressure washers combine the power of premium quality Honda engines with industrial direct drive pumps that are tested at our factory. **Never before has anyone offered this level of performance and quality at these affordable prices!**

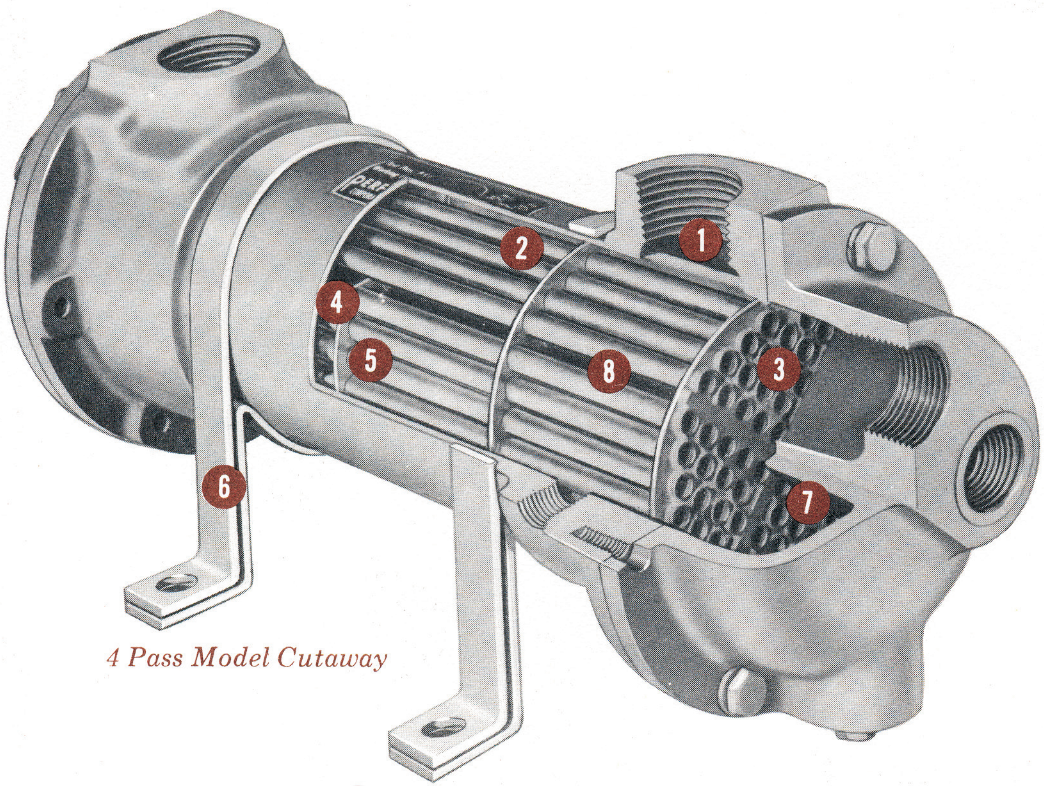
High Performance Features:

- Powered by premium Honda OHV engine featuring cast iron sleeve and low oil shutdown
- Long life Triplex crankshaft pump; thermal protector caps
- Extended push down handle for easy mobility
- 30-ft. non-marking yellow hose
- 25° quick couple nozzle & chemical nozzle
- 10" high pneumatic tires; built-in injector and unloader

Item#	Engine	GPM/PSI	Ship Wt.	Discount Price
157811-B979	5.5 HP Honda	2.5/2500	104 lbs.	$629.99
157813-B979	9 HP Honda	3/3000	123 lbs.	$799.99
157815-B979	13 HP Honda	4/3500	169 lbs.	$999.99

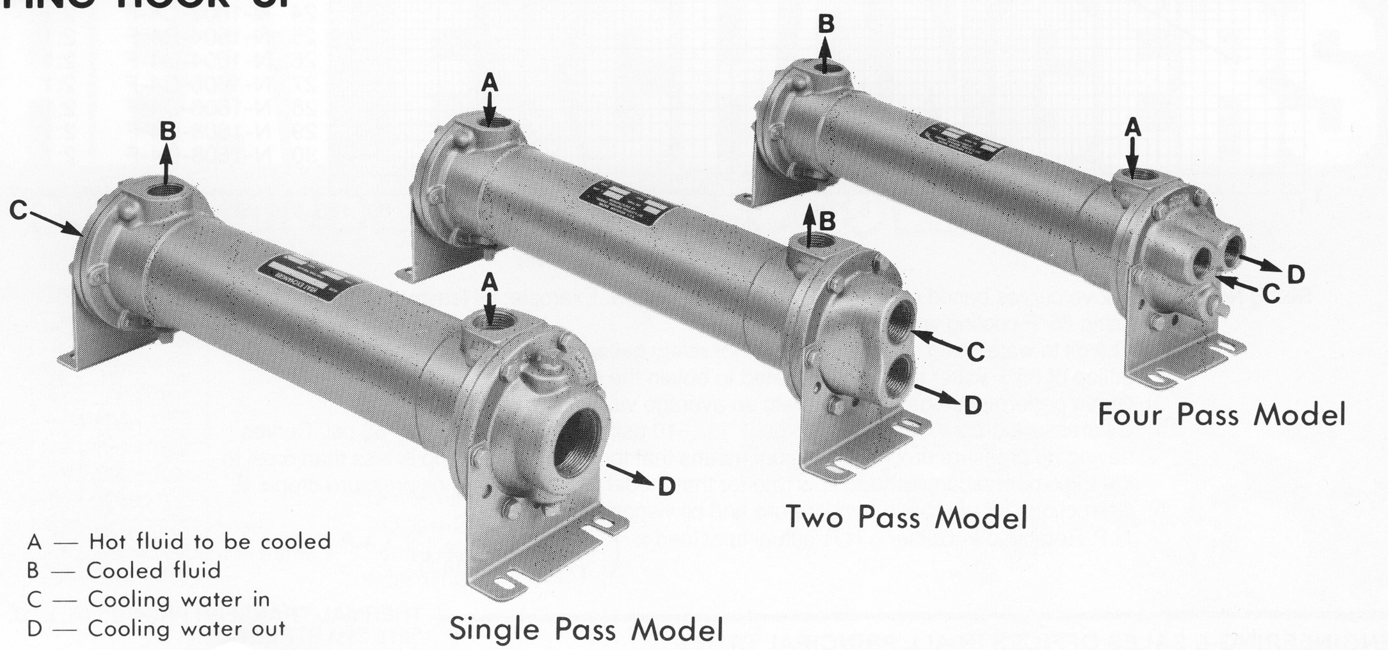
Engine	GPM/PSI
5.5 HP Honda	2.5/2500
9 HP Honda	3/3000
13 HP Honda	4/3500

Heat Exchanger



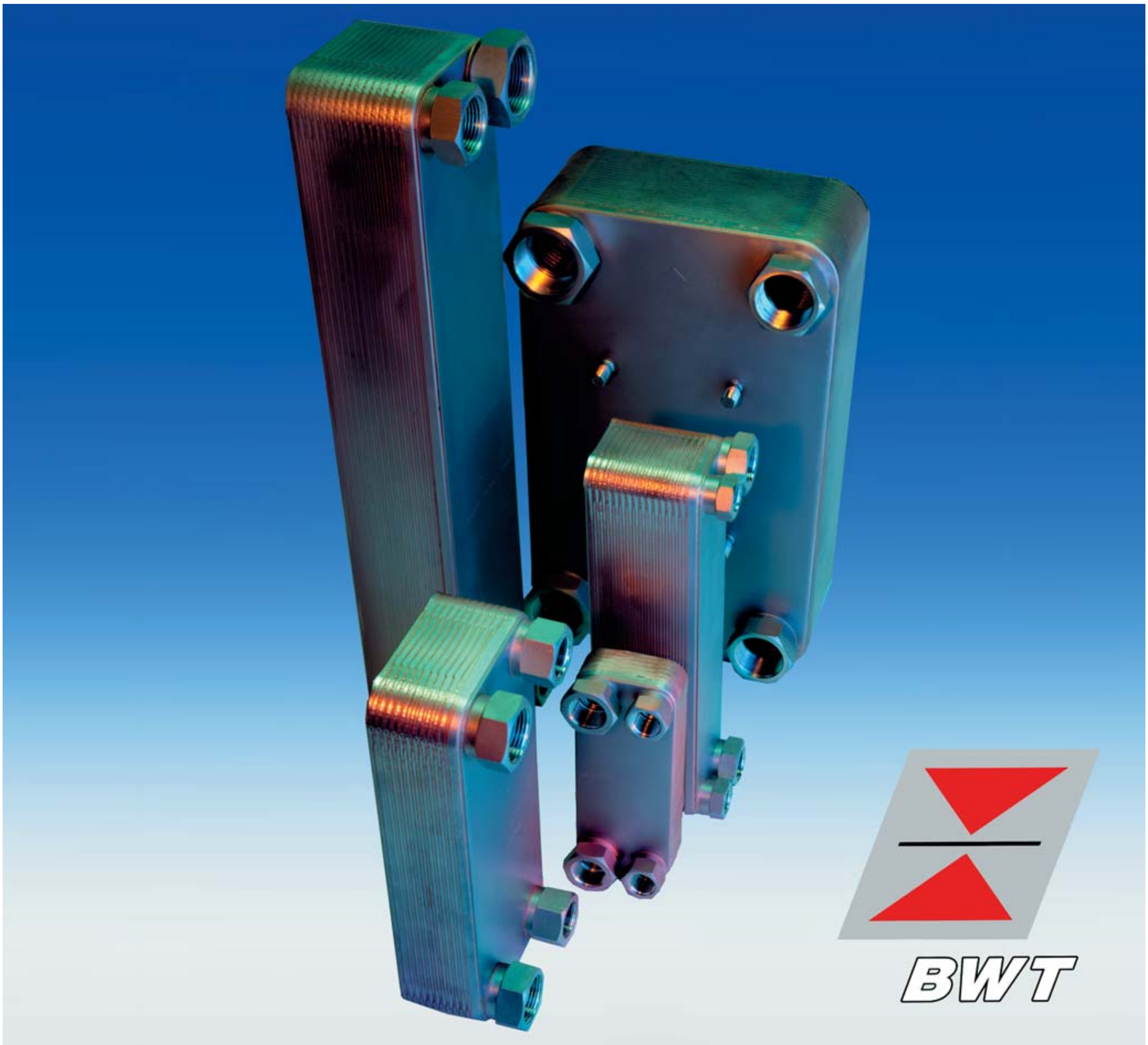
4 Pass Model Cutaway

PIPING HOOK UP



A — Hot fluid to be cooled
B — Cooled fluid
C — Cooling water in
D — Cooling water out

Oil / Water Cooler
Series BWT

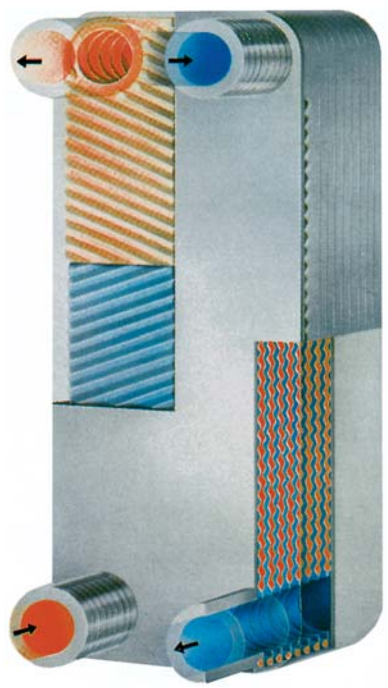


- High exchange efficiency
- Equally distributed turbulent flow
- Little installation space required
- High fatigue life
- Low water consumption
- Maintenance free
- Broad temperature range
- Easy installation

Bühler Technologies GmbH
D - 40880 Ratingen, Harkortstr. 29
Tel.: + 49 (0) 2102 / 4989-0 Fax: + 49 (0) 2102 / 4989-20
e-mail: fluidcontrol@buehler-ratingen.com

Why Coolers?

There are basically two main concepts in the development of fluid power systems. One is to design systems minus a cooler and if the operational conditions show in practice that the system needs a cooler to install it later.
This however requires compromises that usually result in financial overspend.
The other concept recognises that a system originally designed with an integrated cooler can be built more compact, needs less installation space and runs more reliable due to the stabilized temperature of the fluid.

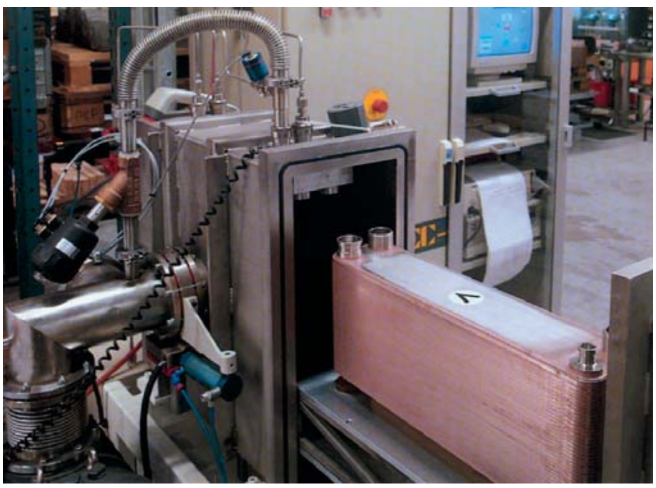


Why Bühler?

Since water is becoming a precious resource, significantly reduced water consumption is favoured by the system designers. After over 25 years experience in design and sales of traditional tube and shell heat exchangers Bühler recognised that a new concept was required to meet the increasing demand for water conservation.
The plate heat exchanger fulfills this requirement particularly for the fluid power market.
In cooperation with a well-known international manufacturer of plate heat exchangers, Bühler has developed a comprehensive range of braced plate coolers specifically for fluid power applications. Bühler has been offering this new concept of oil / water coolers now for over five years with increasing recognition and success.

If our comprehensive standard range of products does not have an answer for your application we will be pleased to find special solutions for your application.

The data contained in this leaflet is sufficient to determine the right cooler for your application. However, we can offer you a software which makes this sizing easier.



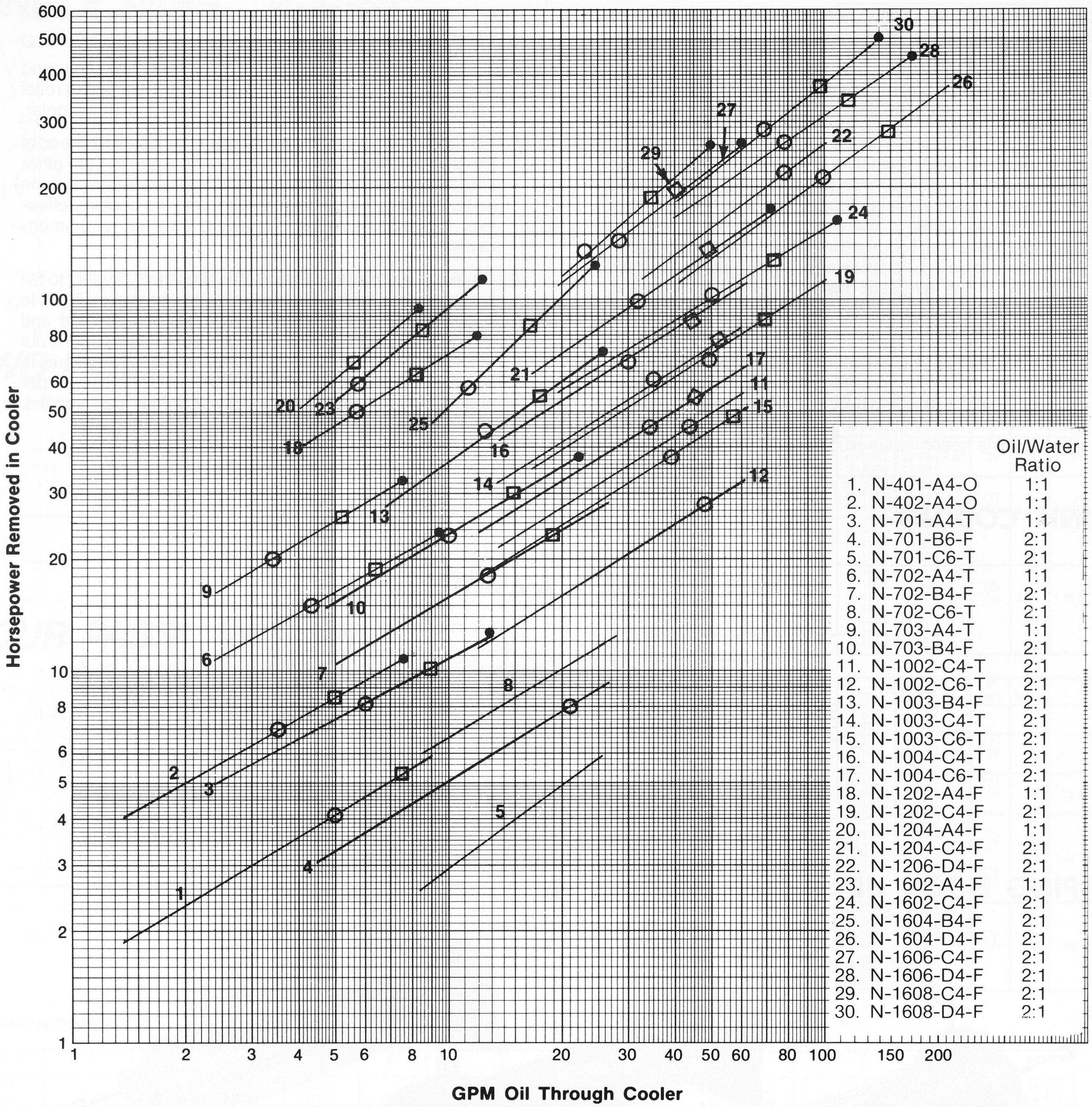
Description

The BWT oil / water coolers consist of a number of profiled stainless steel plates. The direction of the profile is reversed on every other plate so that the ridges on adjacent plates intersect with another forming a network of contact points. The subsequent brazing process creates a very compact and pressure resistant package, which virtually utilizes all material for heat transfer.
Compared with traditional systems the complex geometry of the BWT plate cooler provides a highly turbulent flow with very equal distribution resulting in an outstanding heat transfer efficiency. Even at lower velocities a turbulent flow is insured which is constantly changing direction due to the profile and thus disturbing the boundary layer.
BWT plate coolers are much less prone to fouling than coolers of traditional design thanks to the smooth surface quality of the cooler plates and the turbulent flow. Experience shows that fouling is not a problem in plate coolers providing the application parameters have been indicated correctly.

Heat Exchanger Selection

PERFORMANCE

HEAT ENERGY DISSIPATION RATES



Sizing Notes:
1) Above curves based on 40°F approach temperature. Example: oil leaving cooler at 125°F using 85°F cooling water (125 - 85 = 40).
2) The oil to water ratio of 2:1 means that for every gallon of oil circulated, a minimum of 1/2 gallon of 85°F water must be circulated to obtain the curve results.
3) Curve performance based on oil with an average viscosity of 100 SSU.
4) Oil pressure drop coding: ○ = 5 psi; □ = 10 psi; ● = 20 psi; △ = 50 psi. Curves having no pressure drop code symbol means that the oil pressure drop is less than 5 psi to the maximum recommended flow rate for that model cooler. Multiply oil pressure drop x B.
5) Corrections for approach temperature and oil viscosity:

$$\text{H.P. Removed in Cooler} = \text{H.P. actual heat load} \times \left(\frac{40}{\text{Actual Approach}} \right) \times A$$

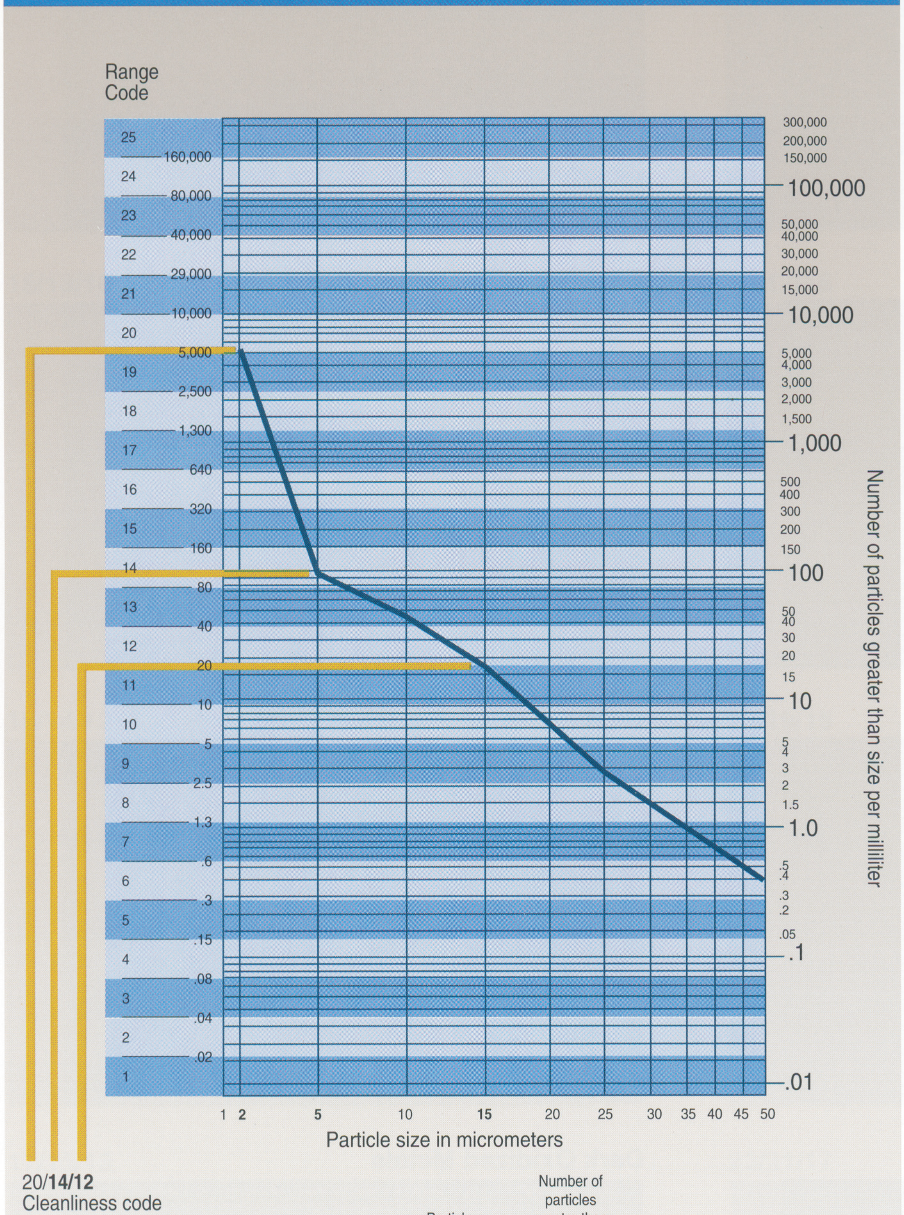
VISCOSITY CORRECTIONS		
Avg. Oil SSU	A	B
50	.84	.83
100	1.0	1.0
200	1.14	1.3
300	1.24	1.7
400	1.31	2.1
500	1.37	2.5
600	1.42	2.8
700	1.46	3.2
800	1.50	3.6
900	1.53	4.0
1000	1.56	4.4

Filtration

Cleanliness Code Graph

The cleanliness code graph uses the industry standard for measuring and depicting the amount and size of particles per milliliter in hydraulic fluid. The ISO 4406 standard requires a log-log^2 graph that charts the amount of particles greater than certain micron sizes per milliliter of fluid. The 5µm and 15 µm measurements are converted to a range code and then used together and referred to as the ISO code for a particular sample *(example 14/12)*. The Vickers Cleanliness Code is similar to the ISO

Cleanliness Code Graph



Example

Particle size "X" in micrometers	Number of particles greater than "X" size in one ml of test fluid	Range code
2	5120	20
5	89	14
10	43	X
15	22	12
25	3	X
50	.4	X

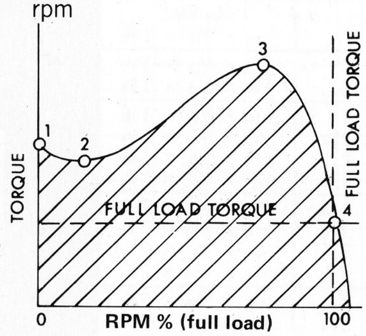
Electrical Engineering Data

TORQUE: NEMA DESIGN CLASSES

Full-load torque on lb-ft $= \dfrac{\text{HP} \times 5250}{\text{rpm}}$

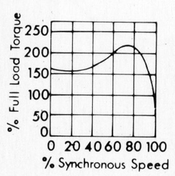
Motor torque is defined at four points as shown by Figure 1.

1. Breakaway or starting
2. Minimum or "pull-up"
3. Breakdown or "pull-out"
4. Full load

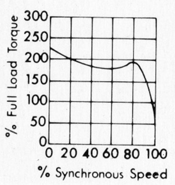


The sectioned area under the curve represents a motor's accelerating torque from zero to full speed. Torque, horsepower, and speed requirements demanded in drives for most machines can be met with one of four design classes of squirrel-cage polyphase induction motors described by National Electrical Manufacturers Association standards.

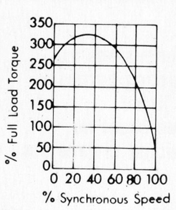
All four designs are suitable for across-the-line starting.



Design B Motors are the standard general purpose design. They have low starting current-normal torque, and normal slip. Their field of application is very broad and includes fans, blowers, pumps, and machine tools. Torque values are listed in this Modification Section.

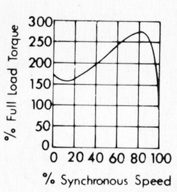


Design C Motors have high breakaway torque, low starting current, and normal slip. The higher breakaway torque makes this motor advantageous for "hard-to-start" applications, such as plunger pumps, conveyors, and compressors.



Design D Motors have a high breakaway torque combined with high slip. Breakaway torque for 4,6 and 8 pole motors is 275% or more of full load torque. Two slip groups are described below.

5-8% and 8-13% Slip Design D Motors are recommended for punch presses, shears, and other high inertia machinery, where it is desired to make use of the energy stored in a flywheel under heavy fluctuating load conditions. They are also used for multimotor conveyor drives where motors operate in mechanical parallel.



Design A covers a wide variety of motors similar to Design B except that their breakdown torque and starting current are higher.

HIGH INERTIA LOADS

$$t = \frac{WK^2 \times \text{rpm}}{308 \times T\text{ av.}}$$

WK^2 = inertia in lb-ft^2

t = accelerating time in sec.
T = Av. accelerating torque lb-ft

$$T = \frac{WK^2 \times \text{rpm}}{308 \times t}$$

$$\text{Inertia reflected to motor} = \text{Load inertia}\left(\frac{\text{Load rpm}}{\text{Motor rpm}}\right)^2$$

RULES OF THUMB (Approximation)

At 1800 rpm, a motor develops a 3 lb-ft per hp
At 1200 rpm, a motor develops a 4.5 lb-ft per hp
At 575 volts, a 3-phase motor draws 1 amp per hp
At 460 volts, a 3-phase motor draws 1.25 amp per hp
At 230 volts, a 3-phase motor draws 2.5 amp per hp
At 230 volts, a single-phase motor draws 5 amp per hp
At 115 volts, a single-phase motor draws 10 amp per hp

MECHANICAL FORMULAS

$$\text{Torque in Lb-ft} = \frac{\text{HP} \times 5250}{\text{rpm}} \qquad \text{HP} = \frac{\text{Torque} \times \text{rpm}}{5250}$$

$$\text{rpm} = \frac{120 \times \text{Frequency}}{\text{No. of Poles}}$$

ELECTRICAL FORMULAS

To Find	Alternating Current	
	Single-Phase	Three-Phase
Amperes when horsepower is known	$\frac{\text{HP} \times 746}{\text{E} \times \text{Eff} \times \text{pf}}$	$\frac{\text{HP} \times 746}{1.73 \times \text{E} \times \text{Eff} \times \text{pf}}$
Amperes when kilowatts are known	$\frac{\text{Kw} \times 1000}{\text{E} \times \text{pf}}$	$\frac{\text{Kw} \times 1000}{1.73 \times \text{E} \times \text{pf}}$
Amperes when kva are known	$\frac{\text{Kva} \times 1000}{\text{E}}$	$\frac{\text{Kva} \times 1000}{1.73 \times \text{E}}$
Kilowatts	$\frac{\text{I} \times \text{E} \times \text{pf}}{1000}$	$\frac{1.73 \times \text{I} \times \text{E} \times \text{pf}}{1000}$
Kva	$\frac{\text{I} \times \text{E}}{1000}$	$\frac{1.73 \times \text{I} \times \text{E}}{1000}$
Horsepower = (Output)	$\frac{\text{I} \times \text{E} \times \text{Eff} \times \text{pf}}{746}$	$\frac{1.73 \times \text{I} \times \text{E} \times \text{Eff} \times \text{pf}}{746}$

I = Amperes; E = Volts; Eff = Efficiency;
pf = Power factor; Kva = Kilovolt-amperes;
Kw = Kilowatts.

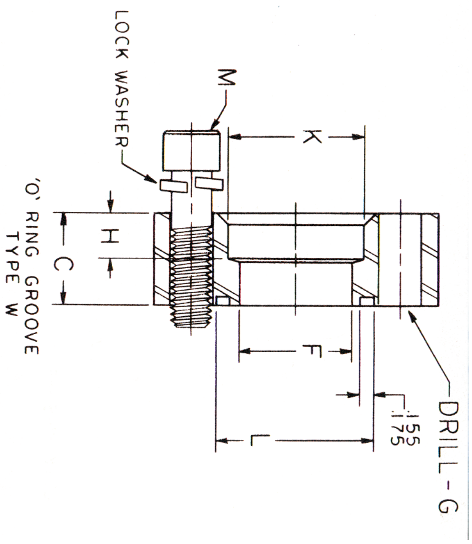
TEMPERATURE CONVERSION

Deg C = (Deg F − 32) × 5/9
Deg F = (Deg C × 9/5) + 32

Pipe Flow & Pressure

Nominal Pipe Size	Outside Diam Inches	Identification Iron	Identification Sch	Wall Thickness Inches	Inside Dia Inches	Working Pressure P.S.I.	Burst Pressure P.S.I.	Flow at Standard Velocities Q = G.P.M. 3 Ft/sec	10 Ft/sec	15 Ft/sec	20 Ft/sec	25 Ft/sec
3/4"	1.050	Std	40	.113	.824	1,270	12,910	5.0	17	25	33	42
		XS	80	.154	.742	2,680	17,600	4.0	13	20	27	34
		—	160	.219	.612	5,100	24,690	2.8	9	14	18	23
		XXS	—	.308	.434	8,810	35,200	1.4	5	7	9	12
1"	1.315	Std	40	.133	1.049	1,540	12,140	8.1	27	40	54	67
		XS	80	.179	.957	2820	16,330	6.7	22	34	45	56
		—	160	.250	.815	4,930	22,810	4.9	16	24	33	41
		XXS	—	.358	.599	8,500	32,670	2.6	9	13	18	22
1¼"	1.660	Std	40	.140	1.380	1,360	10,120	14.0	42	70	93	117
		XS	80	.191	1.278	2,470	13,810	12.0	40	60	80	100
		—	160	.250	1.160	3,820	18,070	9.9	33	50	66	82
		XXS	—	.382	.896	7120	27,610	5.9	20	30	39	49
1½"	1.900	Std	40	.145	1.610	1,280	9,160	19	64	95	127	159
		Stc	80	.200	1.500	2,320	12,630	17	55	83	110	138
		—	160	.281	1.338	3,940	17,750	13	44	66	88	110
		XXS	—	.400	1.100	6,510	25,260	9	30	45	59	74
2"	2.375	Std	40	.154	2.067	1,150	7,780	31	105	157	209	262
		XS	80	.218	1.939	2,110	11,010	28	92	138	184	230
		—	160	.344	1.687	4,120	17,380	21	70	105	140	174
		XXS	—	.436	1.503	5,700	22,030	17	55	83	110	138
2½"	2.875	Std	40	.203	2.469	1,550	8,470	45	149	224	299	373
		XS	80	.276	2.323	2,470	11,520	40	132	198	265	331
		—	160	.375	2.125	3,770	15,650	33	111	166	221	277
		XXS	—	.552	1.771	6,290	23,040	23	77	115	154	192
3"	3.500	Std	40	.216	3.068	1,390	7,410	69	231	346	461	577
		XS	80	.300	2.900	2,260	10,290	62	206	309	412	515
		—	160	.438	2.624	3,750	15,020	51	169	253	337	422
		XXS	—	.600	2.300	5,610	20,680	40	130	194	259	324
3½"	4.000	Std	40	.226	3.548	1,970	6780	93	309	463	617	771
		XS	80	.318	3.364	2,810	9540	83	277	416	555	693
		—	160	—	—	—	—	—	—	—	—	—
		XXS	—	—	—	—	—	—	—	—	—	—
4"	4.500	Std	40	.237	4.026	1830	6,320	119	397	596	794	993
		XS	80	.337	3.826	2640	8,990	108	359	538	717	897
		—	160	.531	3.438	4300	14,160	86	287	431	574	718
		XXS	—	.674	3.152	5590	17,970	73	243	365	487	609
5"	5.563	Std	40	.258	5.047	1600	5570	187	624	936	1248	1560
		XS	80	.374	4.813	2370	8090	170	568	852	1135	1419
		—	160	.625	4.313	4080	13480	137	456	684	911	1140
		XXS	—	.750	4.063	4980	16,180	121	405	607	809	1011
6"	6.625	Std	40	.280	6.065	1460	5070	270	901	1352	1803	2253
		XS	80	.432	5.761	2280	7820	244	813	1220	1627	2033
		—	160	.719	5.187	3930	13,020	198	659	989	1319	1648
		XXS	—	.864	4.897	4800	15,660	176	588	881	1175	1469
8"	8.625	Std	40	.322	7.981	1280	4480	468	1560	2341	3121	3900
		XS	80	.500	7.625	2020	6969	427	1425	2137	2850	3560
		—	160	* .906	6.813	3790	12,600	341	1137	1706	2275	2844
		XXS	—	* .875	6.875	3650	12,140	348	1158	1737	2316	2896

Bolt Torque



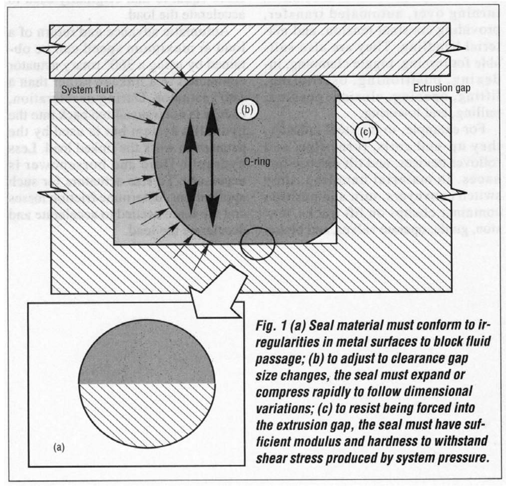


Fig. 1 (a) Seal material must conform to irregularities in metal surfaces to block fluid passage; (b) to adjust to clearance gap size changes, the seal must expand or compress rapidly to follow dimensional variations; (c) to resist being forced into the extrusion gap, the seal must have sufficient modulus and hardness to withstand shear stress produced by system pressure.

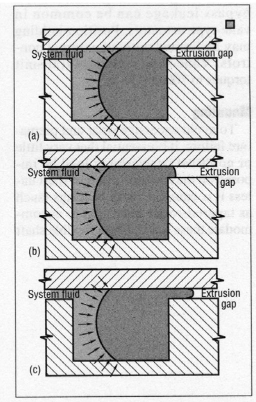


Fig. 2. As system fluid pressure increases, (a) to (b), an O-ring seal is progressively forced into the extrusion gap. Finally, (c), the physical limits of the seal material have been exceeded.

TORQUE INFORMATION

WHAT IS TORQUE?

Torque, by definition, is the result of a force applied to an object through a lever arm, thus tending to rotate the object.

$$T = F \times L$$

T — Torque
F — Applied Force
L — Lever length measured from the center of rotation to, and at 90° to, the direction of force.

Since both force and length can be expressed in many different units of measurements, so can torque. However, the most common units are: Inch pound (in-lb or lb-in), foot pound (ft-lb or lb-ft), meter kilogram (mkg) and Newton meter (N•m).

When torque is applied to a threaded fastener, it produces a clamping force that holds the components together. Too much force, and the fastener will break. Not enough, and the assembly will not stay together. By controlling the amount of torque, the clamping or holding force is controlled.

WHY IS TORQUE IMPORTANT?

SAFETY: Bolts or nuts which are not tightened enough may vibrate loose, while overtightened ones may break.

ECONOMY: Improperly tightened components may cause damage or accelerated wear. "Blown out" gaskets and broken head bolts are typical examples of such costly errors.

PERFORMANCE: Today's equipment is made of many precision parts which need to be assembled just right to achieve maximum efficiency and performance. Improperly tightened head bolts may result in poor compression, overtightened bearings may bind, etc.

GENERAL CONVERSION TABLE FOR TORQUE UNITS

To OBTAIN / MULTIPLY NUMBER OF	Inch Ounces	Inch Pounds	Foot Pounds	Centimeter Kilograms	Meter Kilograms	Newton Meters
Inch Ounces	1	16	192	13.89	1389	141.6
Inch Pounds	.0625[1]	1	12	.8680	86.80	8.851
Foot Pounds	.005208	.08333[2]	1	.07233	7.233	.7376
Centimeter Kilograms	.07201	1.152	13.83	1	100	10.20
Meter Kilogram[3]	.0007201	.01152	.1383	.01	1	.1020
Newton-Meters	.007061	.1130	1.356	.09806	9.806	1

[1]or divide by 16
[2]or divide by 12
[3]Meter Kilogram (mkg) is also known as Meter Kilopond (mkp)

GENERAL TORQUE SPECIFICATION CHART FOR ENGLISH FASTENERS (in Foot Pounds)*

BOLT SIZE / MATERIAL OR GRADE	SAE 2 (Mild Steel)	SAE 5	SAE 8	Socket Head Cap Screws	Brass	Stainless AISI Type 303
1/4-20	6	11	12	13	5	5
1/4-28	7	13	15	16	6	7
5/16-18	13	21	25	27	8	9
5/16-24	14	23	30	33	9	10
3/8-16	23	38	50	52	15	17
3/8-24	26	40	60	60	16	18
7/16-14	37	55	85	86	23	25
7/16-20	41	60	95	95	25	28
1/2-13	57	85	125	130	32	37
1/2-20	64	95	140	145	34	40
9/16-12	80	125	175	180	44	50
9/16-18	91	140	195	210	48	54
5/8-11	111	175	245	255	68	75
5/8-18	128	210	270	290	73	80

GENERAL TORQUE SPECIFICATION CHART FOR METRIC FASTENERS (in Newton Meters)*

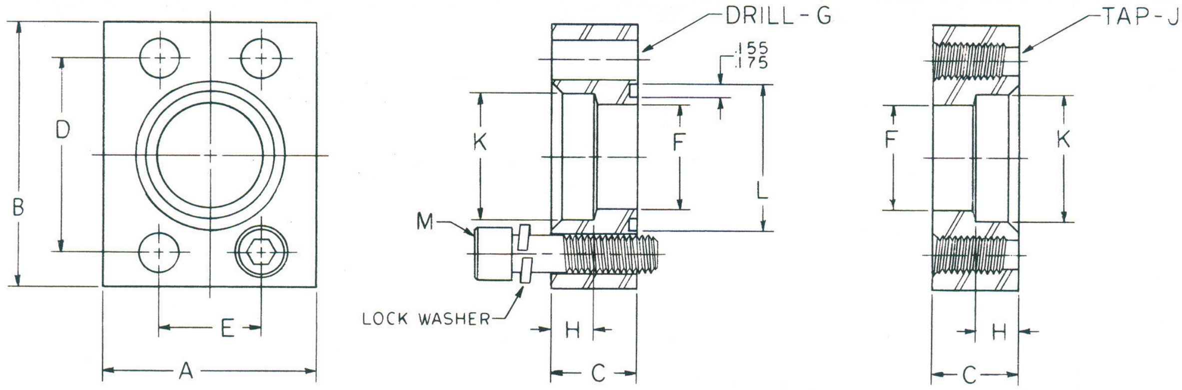
BOLT DIAM. MM	BOLT DIAM. INCH	4.6	4.8	5.8	8.8	9.8	10.9	12.9
5	.197	3	4	5	7	8	11	12
6	.236	5	6	8	12.5	14	17	20
6.3	.248	5.5	8	9.5	14	16	21	24
8	.315	12	16	20	30	34	44	50
10	.394	23	32	40	60	70	85	100
12	.472	40	56	70	103	120	150	180
14	.551	65	90	110	167	190	240	280
16	.630	100	140	170	270	290	380	440
18	.709	137	177	225	350	—	480	580
20	.787	200	—	330	520	—	740	860

*These torque values are approximate and should not be accepted as accurate limits. Indeterminant factors (surface finish, type of plating and lubrication) in specific applications preclude the publication of accurate values for universal use. Manufacturers of various types of equipment usually provide specific tightening instructions which should be followed. **DO NOT USE** the above values for gasketed joints or joints of soft materials. **DO NOT USE** your torque wrench for values greater than its maximum scale reading.

Flange Selection

SOCKET WELD STRAIGHT PIPE FLANGES

SAE BOLT PATTERN



'O' RING GROOVE TYPE W

COMPANION TYPE WC

3000 SERIES PIPE

PIPE SIZE	FLANGE NO.	A	B	C	D	E	F	G	H	J	K	L	MOUNTING KITS[1] M	NO.	'O' RING	WT.
1/2	WSAEP-04-20	1.50	2.12	.75	1.500	.688	.50	.34	.38	—	.860	1.000	5/16-18 x 1 1/4	MK 501	210	0.6
	WCSAEP-04-20	1.50	2.12	.75	1.500	.688	.50	—	.38	5/16-18	.860	—	—	—	—	0.6
3/4	WSAEP-06-20	1.75	2.62	1.00	1.875	.875	.75	.41	.50	—	1.070	1.250	3/8-16 x 1 3/4	MK 502	214	1.0
	WCSAEP-06-20	1.75	2.62	1.00	1.875	.875	.75	—	.50	3/8-16	1.070	—	—	—	—	1.0
1	WSAEP-08-20	2.00	2.82	1.00	2.062	1.031	1.00	.41	.50	—	1.335	1.562	3/8-16 x 1 3/4	MK 503	219	1.2
	WCSAEP-08-20	2.00	2.82	1.00	2.062	1.031	1.00	—	.50	3/8-16	1.335	—	—	—	—	1.2
1 1/4	WSAEP-10-20	2.50	3.19	1.00	2.312	1.188	1.25	.47	.50	—	1.680	1.750	7/16-14 x 1 3/4	MK 504	222	1.7
	WCSAEP-10-20	2.50	3.19	1.00	2.312	1.188	1.25	—	.50	7/16-14	1.680	—	—	—	—	1.7
1 1/2	WSAEP-12-20	2.75	3.75	1.25	2.750	1.406	1.50	.53	.50	—	1.920	2.125	1/2-13 x 2	MK 505	225	2.3
	WCSAEP-12-20	2.75	3.75	1.25	2.750	1.406	1.50	—	.50	1/2-13	1.920	—	—	—	—	2.3
2	WSAEP-16-20	3.25	4.00	1.50	3.062	1.688	1.94	.53	.62	—	2.411	2.500	1/2-13 x 2 1/4	MK 506	228	3.5
	WCSAEP-16-20	3.25	4.00	1.50	3.062	1.688	1.94	—	.62	1/2-13	2.411	—	—	—	—	3.5
2 1/2 [2]	WSAEP-20-20	4.00	4.50	1.75	3.500	2.000	2.38	.53	.75	—	2.911	3.000	1/2-13 x 2 1/2	MK 507	232	5.7
	WCSAEP-20-20	4.00	4.50	1.75	3.500	2.000	2.38	—	.75	1/2-13	2.911	—	—	—	—	5.7
3 [3]	WSAEP-24-20	4.50	5.31	2.00	4.188	2.438	2.94	.66	.75	—	3.540	3.625	5/8-11 x 3	MK 508	237	9.2
	WCSAEP-24-20	4.50	5.31	2.00	4.188	2.438	2.94	—	.75	5/8-11	3.540	—	—	—	—	9.2
3 1/2 [4]	WSAEP-28-20	5.00	6.00	1.50	4.750	2.750	3.38	.66	.75	—	4.045	4.109	5/8-11 x 2 1/2	MK 509	241	7.2
	WCSAEP-28-20	5.00	6.00	1.50	4.750	2.750	3.38	—	.75	5/8-11	4.045	—	—	—	—	7.2
4 [4]	WSAEP-32-20	5.50	6.38	1.50	5.125	3.062	3.88	.66	.75	—	4.550	4.609	5/8-11 x 2 1/2	MK 510	245	8.5
	WCSAEP-32-20	5.50	6.38	1.50	5.125	3.062	3.88	—	.75	5/8-11	4.550	—	—	—	—	8.5
5 [4]	WSAEP-40-20	7.00	7.25	2.00	6.000	3.625	4.81	.66	.75	—	5.625	5.609	5/8-11 x 3	MK 511	253	14.5
	WCSAEP-40-20	7.00	7.25	2.00	6.000	3.625	4.81	—	.75	5/8-11	5.625	—	—	—	—	14.5

6000 SERIES PIPE

PIPE SIZE	FLANGE NO.	A	B	C	D	E	F	G	H	J	K	L	MOUNTING KITS[1] M	NO.	'O' RING	WT.
1/2	WSAEP-04-60	1.75	2.22	1.00	1.594	.718	.50	.34	.50	—	.860	1.000	5/16-18 x 1 1/2	MK 512	210	0.8
	WCSAEP-04-60	1.75	2.22	1.00	1.594	.718	.50	—	.50	5/16-18	.860	—	—	—	—	0.8
3/4	WSAEP-06-60	2.00	2.81	1.00	2.000	.937	.75	.41	.50	—	1.070	1.250	3/8-16 x 1 3/4	MK 502	214	1.2
	WCSAEP-06-60	2.00	2.81	1.00	2.000	.937	.75	—	.50	3/8-16	1.070	—	—	—	—	1.2
1	WSAEP-08-60	2.25	3.19	1.00	2.250	1.093	1.00	.47	.50	—	1.335	1.562	7/16-14 x 1 3/4	MK 513	219	1.5
	WCSAEP-08-60	2.25	3.19	1.00	2.250	1.093	1.00	—	.50	7/16-14	1.335	—	—	—	—	1.5
1 1/4	WSAEP-10-60	2.75	3.75	1.25	2.625	1.250	1.25	.53	.50	—	1.680	1.750	1/2-13 x 2 1/4	MK 514	222	2.7
	WCSAEP-10-60	2.75	3.75	1.25	2.625	1.250	1.25	—	.50	1/2-13	1.680	—	—	—	—	2.7
1 1/2	WSAEP-12-60	3.25	4.50	1.50	3.125	1.437	1.50	.66	.50	—	1.920	2.125	5/8-11 x 2 1/2	MK 515	225	4.7
	WCSAEP-12-60	3.25	4.50	1.50	3.125	1.437	1.50	—	.50	5/8-11	1.920	—	—	—	—	4.7
2	WSAEP-16-60	4.00	5.25	1.75	3.812	1.750	1.94	.78	.62	—	2.411	2.500	3/4-10 x 3	MK 516	228	7.5
	WCSAEP-16-60	4.00	5.25	1.75	3.812	1.750	1.94	—	.62	3/4-10	2.411	—	—	—	—	7.5
2 1/2	WSAEP-20-60	5.00	6.88	2.00	4.875	2.312	2.38	.94	.62	—	2.911	3.000	7/8-9 x 3 1/2	MK 517	232	14.7
	WCSAEP-20-60	5.00	6.88	2.00	4.875	2.312	2.38	—	.62	7/8-9	2.911	—	—	—	—	14.7
3	WSAEP-24-60	6.00	8.50	2.50	6.000	2.812	2.94	1.19	.62	—	3.540	3.625	1 1/8-7 x 4 1/2	MK 718	237	28.0
	WCSAEP-24-60	6.00	8.50	2.50	6.000	2.812	2.94	—	.62	1 1/8-7	3.540	—	—	—	—	28.0
PIPE SIZE	FLANGE NO.	A	B	C	D	E	F	G	H	J	K	L	M (MOUNTING KITS[1])	NO.	'O' RING	WT.

[1] ORDER MOUNTING KITS SEPARATELY. [2] 2500 PSI [3] 2000 PSI [4] 500 PSI

Measure

1 in.	=	25.4 mm
1 in.	=	2.54 cm
1 mm	=	0.03937 in.
1 mm	=	0.00328 ft
1 micron	=	0.000001 meter
1 torr	=	1 mm mercury
10^{-3} torr	=	1 atom mercury
1 ft	=	304.8 mm
1 ft	=	30.48 cm
1 sq. in.	=	6.4516 sq cm
1 sq cm	=	0.155 sq in.
1 sq cm	=	0.00108 sq ft
1 sq ft	=	929.03 sq cm

Circumference of a circle $= 2\pi r = \pi d$

Area of a circle $= \pi r^2 = \dfrac{\pi d^2}{4}$

Weight

1 kg = 2.205 lb
1 cu in. of water (60 F) = 0.073551 cu in. of mercury (32 F)
1 cu in. of mercury (32 F) = 13.596 cu in. of water (60 F)
1 cu in. of mercury (32 F) = 0.4905 lb

Velocity

1 ft per sec = 0.3048 m per sec
1 m per sec = 3.2808 ft per sec

Density

1 lb per cu in. = 27.68 gram per cu cm
1 gr per cu cm = 0.03613 lb per cu in.
1 lb per cu ft = 16.0184 kg per cu m
1 kg per cu m = 0.06243 lb per cu ft

Physical Constants

Base of Natural Logarithms (e)		2.718 281 828 5
Acceleration of Gravity (g)	32.174 ft/sec²	(980.665 cm/sec²)
Pi (π)		3.141 592 653 6

	Degrees Kelvin	Degrees Rankine	Degrees Celsius	Degrees Fahrenheit
Absolute Zero	0	0	− 273.15	− 459.67
Water Freezing Point (14.696 psia)	273.15	491.67	0	32
Water Boiling Point (14.696 psia)	373.15	671.67	100	212

Equivalents of Temperature

To convert degrees Celsius to degrees Fahrenheit:

$$t = 1.8\, t_c + 32$$

To convert degrees Fahrenheit to degrees Celsius:

$$t_c = \frac{t - 32}{1.8}$$

Where: t_c = temperature, in degrees Celsius

Prefixes

atto	a	one-quintillionth	0.000 000 000 000 000 001	10^{-18}
femto	f	one-quadrillionth	0.000 000 000 000 001	10^{-15}
pico	p	one-trillionth	0.000 000 000 001	10^{-12}
nano	n	one-billionth	0.000 000 001	10^{-9}
micro	μ	one-millionth	0.000 001	10^{-6}
milli	m	one-thousandth	0.001	10^{-3}
centi	c	one-hundredth	0.01	10^{-2}
deci	d	one-tenth	0.1	10^{-1}
uni		one	1.0	10^{0}
deka	da	ten	10.0	10^{1}
hecto	h	one hundred	100.0	10^{2}
kilo	k	one thousand	1 000.0	10^{3}
mega	M	one million	1 000 000.0	10^{6}
giga	G	one billion	1 000 000 000.0	10^{9}
tera	T	one trillion	1 000 000 000 000.0	10^{12}

Equivalents of Liquid Measures and Weights

* TO OBTAIN → MULTIPLY BY ↓	U.S. Gallon	Imperial Gallon	U.S. Pint	U.S. Pound Water*	U.S. Cubic Foot	U.S. Cubic Inch	Liter	Cubic Meter
U.S. Gallon	1	0.833	8.	8.337	0.13368	231.	3.78533	0.003785
Imperial Gallon	1.2009	1	9.60752	10.	0.16054	277.42	4.54596	0.004546
U.S. Pint	0.125	0.1041	1	1.042	0.01671	28.875	0.473166	0.000473
U.S. Pound Water*	0.11995	0.1	0.9596	1	0.016035	27.708	0.45405	0.000454
U.S. Cubic Foot	7.48052	6.22888	59.8442	62.365	1	1728.	28.31702	0.028317
U.S. Cubic Inch	0.004329	0.00361	0.034632	0.03609	0.0005787	1	0.016387	0.0000164
Liter	0.2641779	0.2199756	2.113423	2.202	0.0353154	61.02509	1	0.001000
Cubic Meter	264.170	219.969	2113.34	2202.	35.31446	61023.38	999.972	1

*Water at 60 F (15.6C)   1 Barrel = 42 gallons (petroleum measure)

Equivalents of Pressure and Head

TO OBTAIN → MULTIPLY BY ↓	lb/in²	lb/ft²	Atmospheres	kg/cm²	kg/m²	in. water (68 F)*	ft. water (68 F)*	in. mercury (32 F)†	mm mercury (32 F)†	Bars ‡	MegaPascals (MPa)‡
lb/in²	1	144.	0.068046	0.070307	703.070	27.7276	2.3106	2.03602	51.7150	0.06895	0.006895
lb/ft²	0.0069445	1	0.000473	0.000488	4.88241	0.1926	0.01605	0.014139	0.35913	0.000479	0.0000479
Atmospheres	14.696	2116.22	1	1.0332	10332.27	407.484	33.9570	29.921	760.	1.01325	0.101325
kg/cm²	14.2233	2048.155	0.96784	1	10000.	394.38	32.8650	28.959	735.559	0.98067	0.098067
kg/m²	0.001422	0.204768	0.0000968	0.0001	1	0.03944	0.003287	0.002896	0.073556	0.000098	0.0000098
in. water*	0.036092	5.1972	0.002454	0.00253	25.375	1	0.08333	0.073430	1.8651	0.00249	0.000249
ft. water*	0.432781	62.3205	0.029449	0.03043	304.275	12.	1	0.88115	22.3813	0.029839	0.0029839
in. mercury†	0.491154	70.7262	0.033421	0.03453	345.316	13.6185	1.1349	1	25.40005	0.033864	0.0033864
mm mercury†	0.0193368	2.78450	0.0013158	0.0013595	13.59509	0.53616	0.044680	0.03937	1	0.001333	0.0001333
Bars‡	14.5038	2088.55	0.98692	1.01972	10197.2	402.156	33.5130	29.5300	750.062	1	0.10
MPa‡	145.038	20885.5	9.8692	10.1972	101972.0	4021.56	335.130	295.300	7500.62	10.0	1

*Water at 68 F (20C)   †mercury at 32 F (0C)   ‡1 MPa (MegaPascal) = 10 Bars = 1,000,000 N/m² (Newtons/meter²)

FLUID POWER FORMULAS

FORMULA FOR:	WORD FORMULA:		LETTER FORMULA:
FLUID PRESSURE *In Pounds/Square Inch*	PRESSURE	$= \frac{\text{FORCE (pounds)}}{\text{UNIT AREA (Square Inches)}}$	$P = \frac{F}{A}$ or $psi = \frac{F}{A}$
CYLINDER AREA *In Square Inches*	AREA	$= \pi \times \text{RADIUS}^2 \text{ (Inches)}$	$A = \pi r^2$
		$= \frac{\pi}{4} \times \text{Diameter}^2 \text{ (Inches)}$	$A = \frac{\pi D^2}{4}$ or $A = .785D^2$
CYLINDER FORCE *In Pounds, Push or Pull*	FORCE	$= \text{PRESSURE (psi)} \times \text{NET AREA (Square Inches)}$	$F = psi \times A$ or $F = PA$
CYLINDER VELOCITY or SPEED *In Feet/Second*	VELOCITY	$= \frac{231 \times \text{FLOW RATE (GPM)}}{12 \times 60 \times \text{NET AREA (Square Inches)}}$	$v = \frac{231Q}{720A}$ or $v = \frac{.3208Q}{A}$
CYLINDER VOLUME CAPACITY *In Gallons of Fluid*	VOLUME	$= \frac{\pi \times \text{RADIUS}^2 \text{ (Inches)} \times \text{STROKE (Inches)}}{231}$	$V = \frac{\pi r^2 \ell}{231}$
		$= \frac{\text{NET AREA (Square Inches)} \times \text{STROKE (Inches)}}{231}$	$V = \frac{A \ell}{231}$ ℓ = *Length of Stroke*
CYLINDER FLOW RATE *In Gallons Per Minute*	FLOW RATE	$= \frac{12 \times 60 \times \text{VELOCITY (Feet/Sec)} \times \text{NET AREA (Square Inches)}}{231}$	$Q = \frac{720vA}{231}$ or $Q = 3.117vA$
FLUID MOTOR TORQUE *In Inch Pounds*	TORQUE	$= \frac{\text{PRESSURE (psi)} \times \text{F.M. DISPLACEMENT (Cu. In./Rev.)}}{2\pi}$	$T = \frac{psi\ d}{2\pi}$ or $T = \frac{Pd}{2\pi}$
		$= \frac{\text{HORSEPOWER} \times 63025}{\text{RPM}}$	$T = \frac{63025\ HP}{n}$
		$= \frac{\text{FLOW RATE (GPM)} \times \text{PRESSURE (psi)} \times 36.77}{\text{RPM}}$	$T = \frac{36.77QP}{n}$ or $T = \frac{36.77Qpsi}{n}$
FLUID MOTOR TORQUE / 100 psi *In Inch Pounds*	TORQUE/100 psi	$= \frac{\text{F.M. DISPLACEMENT (Cu. Inches/Revolution)}}{.0628}$	$T_{100psi} = \frac{d}{.0628}$
FLUID MOTOR SPEED *In Revolutions/Minute*	SPEED	$= \frac{231 \times \text{FLOW RATE (GPM)}}{\text{F.M. DISPLACEMENT (Cu. Inches/Revolution)}}$	$n = \frac{231Q}{d}$
FLUID MOTOR POWER *In Horsepower Output*	HORSEPOWER	$= \frac{\text{TORQUE OUTPUT (Inch Pounds)} \times \text{RPM}}{63025}$	$HP = \frac{Tn}{63025}$
PUMP OUTLET FLOW *In Gallons/Minute*	FLOW	$= \frac{\text{RPM} \times \text{PUMP DISPLACEMENT (Cu. In./Rev.)}}{231}$	$Q = \frac{nd}{231}$
PUMP INPUT POWER *In Horsepower Required*	HORSEPOWER INPUT	$= \frac{\text{FLOW RATE OUTPUT (GPM)} \times \text{PRESSURE (psi)}}{1714 \times \text{EFFICIENCY (Overall)}}$	$HP_{IN} = \frac{QP}{1714Eff}$ or $\frac{GPM \times psi}{1714Eff}$
FLOW RATE THROUGH PIPING *In Feet/Second Velocity*	VELOCITY	$= \frac{.3208 \times \text{FLOW RATE THROUGH I.D. (GPM)}}{\text{INTERNAL AREA (Square Inches)}}$	$v = \frac{.3208Q}{A}$
COMPRESSIBILITY OF OIL *In Additional Required Oil To Reach Pressure*	ADDITIONAL VOLUME	$= \frac{\text{PRESSURE (psi)} \times \text{VOLUME OF OIL UNDER PRESSURE}}{250{,}000}$	$V_A = \frac{PV}{250{,}000}$ [Approximately ½% Per 1000 psi]

Selected "SI" Units For General Purpose Fluid Power Usage

Quantity	SI Unit for Fluid Power	"Customary US" Unit for Fluid Power	Conversion
length	millimeter (mm)	inch (in.)	1 in. = 25.4 mm
pressure[1]	bar (assumed to be "gage" unless otherwise stated)	pounds per square inch (psig or psia)	1 bar = 14.5 psi
pressure[2]	bar (a value less than 1.0, for example 0.95 bar)	inches of mercury (in. Hg)	1 in Hg (at 60 F) = 0.034 bar
flow[3]	liters per minute (l/min)	gallons per minute (USgpm)	1 USgpm = 3.79 l/min
flow[4]	cubic decimeters per second (dm^3/s)	cubic feet per minute (cfm)	1 dm^3/s = 2.12 cfm
force	newton (N)	pound(f) lb(f)	1 lb(f) = 4.44 N
mass	kilogram (kg)	pound(m) lb(m)	1 kg = 2.20 lb(m)
time	second (s)	second (s)	...
volume[3]	liter (l)	gallon (US gal)	1 US gal = 3.79 l
temperature	degrees Celsius (C)	degrees Fahrenheit (F)	C = 5/9 (F −32)
torque	newton-meters (N•m)	pounds(f)-inches lb(f)-in.	1 N•m = 8.88 lb(f)-in.
power	kilowatt (kw)	horsepower (hp)	1 kw = 1.34 hp
shaft speed	revolutions per minute (rev/min)	revolutions per minute (rpm)	...

frequency	hertz (Hz)	cycles per second (cps)	1 Hz = 1 cps
displacement[3]	milliliters per revolution (ml/rev)	cubic inches per revolution (cipr)	1 ml/rev = 0.061 cipr
kinematic viscosity	centistokes (cSt)	saybolt universal seconds (SUS)	cSt = (4.635) SUS[5]
velocity	meters per second (m/s)	feet per second (fps)	1 m/s = 3.28 fps
material stress	decanewtons per square millimeter (daN/mm^2)	pounds per square inch (psi)	1 daN/mm^2 = 1,450 psi
surface roughness	micro-meter (μm)	micro-inch ($\mu in.$)	1 $\mu in.$ = 0.025 μm

1. pressure above atmospheric
2. pressures below atmospheric
3. liquid
4. gas under standard temperature, humidity, and pressure conditions
5. at 38 C; factor is 4.667 at 99 C

TESTING AND STANDARDS

QUANTITIES					Conversion				
Name	Symbol	Typical Applications	Customary U.S. Units	Metric Units	U.S. Unit	Multiply By	To Get Metric	Multiply By	To Get U.S.
Stress, Normal	σ	normal stress, shear stress, material strength	pound per square inch	megapascal	lb/in^2 (psi)	0.006895	MPa	145.0	lb/in^2 (psi)
Shear	τ								
Temperature, Customary	θ	thermal operating limits dryer dewpoint	degree Fahrenheit	degree Celsius	°F	$t_{°C} = \frac{(t_{°F} - 32)}{1.8}$*	°C	$t_{°F} = 1.8(°C)+32$*	°F
Temperature Thermodynamic (absolute)	T	pneumatic system calculations	degree Rankine	kelvin	°R	1/1.8*	K	1.8*	°R
Time	t	component response, system cycle, maintenance cycle	second	second	sec	1.0*	s	1.0*	sec
			minute	minute	min	1.0*	min	1.0*	min
			hour	hour	hr	1.0*	h	1.0*	hr
Torque	T	motor, rotary actuator	pound foot	newton metre	lb·ft	1.356	N·m	0.7376	lb·ft
		output, bolt tightening	pound inch	newton metre	lb·in	0.1130	N·m	8.851	lb·in
Velocity, linear	v	fluid flow	foot per second	metre per second	ft/sec	0.3048*	m/s	3.281	ft/sec
		cylinder actuator	inch per second	millimetre per second	in/sec	25.4*	mm/s	0.03937	in/sec
Viscosity, Dynamic	η	system fluid properties	centipoise	millipascal second	cP	1.0*	mPa·s	1.0*	cP
Viscosity, Kinematic	ν	system fluid properties	centistokes (Note 7)	square millimetre per second	cSt	1.0*	mm^2/s	1.0*	cSt
Volume	V	lubricators, accumulators, reservoir capacity — pneumatics	cubic foot	cubic decimetre	ft^3	28.32	dm^3	0.0353	ft^3
		pneumatics	cubic inch	cubic centimetre	in^3	16.39	cm^3	0.06102	in^3
		hydraulics	gallon	litre	gal	3.785	L (Note 2)	0.2642	gal
		hydraulics	ounce	millilitre	oz	29.57	mL (Note 2)	0.03381	oz
Work	W	cylinders, linear actuators	foot pound	joule	ft·lb	1.356	J	0.7376	ft·lb

Notes: **An asterisk indicates the conversion factor is exact and all subsequent digits are zero. All other conversion factors have been rounded to 4 significant digits.* **2.** *The international symbol for litre is the lower case "l" which can be easily confused with the numeral "1". Accordingly the symbol "L" is recommended for US fluid power use.* **7.** *Viscosity is frequently expressed in SUS (Saybolt Universal Seconds). SUS is the time in seconds for 60 mL of fluid to flow through a standard orifice at a specified temperature. Conversion between kinematic viscosity, mm^2/s (centistokes) and SUS can be made*

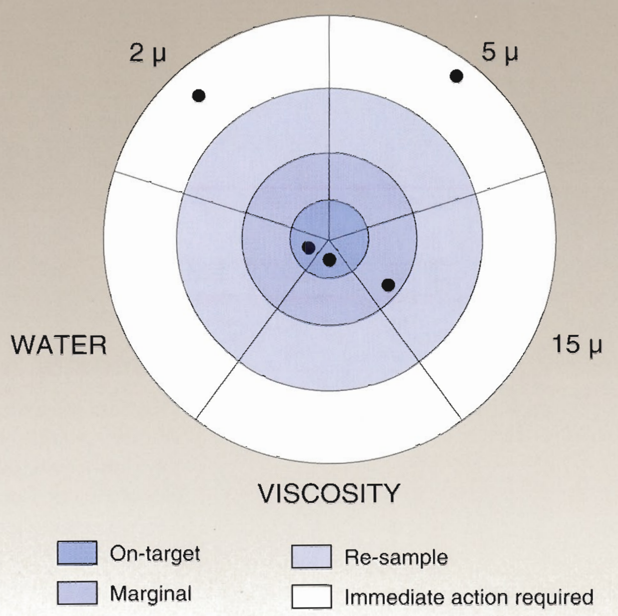
Fluid Results

The Vickers Fluid Analysis Service presents your fluid results in an easy to understand format. The fluid results target is a quick way to visualize the condition of your sample.

A comparison is made between your actual fluid cleanliness results and your target cleanliness level. If you do not know what your target should be, Vickers will analyze your system and help you set a target based on the information you provide with the sample.

The recommendation and comment section of the report provides you with valuable information on the cleanliness of your hydraulic system, as well as tips on maintaining or improving the current condition.

Fluid Results



Trend Information

The Vickers Fluid Analysis Service provides data from your previous two samples along with the current samples' results, to provide you with a trend analysis of the critical measurements of your system's condition. This trend analysis provides quantitative measurement as to the condition of the fluid over time and the status of corrective actions taken.

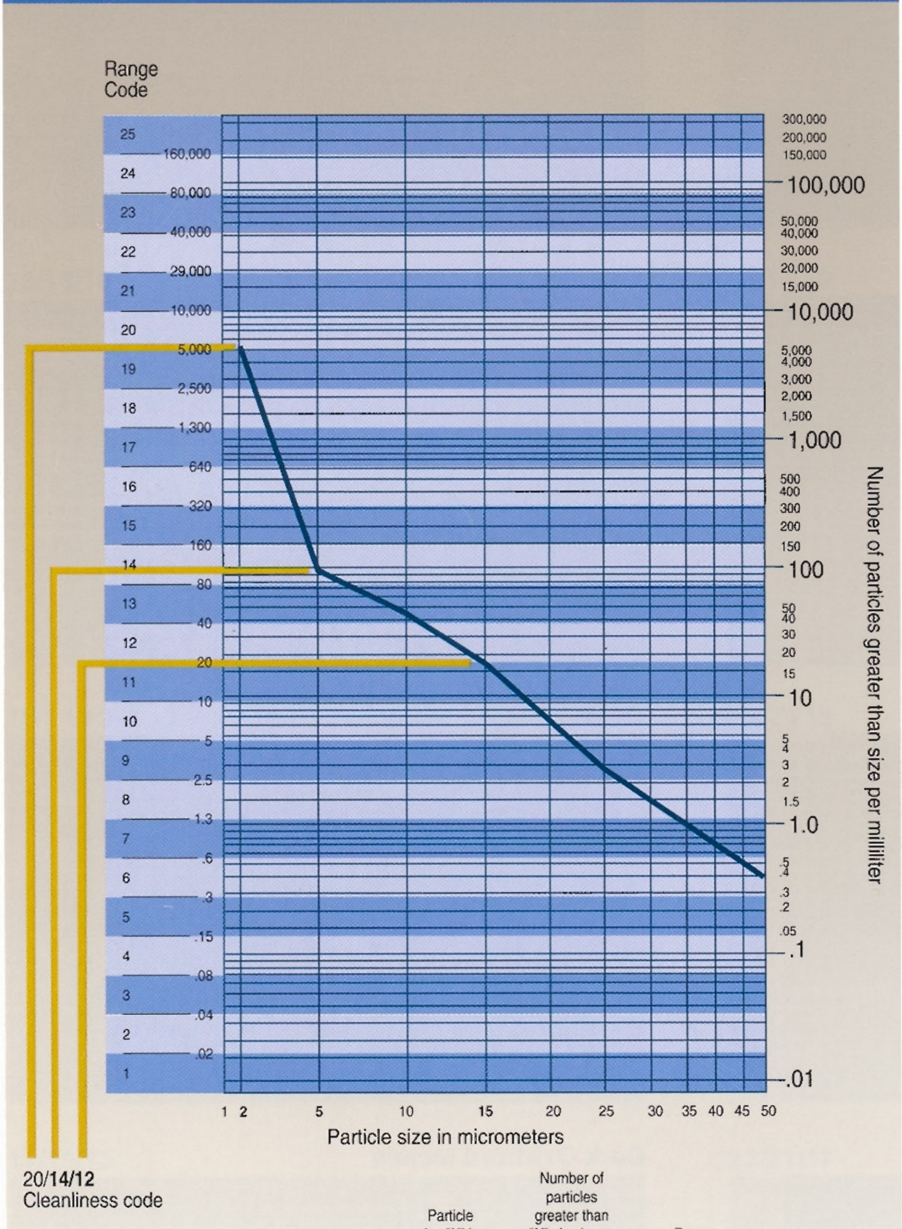
	PREVIOUS	PREVIOUS	CURRENT
DATE	6/22/92	7/25/92	8/31/92
VISCOSITY @ 100°F cSt (SUS)	45.0 (210)	45.5 (212)	45.8 (213)
WATER % WEIGHT	0.03%	0.03%	0.03%
pH NOTE: pH is for water containing fluids only.	9.4	9.5	9.6
TAN mg KOH/gm NOTE: TAN is for synthetic fluids only.	2.1	2.0	2.1

PARTICLE COUNT SUMMARY			
	PREVIOUS	PREVIOUS	CURRENT
DATE	6/22/92	7/25/92	8/31/92
> 2 μ	65,120	4,100	418
> 5 μ	12,220	1,250	88
> 10 μ	5,800	700	39
> 15 μ	900	250	22
> 25 μ	125	60	4
> 50 μ	12.0	5.0	1.0
CLEANLINESS CODE	23/21/17	19/17/15	16/14/12

Cleanliness Code Graph

The cleanliness code graph uses the industry standard for measuring and depicting the amount and size of particles per milliliter in hydraulic fluid. The ISO 4406 standard requires a log-log[2] graph that charts the amount of particles greater than certain micron sizes per milliliter of fluid. The 5µm and 15 µm measurements are converted to a range code and then used together and referred to as the ISO code for a particular sample *(example 14/12)*. The Vickers Cleanliness Code is similar to the ISO code, but it includes the 2 µm measurement as well *(example 20/14/12)*. The 2µm measurement enables the Vickers Fluid Analysis Service to measure the efficiency of finer filtration on today's higher pressure systems with closer tolerances.

Cleanliness Code Graph



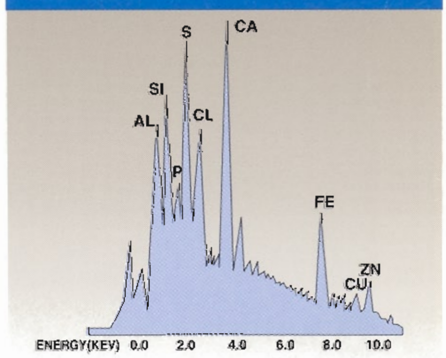
Example

Particle size "X" in micrometers	Number of particles greater than "X" size in one ml of test fluid	Range code
2	5120	20
5	89	14
10	43	X
15	22	12
25	3	X
50	.4	X

Scanning Electron Microscope (SEM)/ Energy Dispersive X-ray Analysis (EDX)

The SEM/EDX analysis is automatically performed on samples submitted with extremely high concentrations of particulate contamination (range codes over "22"). The SEM/EDX analysis detects chemical elements with atomic numbers greater than 10 (including most metals, chlorine and sulfur) but does not detect the lighter chemical elements typical of organic materials, such as carbon and oxygen. The elemental composition results are shown in a spectrum (See chart below). The spectrum shows the relative amounts of each element present in particle form in the hydraulic fluid. This elemental chemical analysis helps establish the origin of the contaminants so corrective action may be taken.

SEM/EDX



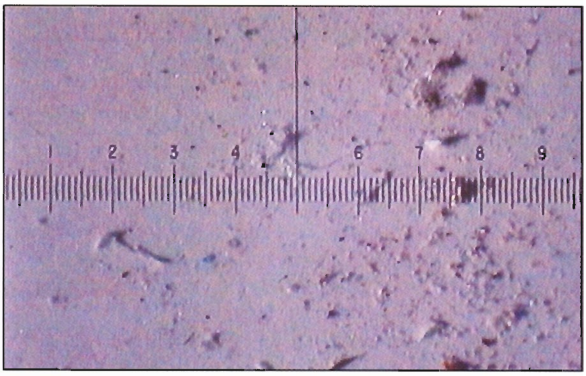
Spectrometric Analysis

This test method provides a means of determining the concentration of oil-soluble elements. Spectrometric Analysis gives an indication of the additives and trace metal content in the fluid, but this method is *particle size dependent*. The inability to detect large wear particles has led to hydraulic component failures without prior indication by spectrometric method. *This technique should be used to evaluate the condition of the additives in a fluid rather than the particulate contamination.*

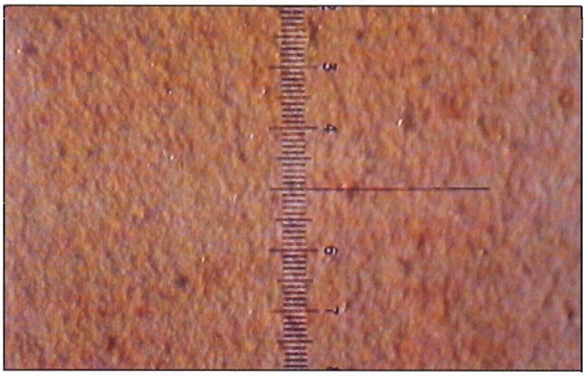
Typical Photomicrographs Showing Various *Types* of Contamination

Photos show typical results obtained by the drawdown particle isolation procedure used by the Vickers Fluid Analysis Laboratory.
(Photomicrographs taken at 100x)

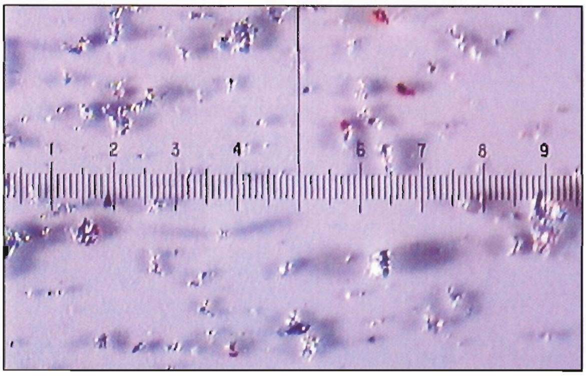
Silica



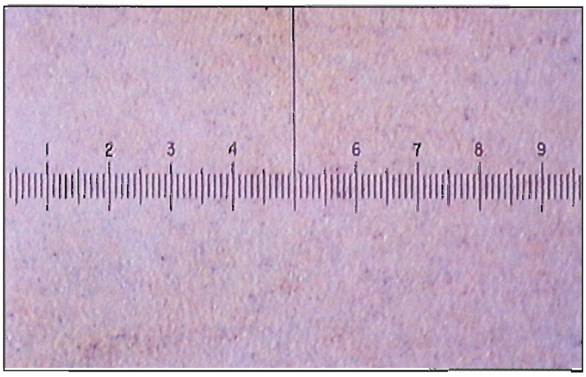
Additives



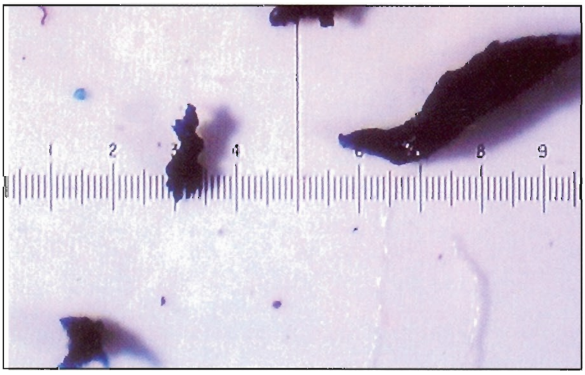
Bright Metals



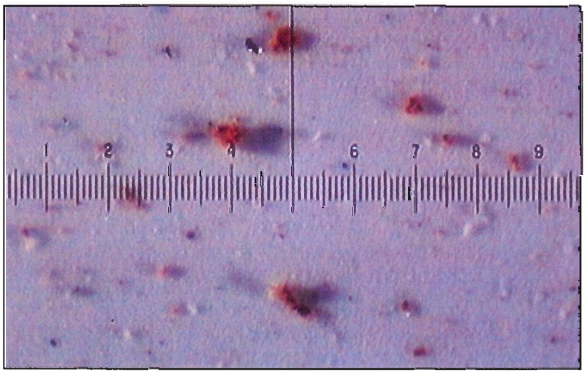
Silt



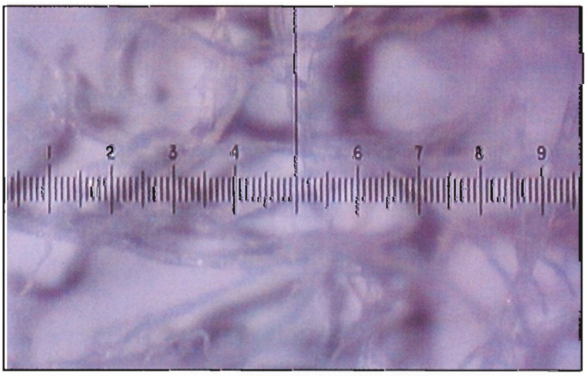
Elastomer



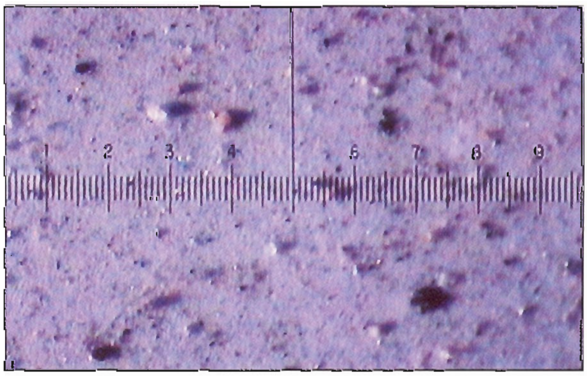
Rust



Fibers



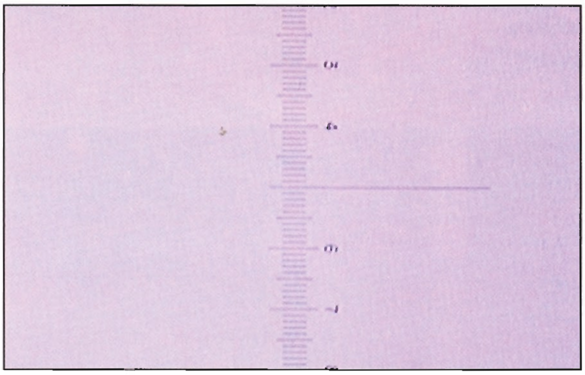
Dark Oxidized Metals



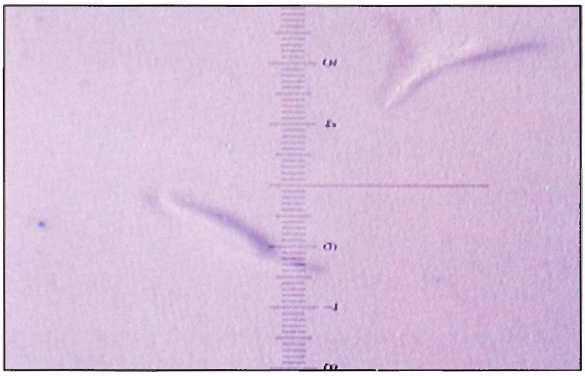
Typical Photomicrographs Showing Various *Degrees* of Contamination

(Photomicrographs taken at 100x)

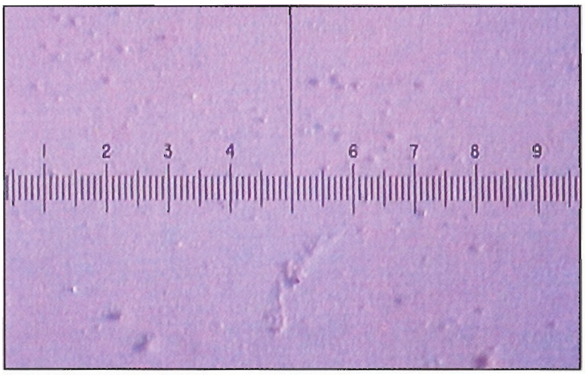
Blank Patch (no contamination)



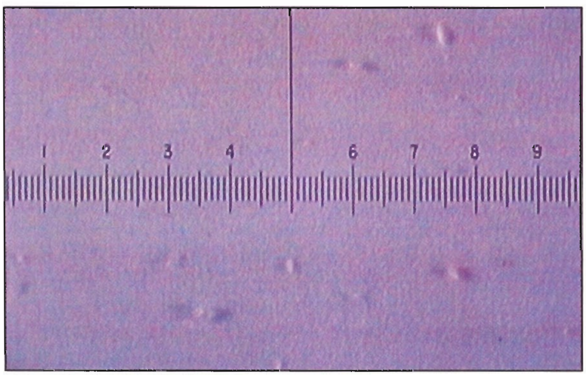
15/13/10



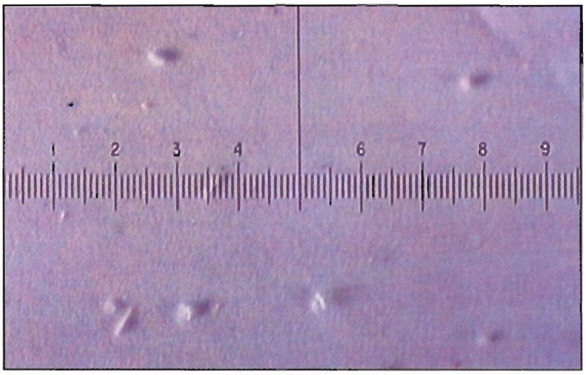
16/14/12



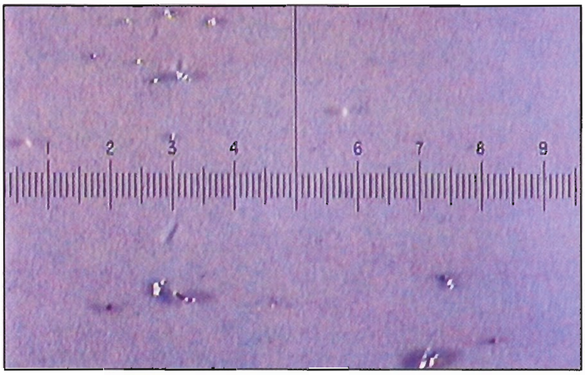
17/15/13



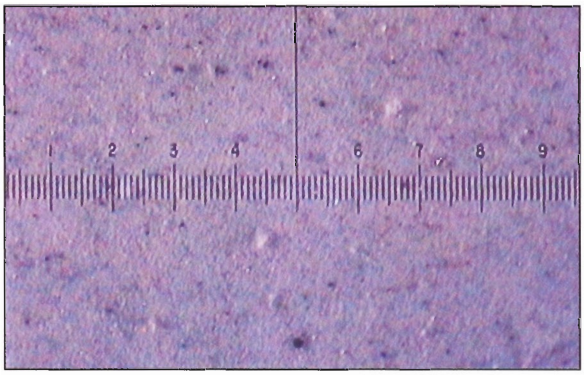
18/16/14



19/17/15



21/19/15



23/21/17