BRUCE WILNECHENKO

Fluid Power

DATA BOOK

☆ CHARTS
☆ CONVERSIONS
☆ CIRCUITS
☆ ASA SYMBOLS
☆ TROUBLE SHOOTING
☆ DESIGN DATA

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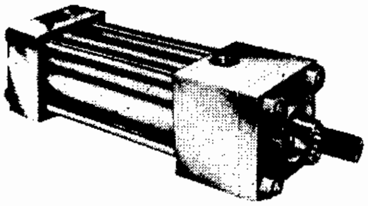
Fluid Power Data Book

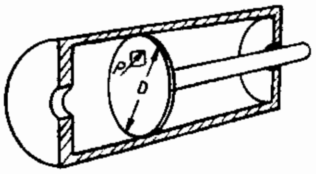
A collection of useful fluid power data. Published in this condensed form for convenient reference. For expanded educational material on fluid power.

CYLINDERS

A cylinder is used in the majority of applications to convert fluid power into push-pull motion.
The illustration shows one brand of popular cylinder. These are available in a wide range of bore diameters, mounting styles, and stroke lengths.



OPERATING PRINCIPLES



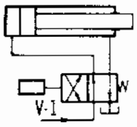
The operating principle of a cylinder is very simple: fluid pressure is applied to one side of the piston, and the opposite side of the piston is exhausted, usually to atmospheric pressure. Thrust developed on the piston rod is easily calculated by multiplying gauge pressure times the piston area. Piston area is calculated from the formula:

$$\text{Area} = \frac{\pi D^2}{4}$$

EXAMPLE: Find the thrust developed by an air cylinder of 4" piston diameter operating on a line pressure of 90 PSI (lbs. per square inch):

Solving the above formula, the piston area is 12.566 square inches. Since the air pressure exerts a pressure, P, of 90 lbs. on each square, then the total thrust will be 90 x 12.566 or 1131 lbs.

Remember, when calculating force developed on the retraction stroke, the pressure does not act on the area covered by the piston rod; therefore, the rod area must be subtracted from the total piston area.
To save mathematical calculations, we have published tables on the following pages to show force developed by cylinders of various diameters operating at various fluid pressures. Tables will also be found for calculating the speed of hydraulic cylinders and estimating the speed of air cylinders.

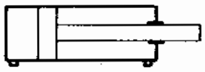


DIRECTION OF TRAVEL of a double-acting air or hydraulic cylinder is usually controlled with a 4-way valve as shown in this illustration. The valve may be two-position as shown, or may have a center, neutral, position also.
Direction of travel of a single-acting (spring or gravity returned) cylinder may be controlled with a 3-way valve, although a 4-way valve is frequently employed for hydraulic single action cylinders to unload the pump while the cylinder is retracting.

TRAVEL SPEED of air cylinders is usually controlled with a "flow control" valve which is a needle valve by-passed with a check valve in the same envelope. This gives controlled speed in one direction and free flow in the reverse direction. These valves may also be used for hydraulic cylinders, or the pressure compensated flow control valve may be used for accurate control under varying load conditions.

CUSHIONS. Both air and hydraulic cylinders may be ordered with cushions on rod end, blind end, or both ends. The cushion consists of a closed chamber approximately 1 inch from the end of the stroke for trapping the operating fluid when the piston reaches the cushion entrance. From this point the fluid is metered out slowly in order to slow the cylinder movement. On most cylinders the metering rate is controlled with a built-in adjustable needle valve, by-passed with a check valve, for quick start-up in the opposite direction.
Cushions are quite effective on hydraulic cylinders, but effective on air cylinders for cushioning fast speeds only when the momentum load is very small. For cushioning high momentum loads on air cylinders, use limit switch or cam valve deceleration.

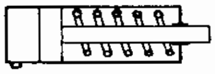
CYLINDER TYPES



STANDARD DOUBLE-ACTING. Provides a power stroke in both directions. This is the standard type used for the majority of applications.

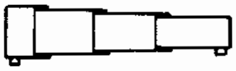
SINGLE-ACTING CYLINDER. Where thrust is needed in only one direction, a double-acting cylinder may be used, with the inactive end vented to atmosphere through a breather/filter in the case of an air cylinder, or vented to reservoir, below the oil level in the case of a hydraulic cylinder.

DOUBLE ROD CYLINDERS are available in most brands, and are used where equal displacement is needed on both sides of the cylinder piston, or where it is mechanically advantageous to couple a load to each end. Sometimes the extra end is used solely to mount cams for operating limit switches, etc.



SPRING RETURN, SINGLE-ACTING cylinders are usually limited to very small, short-stroke cylinders used mainly for holding and clamping. The long length required to contain the return spring makes their use undesirable where a long stroke is needed.

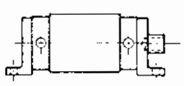
RAM-TYPE SINGLE-ACTING CYLINDERS have only one fluid chamber. Most commonly they are mounted vertically and are retracted by the load weight. They are practical for long strokes. They are sometimes known as "displacement cylinders." and are used for hydraulic house jacks and filling station lifts.



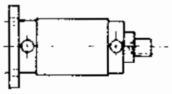
TELESCOPIC CYLINDERS are used where collapsed length must be shorter than could be obtained with a standard cylinder. These can be obtained with up to 4 or 5 sleeves, either single-acting or double-acting type. They are relatively expensive as compared to standard cylinders.

MOUNTING STYLES

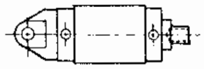
These sketches show the most popular cylinder body mounts. Most brands are available with either male or female threaded end on piston rod, or with clevis or tang screwed or welded to end of rod.



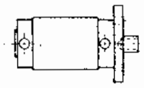
FOOT BRACKETS, side lugs, or centerline lugs, etc.



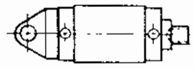
FLANGE attached to blind end of cylinder.



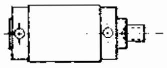
CLEVIS attached to blind end of cylinder.



FLANGE attached to rod end of cylinder.



TANG attached to blind end of cylinder.

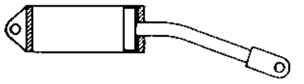


DRILLED HOLES in end caps or body.

Designing With Cylinders

Standard cylinders are never designed to take any side loading on the piston rod. They must be carefully and accurately mounted so the rod is not placed in a bind at any part of the stroke. In many cases the cylinder must have a clevis or trunnion mount to allow it to swing as the direction of the load changes. Use guides on the load mechanism, if necessary, to assure that no side load is transmitted to the cylinder rod.

ROD BUCKLING



Column failure, or the buckling of the rod, may occur if the cylinder stroke is too long in relation to the rod diameter. The exact ratio of rod length to rod diameter at which column failure will occur cannot be accurately calculated, but the "Minimum Piston Rod Diameter" chart in this manual will give the minimum safe ratio for normal applications.

TENSION AND COMPRESSION FAILURES

All standard cylinders have been designed with sufficiently large piston rods so they will never fail either in compression or tension, if the cylinder is operated within the pressure rating of the manufacturer.

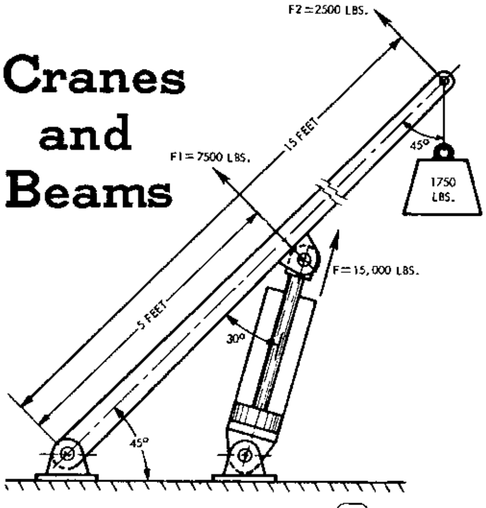
ROD BEARING FAILURE

Rod bearing failures usually occur when the cylinder is at maximum extension, and occur most often on hinge or trunnion mount cylinders, in which the rear support point is located considerably behind the rod bearing. Where space permits, it is wise to order cylinders with longer stroke than actually needed, and not permit the piston to approach close to the front end under load.

STOP COLLAR

On those applications where it is necessary to allow the piston to "bottom out" on the front end, the cylinder may be ordered with a stop collar. The stop collar should especially be considered on long strokes if the length between supports exceeds 10 times the rod diameter, if the maximum thrust is required at full extension, and if the cylinder has rear flange, trunnion, rear clevis or tang mounting.

Cranes and Beams



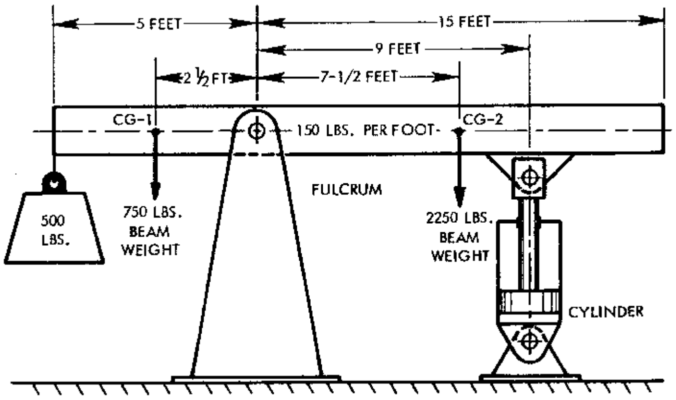
Since the working angles on a crane are constantly changing, it may be necessary to construct a rough model on a sheet of paper, to exact scale, with cardboard arms and thumbtack hinge pins. This will show the point at which the greatest cylinder thrust is needed. An exact calculation can then be made for this condition.

Only that part of the cylinder thrust at right angles to the beam axis is effective for turning the beam. This can be calculated by the method shown elsewhere in this manual. For heavy beams the beam weight will have to be entered into the calculations.

EXAMPLE: (Crane calculation using the figure on the preceding page.) Starting with a cylinder thrust, F, of 15,000 lbs., find the maximum load that can be lifted by the crane when the angles are as shown. First, translate the 15,000 lbs. cylinder thrust into F1, 7500 lbs. at right angles to the beam, using power factor of 0.500 from the table on the next page, for a 30° angle. Next, translate this to F2, 2500 lbs. thrust at the end of the beam where the weight is hanging. This is done with simple proportion by the length of each arm from the base pivot point. F2 is ⅓rd F1 since the lever arm is 3 times as long. Next, find the maximum hanging load that can be lifted, at a 45° angle between beam and load weight, using the power factor table on the next page.

2500 x 0.707 = 1750 lbs.

Calculations for a heavy beam



On heavy beams it is necessary not only to calculate for concentrated loads such as suspended weights and cylinder thrusts, but to take into account the distributed weight of the beam itself. If the beam is uniform, (so many) pounds per foot length, the calculation is relatively easy. In the above example the beam has a uniform weight of 150 pounds per foot, is partially counterbalanced by a load weight of 500 pounds on the left side of the fulcrum, and must be raised by the force of a cylinder applied at a point 9 feet from the right side of the fulcrum.

The best method of solution is to use the principle of moments. A moment is a torque force consisting of (so many) pounds applied at a lever distance of (so many) feet or inches. The solution here is to find how much cylinder thrust is needed to just balance the beam. Then by increasing the hydraulic cylinder thrust about 5 to 10% to take care of friction losses, the cylinder would be able to raise the beam.

Using the principle of moments, it is necessary to calculate all of the moment forces which are trying to turn the beam clockwise, then calculate all the moment forces trying to turn the beam counter-clockwise, then subtract the two. In this case they must be equal to balance the beam.

Clockwise moment due to the 15 feet of beam on the right side of the fulcrum: This can be considered as a concentrated weight acting at its center of gravity 7½ feet from the fulcrum. Moment = 150 (lbs. per foot) × 15 feet × 7½ feet = 16,875 foot pounds.

Counter-clockwise moment due to the 5 feet of beam on the left side of the fulcrum: 150 (lbs. per foot) × 5 feet × 2½ feet (CG distance) = 1875 foot pounds.

Counter-clockwise moment due to hanging weight of 500 pounds: 500 lbs. × 5 feet = 2500 foot pounds.

Subtracting counter-clockwise from clockwise moments: 16,875 – 1875 – 2500 = 12,500 foot pounds that must be supplied by the cylinder for balance conditions. To find cylinder thrust: 12,500 foot pounds ÷ 9 feet (distance from fulcrum) = 1388.8 pounds.

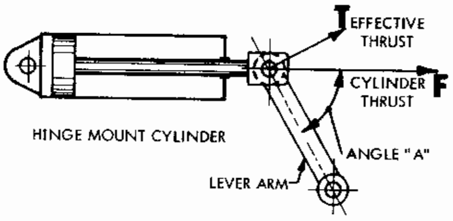
Remember, when working with moments, that only the portion of the total force which is at right angles to the beam is effective as a moment force. If the beam is at an angle to the cylinder or to the horizontal, then the effective portion of the concentrated or distributed weight, and the cylinder thrust, can be calculated with power factors by the method on the next page.

Cylinder Working at an Angle

To find the effective force exerted by a cylinder pushing at an angle to the machine travel.

Cylinder thrust, F, is horizontal in this figure. Only that portion, T, which is at right angles to the lever axis is effective for turning the lever. The value of T varies with the acute angle "A" between cylinder and lever axes.

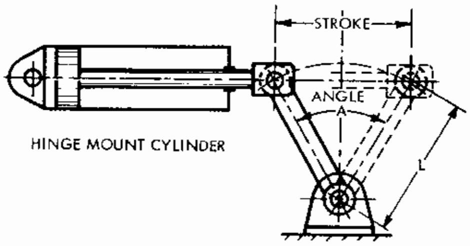
To calculate T, multiply cylinder thrust times the power factor taken from the table below.



Angle A Degrees	Pwr. Factor (sin A)	Angle A Degrees	Pwr. Factor (sin A)	Angle A Degrees	Pwr. Factor (sin A)
5	0.087	35	0.573	65	0.906
10	0.174	40	0.643	70	0.940
15	0.259	45	0.707	75	0.966
20	0.342	50	0.766	80	0.985
25	0.423	55	0.819	85	0.966
30	0.500	60	0.867	90	1.000

EXAMPLE: A 4-inch bore cylinder working at 750 PSI gauge pressure will develop a 9425 lb. thrust (12.5664 sq. in. area x 750). Effective thrust when working at a 65° angle is: 9425 x 0.906 (from above table) = 8539 lbs.

To find the cylinder stroke for operating a hinged lever, using the chord factor method.



If the cylinder is rotating the lever to an equal angle each side of the perpendicular as in this figure, the length of stroke can very easily be determined by multiplying lever length (pin-to-pin) times the chord factor from the table below. If the movement is not equal on each side of the perpendicular, the stroke may be determined by the method on the next page.

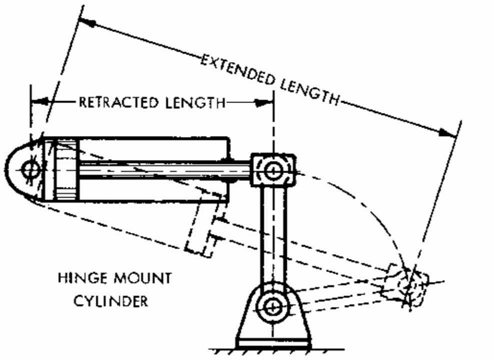
A cylinder which operates any hinged device must be free to swing with the motion. It may have clevis or tang mounting on the rear end, or have a trunnion mount. Its rod must have a tang or clevis end with a throat deep enough so the lever will not touch the bottom of the slot on extreme angular movements.

Angle A Degrees	Chord Factor	Angle A Degrees	Chord Factor	Angle A Degrees	Chord Factor	Angle A Degrees	Chord Factor
5	0.087	45	0.765	85	1.351	125	1.774
10	0.174	50	0.845	90	1.414	130	1.813
15	0.261	55	0.923	95	1.475	135	1.848
20	0.347	60	1.000	100	1.532	140	1.879
25	0.433	65	1.075	105	1.587	145	1.907
30	0.518	70	1.147	110	1.638	150	1.932
35	0.601	75	1.217	115	1.687	155	1.953
40	0.684	80	1.286	120	1.732	160	1.970

EXAMPLE: The cylinder stroke needed to swing a 14-inch lever through a 105 degree arc, when mounted as in the figure, is found by taking the factor 1.587 from the chart times 14″ lever length = 22.218″ stroke length.

Many times a stock cylinder with standardized stroke length can be used by lengthening or shortening the lever arm for the desired travel.

To find the cylinder stroke for operating a hinged lever, using the scale layout method.



In all cases a sketch should be made, showing the length and angular travel of the lever, and showing the mounting position of the cylinder.

If desired, an exact solution can be worked out by mathematics.

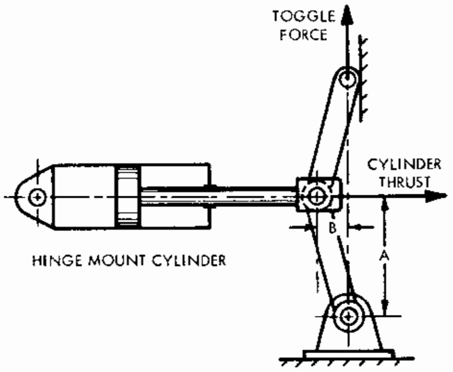
For those not familiar with mathematical methods, an easy solution is to lay out all parts to exact scale, either to full or reduced size. Pin-to-pin centers on the proposed cylinder can be obtained from the manufacturer's drawings.

A ruler, tape, or scale can be used to measure the distance from the cylinder rear hinge to the starting and ending points of the lever travel. These will be the retracted and extended cylinder lengths. The travel of the cylinder piston (or stroke) will be the difference between these two measurements.

It may be necessary to experiment with different hinge point locations until the best mounting position for the cylinder can be determined.

As a matter of interest, for a given amount of angular travel, the longest cylinder stroke is required when the cylinder is mounted at right angles to the lever center position as on the preceding page. All other cylinder mounting locations will need a shorter stroke.

Toggle Mechanism Operated With a Cylinder



For operations such as coining and marking requiring exact depth control, and requiring extremely high force for a very short distance, the toggle lever system is useful.

In this figure, cylinder thrust is horizontal and toggle force is taken off vertically. Bearings at each end of the toggle lever must be closely fitted and heavy enough to carry the full toggle thrust.

A calculation of toggle force can be made with the following formula, with T and F in the same units, and A and B in the same units. Note that dimension A is not the lever length, but for high leverage toggle calculations it can be used for lever length, with only small error, since the lever is nearly vertical.

$$T \text{ (Toggle Force)} = \frac{F \text{ (Cylinder Thrust)} \times A}{2B}$$

EXAMPLE: Find the toggle force from a cylinder thrust of 5600 lbs., if the toggle lever is 15 inches long and is ½ inch from vertical (Distance B).

SOLUTION: $T = 5600 \times 15 \div 2 \times \frac{1}{2} = 84{,}000$ lbs. This is a multiplication of 15 times the direct cylinder thrust. The remaining travel distance of the toggle arm at any point in the cylinder stroke is twice the difference between distance A and the true length, pin-to-pin, of the toggle arm. Distance A can be found by geometry or from a scale layout.

By properly setting your circuit relief valve, you can limit maximum circuit pressure to any desired value less than the maximum rating.

When connected to a cylinder, a pump will develop only that pressure required to move the cylinder piston against the load, and no more. However, when the piston stalls by hitting the cylinder end, pump pressure will rise theoretically to an infinite value. If there were no relief or safety valve in the circuit, either the driving motor would stall, the pump would break, or the cylinder, piping or valves would break. For this reason a relief valve, properly set, must ALWAYS be included in the circuit, and should be plumbed in as close to the pump pressure port as possible.

CAUTION: A relief valve built into a power-driven pump should not be used as the main circuit relief valve unless it discharges back into the oil reservoir. If it discharges back into the inlet of the pump, severe local overheating of the oil may occur in a very few minutes. These built-in relief valves are intended only for emergency protection, and should be set higher than the main relief valve.

Example: How much pump pressures in PSI (pounds per square inch) is required to develop a 30-ton thrust from a cylinder with 7-inch diameter piston?

Solution: 30 x 2000 = 60,000 pounds thrust required. From the "Hydraulic Cylinder Force" chart in this manual, a 7-inch piston has a 38.49 square inch area. Thus, 60,000 ÷ 38.49 = 1558 PSI. This means that if the pump develops 1558 pounds of pressure on every square inch of the piston surface, the total force developed will be 38.49 x 1558 = 60,000 pounds. Select a pump and drive motor with about 10 to 25% extra pressure capacity to take care of pressure losses in valves, piping, and friction in the cylinder.

MOTOR OR ENGINE HORSEPOWER

Example: What horsepower electric motor is required to develop 50 tons pressure with an 8-inch diameter ram?

Solution: Horsepower is the RATE or SPEED of doing work. Therefore, 50 tons pressure can be developed with almost any size motor. The more horsepower is available, the faster the work can be performed.

In the example it is necessary to specify how fast the 8-inch ram must MOVE the 50-ton load. Assume a speed of 15 inches per minute is desired. 50 tons = 100,000 pounds, the 8-inch ram has an area of 50.27 square inches. Pump pressure required is, therefore, 100,000 ÷ 50.27 = 2000 PSI in round figures. Oil volume in gallons per minute (GPM) required to move the ram 15 inches in one minute is 50.27 x 15 ÷ 231 = 3.3 GPM. Referring to the "Motor Horsepower to Drive a Hydraulic Pump" chart in this manual, a pump delivering 3½ GPM at 2000 PSI requires 4.9 HP.

Example: What HP gasoline or diesel engine is needed to drive a hydraulic pump at 2000 PSI and 3½ GPM?

Solution: Use the "Motor HP to Drive a Hydraulic Pump" chart in this manual. This indicates 4.9 HP. Most gasoline engines are rated at their most efficient operating speed, which is usually not the correct pump speed. The best procedure is to obtain an engine performance curve from the manufacturer to determine the actual HP output at the desired speed. Then an additional allowance of 25 to 100% should be made to take care of such intangible factors as loss of compression due to aging, loss of power with altitude, etc. An engine does not have the tremendous reserve capacity of an electric motor for sustaining short duration overloads. In this example, we recommend an engine which will develop 7½ to 10 HP at the desired operating speed. A good arrangement is to belt drive the pump. Then by the proper choice of sheave diameters, the engine can run at its most efficient speed, and the pump can be driven at its optimum speed.

HAND PUMPS

Hand pumps can develop the same cylinder force as motor driven pumps but usually at a much slower rate.

Example: A certain cylinder has a 4-inch diameter piston. Using a hand pump with 1½ cubic inches per cycle (double stroke). how many strokes of the handle would be required to move the piston 5 inches?

Solution: The Area of a 4-inch diameter piston is 12.57 square inches. The volume of oil which must be pumped to move the piston 5 inches is 12.57 x 5 =62.85 cubic inches. If the pump will deliver 1½ cubic inches in one cycle (double stroke) strokes required will be: 62.45 ÷ 1½ =42 stroking cycles.

Hydraulic Cylinder Speeds

Figures in the body of this chart are cylinder rod travel speeds in "inches per minute". Horizontal lines in heavy type are extension speeds, using the full piston area. Horizontal lines in light type are retraction speeds, using "net" piston area, for the various rod diameters shown in Column 2.

Piston Diam.	Rod Diam.	1 GPM	3 GPM	5 GPM	8 GPM	12 GPM	15 GPM	20 GPM	25 GPM	30 GPM	40 GPM	50 GPM	75 GPM
1½″	**None**	**130**	**392**	**654**	**1034**								
	⅝	158	476	792	1265								
	1	235	706	1176	1880								
1¾″	**None**	**96**	**288**	**480**	**767**	**1150**							
	¾	118	354	588	940	1410							
	1	142	428	713	1140	1710							
	1¼	196	588	980	1570	2350							
2″	**None**	**73**	**221**	**368**	**588**	**883**	**1120**						
	¾	85	257	428	685	1025	1283						
	1	97	294	490	782	1175	1465						
	1⅜	139	418	697	1115	1673	2090						
2½″	**None**	**47**	**141**	**235**	**376**	**565**	**675**	**940**	**1175**				
	1	56	168	280	448	672	840	1120	1400				
	1⅜	67	203	339	542	813	1015	1355	1695				
	1¾	92	277	463	740	1110	1385	1850	2310				
3″	**None**	**32**	**98**	**163**	**261**	**392**	**490**	**653**	**817**	**982**	**1300**		
	1	36	110	184	294	440	551	735	920	1100	1470		
	1½	43	131	218	349	523	655	872	1090	1350	1800		
	2	58	176	294	470	705	882	1175	1470	1760	2350		
3¼″	**None**	**28**	**83**	**139**	**223**	**334**	**417**	**557**	**696**	**836**	**1115**		
	1⅜	34	102	170	271	407	510	680	850	1020	1360		
	1¾	39	118	196	313	472	588	784	980	1176	1568		
	2	44	134	224	358	537	672	896	1120	1344	1792		
	2¼	53	160	267	428	642	800	1068	1335	1600	2126		
3½″	**None**	**24**	**72**	**120**	**192**	**288**	**360**	**480**	**600**	**720**	**960**	**1200**	
	1¼	27	82	137	220	330	411	548	685	822	1096	1370	
	1¾	32	96	160	256	384	480	640	800	960	1280	1600	
	2	35	107	178	285	428	534	712	890	1068	1424	1780	
	2⅜	44	134	222	356	534	666	888	1110	1332	1776	2220	
4″	**None**	**18**	**55**	**92**	**147**	**220**	**276**	**368**	**460**	**552**	**736**	**920**	
	1¼	20	61	102	163	244	306	408	510	612	816	1020	
	1¾	22	68	113	182	273	339	452	565	678	904	1130	
	2	24	73	122	196	294	366	488	610	732	976	1220	
	2½	30	90	150	241	362	450	600	750	900	1200	1500	
	2¾	35	105	174	279	418	522	696	870	1044	1392	1740	
5″	**None**	**12**	**35**	**58**	**94**	**141**	**174**	**232**	**290**	**348**	**464**	**580**	**870**
	1½	13	39	64	103	155	193	258	320	387	516	645	960
	2	14	42	70	112	168	210	280	350	420	560	700	1050
	2½	16	47	78	125	188	235	315	390	470	630	780	1170
	3	18	55	92	147	220	275	365	460	550	730	920	1380
	3½	22	66	111	178	266	333	444	555	665	888	1110	1665
6″	**None**	**8**	**24**	**41**	**65**	**98**	**123**	**162**	**202**	**245**	**320**	**405**	**606**
	1¾	9	27	45	71	107	135	180	225	270	360	445	675
	2½	10	30	50	79	118	150	200	250	300	400	495	750
	3	11	33	54	87	130	165	206	270	325	435	545	810
	3½	12	37	62	99	148	185	245	310	370	495	615	930
	4	15	44	73	117	176	220	295	365	440	585	735	1095
7″	**None**	**6**	**18**	**30**	**48**	**72**	**90**	**120**	**150**	**180**	**240**	**300**	**450**
	3	7	22	37	59	88	110	145	185	220	295	365	555
	3½	8	24	40	64	96	120	160	200	240	320	400	600
	4	9	27	45	71	107	135	180	225	270	360	445	675
	4½	10	31	51	82	122	153	205	255	305	410	515	765
	5	12	37	61	98	147	185	245	305	370	490	615	915
8″	**None**	**4**	**14**	**23**	**36**	**55**	**69**	**92**	**115**	**135**	**185**	**230**	**345**
	2¼	5	15	25	40	60	75	100	125	150	200	250	375
	3½	5½	17	28	45	68	85	115	140	170	230	285	420
	4	6	18	30	49	73	90	122	150	180	240	305	450
	4½	6½	20	33	53	80	100	135	165	200	265	335	495
	5	7½	22	38	60	90	114	150	185	225	300	375	555
	5½	8½	26	43	70	104	129	172	215	255	345	430	645
10″	**None**	**3**	**9**	**15**	**23**	**35**	**44**	**60**	**73**	**88**	**115**	**145**	**220**
	4½	3½	11	18	29	44	55	75	92	111	150	185	275
	5	4	12	20	31	47	60	80	100	120	155	195	300
	5½	4½	13	21	34	50	63	84	105	132	165	210	315
	7	5½	17	29	46	69	87	115	145	174	230	285	435

Hydraulic Cylinder Force

This chart shows the thrust developed by cylinders of various bore diameters when working at various pressures. These figures do not include frictional losses in the cylinder, which may be estimated at about 5% to 10%. Additional pressure must also be developed by the pump to overcome flow losses in the circuitry. Therefore, the useable pressure in the cylinder may be from 10% to 25% less than the pump and relief valve gauge reading.

For finding thrust at pressures not listed, for example 3000 PSI, use twice the thrust shown for 1500 PSI, or use the total of thrust for 2000 PSI plus thrust shown for 1000 PSI. For intermediate pressures, for example 1250 PSI, split the difference between value for 1500 and value for 1000 PSI, since 1250 is halfway between the two.

CYLINDER THRUST, IN POUNDS

Piston or Rod Dia.	Area Sq. In.	250 PSI	500 PSI	750 PSI	1000 PSI	1500 PSI	2000 PSI
⅝	.31	77	153	230	307	460	614
¾	.44	110	221	331	442	663	884
⅞	.60	150	300	450	600	900	1200
1	.79	196	393	589	785	1178	1571
1⅛	.99	250	500	750	1000	1500	2000
1¼	1.23	307	613	920	1227	1840	2454
1⅜	1.48	370	740	1110	1480	2220	2960
1½	1.77	442	883	1325	1767	2650	3534
1¾	2.41	601	1202	1804	2405	3607	4810
2	3.14	785	1570	2356	3141	4711	6282
2¼	3.98	1000	2000	3000	4000	6000	8000
2½	4.91	1227	2454	3682	4909	7363	9818
3	7.07	1767	3534	5301	7069	10,603	14,137
3¼	8.30	2150	4300	6450	8600	12,900	17,200
3½	9.62	2405	4811	7216	9621	14,432	19,242
4	12.57	3141	6283	9424	12,566	18,849	25,132
5	19.64	4909	9817	14,726	19,635	29,452	39,270
6	28.27	7068	14,137	21,205	28,274	42,411	56,548
7	38.49	8621	19,242	28,864	38,485	57,727	76,970
8	50.27	12,566	25,132	37,699	50,265	75,397	100,530
10	78.54	19,635	39,270	58,905	78,540	117,810	157,080
12	113.10	28,300	56,600	84,900	113,200	169,800	226,400
14	153.94	38,500	77,000	115,000	154,000	230,000	308,000
16	201.06	50,000	100,000	150,000	200,000	300,000	400,000

Cylinder Air Consumption

Figures in the second column are actual physical volumes per inch of cylinder length. Figures in the last 5 columns show the amount of free air which is contained per inch of physical volume, when compressed to the pressure shown in the column heading.

To use the chart, first find your cylinder bore diameter in Column 1, and follow this horizontal line to the column headed by your operating line pressure. Use this figure as your actual cubic foot consumption of air per inch of cylinder stroke.

To solve for CFM air consumption, multiply cylinder stroke times number of strokes per minute, then times 2 (for return stroke, neglecting rod volume), then times the factor from chart for your cylinder bore and operating PSI.

Free Air Consumption at Various Operating Pressures

Cyl. Bore Dia.	Cu. Ft. Vol. Per Inch of Stroke	Cu. Ft. Air Consumption per Inch of Stroke				
		40 PSI	60 PSI	80 PSI	100 PSI	125 PSI
1¾	.00139	.0052	.0071	.0090	.0108	.0132
2	.00182	.0068	.0093	.0117	.0142	.0172
2½	.00284	.0106	.0144	.0183	.0223	.0270
3	.00409	.0152	.0208	.0267	.0319	.0388
3½	.00555	.0207	.0282	.0357	.0433	.0527
4	.00725	.0270	.0369	.0467	.0566	.069
5	.0114	.0426	.0580	.0735	.089	.108
6	.0164	.061	.083	.105	.128	.156
8	.0285	.106	.145	.183	.222	.270
10	.0454	.170	.231	.292	.354	.430

Column Strength Of Long, Slim Piston Rods

Long, slim piston rods, when subjected to a heavy push load, may be subject to buckling (column failure). The table on this page gives recommended minimum piston rod diameters to carry various compression loads at various lengths of unsupported rod. These recommendations are based on the assumption that the rod is rigidly supported at the cylinder end by the piston and rod bearing, and that the other end of the rod is fixed to a guided member. It is further assumed there will be no side load or bending stress at any point on the piston rod.

HOW TO USE THE TABLE

Figures on the horizontal scale represent the length in inches of exposed piston rod at full extension (not cylinder stroke). Figures on the vertical scale represent the total load in tons (not the PSI load). If your piston rod is RIGIDLY anchored to a load which is firmly guided by ways or guide rods in such a manner that there cannot be any bending stress transferred to the rod, you will be safe in using for L (horizontal scale) ½ the exposed rod length.

MINIMUM PISTON ROD DIAMETER

Figures in body of chart are rod diameters in inches.

Tons Load	Exposed length of piston rod in inches 10	20	40	60	70	80	100	120
½			¾	1				
¾			13/16	1 1/16				
1		⅝	⅞	1 ⅛	1 ¼	1 ⅜		
1½		11/16	15/16	1 3/16	1 ⅜	1 ½		
2	11/16	¾	1	1 ¼	1 7/16	1 9/16	1 13/16	
3	13/16	⅞	1 ⅛	1 ⅜	1 9/16	1 ⅝	1 ⅞	
4	15/16	1	1 3/16	1 ½	1 ⅝	1 ¾	2	2 ¼
5	1	1 ⅛	1 5/16	1 9/16	1 ¾	1 ⅞	2 ⅛	2 ⅜
7 ½	1 3/16	1 ¼	1 7/16	1 ¾	1 ⅞	2	2 ¼	2 ½
10	1 ⅜	1 7/16	1 ⅝	1 ⅞	2	2 ⅛	2 7/16	2 ¾
15	1 11/16	1 ¾	1 ⅞	2 ⅛	2 ¼	2 ⅜	2 11/16	3
20	2	2	2 ⅛	2 ⅜	2 ½	2 ⅝	2 ⅞	3 ¼
30	2 ⅜	2 7/16	2 ½	2 ¾	2 ¾	2 ⅞	3 ¼	3 ½
40	2 ¾	2 ¾	2 ⅞	3	3	3 ¼	3 ½	3 ¾
50	3 ⅛	3 ⅛	3 ¼	3 ⅜	3 ½	3 ½	3 ¾	4
75	3 ¾	3 ¾	3 ⅞	4	4	4 ⅛	4 ⅜	4 ½
100	4 ⅜	4 ⅜	4 ⅜	4 ½	4 ¾	4 ¾	4 ⅞	5
150	5 ⅜	5 ⅜	5 ⅜	5 ½	5 ½	5 ½	5 ¾	6

Air Cylinder Force

IMPORTANT: An AIR cylinder must always be overpowered in order to MOVE a load. For example, a cylinder which exerts 1000 lbs. force can balance a 1000 lb. load but cannot move it. Design with at least 20 to 25% extra force for moving the load slowly, or with 100% extra force if the load must be moved fast.

When calculating cylinder force on the return (pull) stroke, deduct rod area from piston area. On double rod cylinders, deduct rod area for both directions.

An area chart for other piston diameters appears elsewhere in this manual. For conditions not shown, multiply PSI gauge pressure times piston area in square inches.

AIR CYLINDER FORCE IN POUNDS

Bore Dia.	Line pressure in pounds per sq. inch (gauge) 30	40	50	60	70	80	90	100	110	120	Piston Area
1¾	71	95	119	140	166	190	213	237	261	284	2.37
2	94	125	157	188	220	250	283	314	346	377	3.14
2½	145	193	242	290	338	387	432	483	530	580	4.83
3	212	282	353	424	494	565	635	706	777	847	7.06
3½	288	385	480	576	673	768	865	962	1005	1150	9.62
4	376	503	628	753	880	1000	1130	1256	1380	1500	12.56
5	588	785	980	1175	1375	1570	1765	1964	2160	2350	19.64
6	850	1130	1410	1700	1980	2260	2550	2827	3100	3400	28.27
8	1500	2000	2500	3000	3500	4000	4500	5000	5500	6000	50.27
10	2350	3150	3900	4700	5500	6250	7000	7850	8600	9400	78.54

Air Cylinder Speed

A Guide for Selection of Valve Port Size

The exact speed of an air cylinder cannot be calculated because it depends on the degree of unbalance between the load resistance and the force developed by air pressure acting on the piston, and the rate at which the space ahead of the advancing piston can be vented. Many factors which affect the speed of an air cylinder are not known and cannot be determined.

Where fast speed is required, select the bore size and line pressure which will develop about twice the thrust needed to balance the load resistance. Then choose a directional valve of ample size and use short lines of generous size. If moderate speed is adequate, the cylinder need be oversized only about 20 to 25%.

A good rule in selecting a directional valve is to use one with orifice diameter equal to cylinder connection size. Certain small valves have an orifice much smaller than their connection size. This chart shows proper valve size under average conditions.

ESTIMATED CYLINDER SPEED

Figures in the body of the table are speeds in inches per second.

Cyl. Bore ↓	ACTUAL VALVE ORIFICE SIZE 1/32″	1/16″	⅛″	¼″	⅜″	½″	¾″	1″
1	6	15	37	110				
1⅛	5	12	28	85				
1½	3	7	16	50	125			
1¾	2	5	11	35	87	140		
2	1	4	9	28	70	112		
2½		2	6	18	45	72	155	
3		1	4	12	30	48	100	
3½			3	9	22	36	78	165
4			2	7	17	28	60	130
5			1	4	11	18	40	82
6				3	7	12	26	55
8				1	4	7	15	32
10					2	4	9	20
12					1	3	6	14

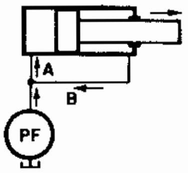
NOTE: This chart is for average conditions where the cylinder has twice the thrust needed to overcome the load resistance, and is operating on an 80 to 100 PSI air line with an ample air supply, and operated with a 4-way valve with full orifice.

Air Line Pipe Size

Figures in the body of the chart are the black iron pipe sizes suggested for long runs, to keep pressure loss to a reasonable value. The pipe size shown should be used ALL the way back to the compressor. For example, 25 CFM requires ¾″ pipe for distances to 200 feet. For 200 feet and over, 1″ pipe should be used the ENTIRE distance. If CFM air flow is not known, use the second column as a guide, allowing 3 to 4 CFM air flow for every 1HP of the compressor, if the air flow is at a uniform rate.

Air Flow, CFM	Comp. HP.	25 Feet	50 Feet	75 Feet	100 Feet	150 Feet	200 Feet	250 Feet	300 Feet
5 or Less	1.4	½″ →	→	→	→	→	→	→	→
10	2.8	½″ →	→	→	¾″ →	→	→	→	→
15	4.3	½″ →	¾″ →	→	→	→	→	→	→
20	5.6	¾″ →	→	→	→	→	→	→	→
25	7.0	¾″ →	→	→	→	→	1″ →	→	→
30	8.5	¾″ →	→	→	→	1″ →	→	→	→
35	10.0	¾″ →	→	1″ →	→	→	→	→	→
40	11.2	¾″ →	1″ →	→	→	→	→	→	→
50	14.0	1″ →	→	→	→	→	→	→	→
70	20.0	1″ →	→	→	→	1¼″ →	→	→	→

Regenerative Circuit Calculations



A regenerative circuit is sometimes employed to cause a cylinder to advance more rapidly than it could with the pump volume alone. It can only be used to extend a cylinder — never to retract it.

The basic principle of regeneration is, by means of suitable valving, to connect the rod end of the cylinder with the blind end, so the oil which normally would be discharged to tank from the rod end will join the pump oil causing the cylinder to advance at an increased rate of speed.

The accompanying circuit is only a schematic representation of the regeneration principle. Actual workable circuits are presented on the next page.

CYLINDER FORCE. Since equal pressure is applied to both sides of the piston, the net thrust delivered by the rod will be the same as if the pump pressure were applied only to the rod area. Therefore, thrust is pump PSI x rod area.

CYLINDER SPEED. Since return oil from the rod end fills up an equivalent volume on the blind side of the piston, the pump volume need only fill up a space equivalent to the volume of the rod. Therefore, to calculate rod speed, take pump volume, in cubic inches per minute, and divide the rod area (in square inches) into it. This will give speed in inches per minute.

OIL FLOW VOLUME. First, calculate rod speed as in paragraph above. To find oil flow at Point "A", calculate how much oil will have to flow to make the piston travel at this speed. This will be speed (inches per minute) x piston area (square inches). Convert to GPM by dividing by 231 (cubic inches to the gallon). To find the oil flow at Point "B", take the above result and subtract the pump volume from it.

SAMPLE CALCULATION. Assume a pump pressure of 1200 PSI, pump volume of 8 GPM, piston diameter 10", rod diameter 7". Force = 38 square inches (rod area) x 1200 PSI = 45,600 lbs. Speed = 8 GPM x 231 (cu. ins./gal.) ÷ 38 square inches = 48" per minute. Oil flow at "A" = 48 x 78.5 (piston area) ÷ 231 = 16.3 GPM. Oil flow at "B" = 16.3 — 8 GPM (from pump) = 8.3 GPM.

Notes on Regenerative Circuits

1. In regeneration, the force generated will be that of pump pressure acting only on the rod area. The remainder of the piston area is cancelled out by an equal and opposing area on the rod side.

2. If and when the full force of the cylinder is required, the rod end must be disconnected from the blind end and connected to tank. The circuits on the next page show valving for doing this.

3. Regeneration is mainly used with big rod cylinders, especially 2:1 (piston to rod area ratio). If used with small rod cylinders, the extending speed is too great, the thrust too small, and the return speed too slow.

4. When a 2:1 ratio cylinder is retracting, discharge oil from the blind end is twice the pump volume. Select valving large enough to handle this volume.

5. With 2:1 ratio cylinders (usually used in regenerative circuits), pressure intensification occurs in the rod end during the forward stroke if the discharge oil is restricted or blocked. Install a safety relief valve in the rod end if intensification could endanger the cylinder or plumbing.

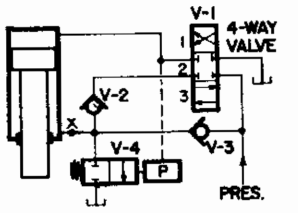
6. The regenerative portion of the cycle is usually for moving the machine part rapidly into working position, and the actual thrust required is relatively slight. Therefore, it is permissible, and good practice, to allow the oil velocity to be high in the valving and plumbing. The high pressure drops developed will not be harmful, and considerable expense can be saved by using smaller valving and plumbing than would be considered good practice for more conventional circuits.

7. Because large rod cylinders are used in regenerative circuits, the oil level in the reservoir will fluctuate more than for small rod cylinders. Make sure the reservoir is large enough so the oil level will not drop dangerously low as the cylinder extends.

Typical Regenerative Circuits

Circuits on this page are limited to bare essentials of the regenerative valving, with pump, relief valve, and accessories omitted.

A regenerative circuit using an accumulator will be found under "Accumulator Circuits" in this manual.

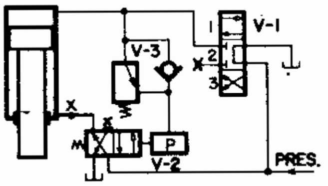


Shifting the 4-way to Position 1 causes the cylinder to start extending. It is in regeneration because discharge oil from rod end passes through check valve V-3 and joins pump oil to the blind end of the cylinder. Circuit stays in regeneration until or unless work resistance builds up on pilot of V-4 causing it to shift. Rod oil then goes directly to tank and circuit becomes nonregenerative and capable of developing full tonnage.

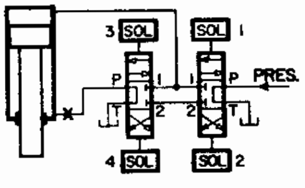
Shifting the 4-way to Position 3 causes the cylinder to retract, pump oil passing through check valve V-2. Valve V-4, with no pilot pressure, closes, preventing pump oil from by-passing to tank.

To keep the cylinder from dropping by gravity, add a counterbalance valve at Point X. Valve V-4 must be a spool-type (which has better throttling characteristics) rather than a poppet type. A pilot-operated check valve does not work well.

Directional Valve V-1 is shown with closed center spool. It may have any other spool center in which both cylinder ports are isolated from the pump in neutral, such as a tandem center spool for unloading the pump.



A variation of the circuit above. Valve V-1 is standard 4-way with "CYL 2" port plugged and cylinder connected to "CYL 1" port V-3 is a ¼" sequence valve with internal free flow check. It must be connected for internal pilot, external drain. Valve V-2 is a standard 4-way, pilot operated, with "CYL 2" plugged. It can have about half the capacity of V-1.

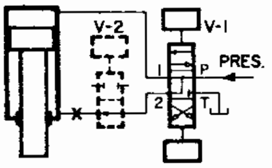


Solenoid control of regenerative forward, normal forward, reverse, or stop, at any point in the cylinder stroke.

Energize Solenoids 1 and 3 for regenerative forward, Solenoid 1 only for normal forward, Solenoids 2 and 4 for retract. De-energize all solenoids for stop.

Add counterbalance valve, if needed, at Point "X". A pressure switch connected to the blind end of the cylinder will give automatic changeover from regeneration to normal. CAUTION: Use a holding relay in conjunction with the pressure switch, wired to take the pressure switch out of the circuit for the rest of the cycle once it has tripped. Otherwise, a condition of "hunting" will occur during the period of changeover.

A microswitch may be mounted so as to be actuated by the ram at a certain point in its travel, to take the circuit out of regeneration.



The spool of Valve V-1 has both cylinder ports connected to pressure when it is in center position. This is the regenerative position. The side positions of V-1 are normal extend, and retract. If necessary to stop the cylinder in a mid position, Valve V-2 must be installed. If a counterbalance valve is needed, install it at Position "X". NOTE: A 4-way valve with spool as shown usually has about ½ rated flow capacity when in center position. Therefore, choose a valve with double the usual size for this circuit.

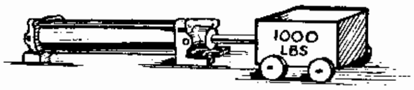
Moving Horizontal Loads

SLIDING LOAD. Cylinders may be used for moving high-friction sliding loads such as machine slides, lathe tailstocks, milling machine and grinder tables.

On rapid indexing from one positive to another positive stop, an air cylinder will usually give more rapid action than hydraulics, if the load is within its capacity. DO NOT use an air cylinder for slow or controlled feeding of a slide which has a large area of surface friction. The motion, most probably, will be jerky. Use a hydraulic cylinder with meter-out speed control for this. In some cases an air-over-oil system is satisfactory.

How much cylinder force is needed to push a sliding weight? This varies with the surface materials, lubrication, unit loading, speed, and other factors to some degree. For machine slides, lightly lubricated, the cylinder should be figured with a thrust equal to 1/2 to 3/4 of the load weight, to get the load started. Once in motion, a thrust of 1/5 to 1/6 the load weight will keep it moving.

Where high speed is required of an air cylinder, it should be sized to develop at least twice the thrust needed to just balance the load.

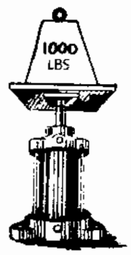


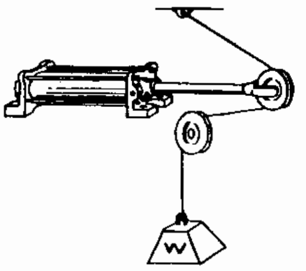
ROLLING LOAD. Cylinders are sometimes used to move loads which operate on low-friction needle, roller, or ball bearings. As a rule, these require a cylinder thrust of about 1/10th the load weight. On some applications an air cylinder with meter-out speed controls can be successfully used even at slow feed rates. Attention should be given to deceleration at the end of cylinder travel to prevent load momentum energy from damaging the cylinder or the machine.

Cylinders for Lifting

VERTICAL AIR LIFT. An air cylinder must always be overpowered in order to MOVE or LIFT a load. For example, a cylinder which exerts 1000 lbs. force can BALANCE a 1000 lb. load but cannot move it. The more the air cylinder is overpowered, the faster it will move the load. For normal use, choose a cylinder bore diameter which will give at least 25% more thrust than needed to balance the load. If very fast movement is required, the cylinder should be capable of developing about twice the force needed to just balance the load.

An air cylinder is not satisfactory for a platform lift if the lift is to be stopped at some intermediate point in the stroke for loading or unloading. Because of the compressibility of air, the lift would rise or sag as the loading was decreased or increased. A hydraulic system or an air-over-oil system must be employed for these applications.





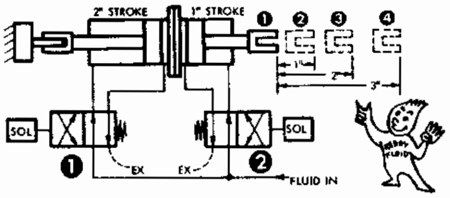
DIFFERENTIAL LIFT. This arrangement gives a 2:1 mechanical reduction. The cylinder is sized with twice the piston AREA but with half the stroke that would be used for a straight lift of the same load.

The pulley attached to the cylinder rod should run in horizontal guides to remove possible side thrust from the cylinder.

This is sometimes an ideal solution to a lift problem where head room is not sufficient for a direct lift. The shorter length, larger diameter, is usually a better cylinder configuration. Other advantages are that the cylinder is working on full piston area, and the rod packings are not subjected to high pressure.

Multiple-Position Indexing

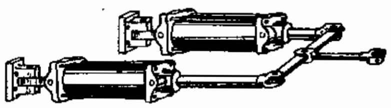
TANDEM CYLINDERS. Two cylinders of different stroke lengths can be used to give four positive indexing positions when joined back-to-back and operated with two 4-way valves in the circuit below. This is especially useful on air cylinders, as a single air cylinder is very unstable when stopped in a mid-position. The same idea can be extended by adding more cylinders to the string for additional fixed positions. On long strings, a stroke adjustment collar can be used on each cylinder to permit very precise adjustment of the stroke. This will prevent slight errors in the stroke from accumulating into large errors on long strings.



In this circuit for the tandem cylinders shown above, the two cylinders are chosen with strokes of 1" and 2", to illustrate four indexed positions of 0, 1, 2, and 3 inches.

With solenoid Valves 1 and 2 de-energized, both cylinders are retracted and the rod clevis is at Position 1. With both valves energized, both cylinders are extended, and the rod clevis is extended 3 inches to Position No. 4. If only Valve 2 is energized, the cylinder rod would extend 1 inch to Position No. 2. In Position No. 3. Solenoid valves are illustrated, but any other type could be used.

NOTE: If both cylinders have the same stroke, only three positions will be obtained.



YOKED CYLINDERS. It may sometimes be more convenient to obtain several fixed index positions from air cylinders using the yoking method.

The output link can either be located in the center of the cross yoke, or off center, to obtain certain desired positions.

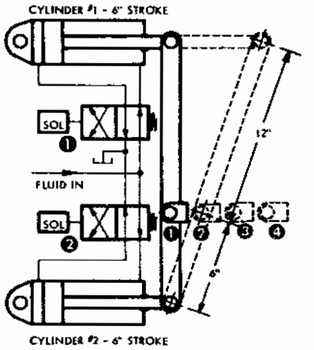
Much the same circuitry is used here as in the previous example, and is illustrated below. Although the valves are shown as solenoid type, other actuators, such as manual lever, may be used. The various positions are obtained by energizing either one or both valves, or de-energizing them both.

The fluid circuit for the yoked cylinders is shown here. A separate 4-way valve is employed for each cylinder.

It is important to use hinge mounted cylinders and to use tangs, clevis', or universal joints on the rods so the cylinders can swing with the stroke. The output clevis can be made to travel in a guide for stability.

Many variations and position possibilities can be attained by choosing cylinders with different strokes, and by coupling to the yoking bar at different points.

For example, with two 6" stroke cylinders, as shown, and with the output clevis located ⅓ the length of the crossbar, equal positions spaced 2" apart are obtained.



Rotary Actuators

GENERAL ENGINEERING NOTES

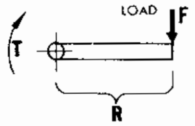
Rotary actuators or "oscillating motors' are used to provide high torque at low speed. Their rotation is limited to less than 1 turn in each direction. The actuators referred to on these two pages are of the vane type.

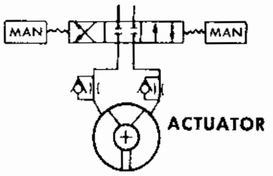
TORQUE is a turning or twisting moment produced in the rotary actuator by pressure acting against the driving area of the rotor vane(s).

To find the force, F, produced at a distance, R, by torque, T, developed at the actuator shaft:

$$F = T \div R$$

The usual units are (inch-lbs.) torque, to produce (lbs.) force at (inches) distance from the shaft.





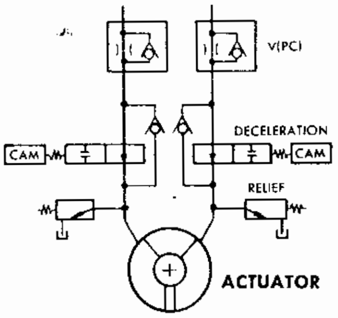
DIRECTIONAL CONTROL is accomplished with a 4-way directional valve in exactly the same manner as for a double-acting cylinder. There are many circuits in this manual illustrating this.

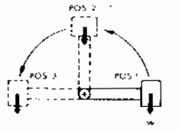
SPEED CONTROL for slow speed, lightly loaded applications is done with flow control valves the same as for cylinders, as illustrated here.

EXTERNAL STOPS mounted securely to machine framework should be used to stop the load. The rotor vanes should not contact the internal stops except under very light loads.

SEVERE SHOCK and possible damage to the system can occur on hydraulic applications by too sudden or complete restriction of outgoing fluid, which allows the moving mass to generate high surge or transient shock wave pressures.

DE-CELERATION VALVES, actuated mechanically, as shown, or by limit switches, are often used to more gradually restrict the fluid and stop the moving mass. Usually, relief valves plumbed as shown, or plumbed from one line to the other, each direction, will limit the generation of surge pressures to a safe value. If cam valves are used, the cam shape should provide a gentle ramp transition, and the spool should be tapered to provide a gradual closing off of fluid.





LOWERING a mass from Position 2 and stopping it at Position 1 or 3 may require a deceleration valve. However, if the load is to be stopped in a straight downward position, the load weight helps to decelerate at each end of the travel, and deceleration valves may not be required.

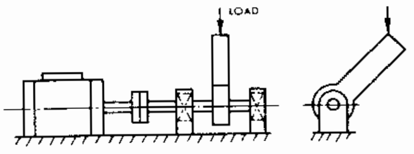
AIR BLEEDING in hydraulic systems is usually not required if actuator is mounted with supply ports upward. In other positions, air will gradually dissolve in the oil and be carried away as the actuator is cycled. Special bleed connections are available as an optional feature on some actuators, if specified when ordering.

STATIC-FRICTION BREAK-AWAY pressure does not usually exceed 30 PSI for hydraulic units, and 15 PSI for pneumatic units.

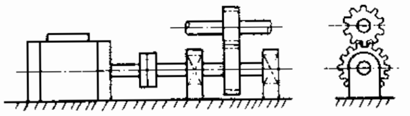
INTERNAL BY-PASS FLOW is always present to a small degree, and increases with increase of pressure. On air applications it must be recognized that on stall-out applications, under air pressure, there will be a small continuous by-pass flow.

Pure Torque Output

If used at less than maximum ratings, rotary actuators can often tolerate a certain amount of side or end loading of the shaft. However, for maximum bearing life under fully loaded conditions, it is essential that side loading be carried on a jack shaft supported in bearing blocks.



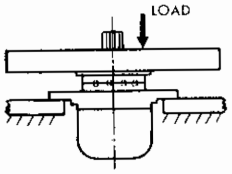
Gear Drive



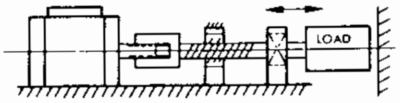
This sketch shows an actuator driving a gear train. The actuator should be coupled with a flexible or semi-flexible coupling, or, if this is not possible, the actuator and associated equipment must be very accurately aligned to prevent undue actuator bearing loading. A pilot boss on the actuator, fitting closely into an accurately machined pilot recess in the mating equipment is often advised as a means of aligning shaft centers precisely.

End Loading

End thrust or axial loading of the actuator shaft is not advised. A thrust bearing between actuator and load, driven through sliding spline or other means, is recommended to minimize internal wear in the actuator and allow maximum life.



Screw Clamping

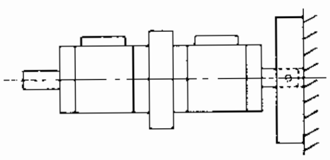


Clamping by means of a steep angle screw powered with a rotary actuator. Screw is rotated in a stationary nut. The coupling slides on a splined shaft to relieve actuator of axial loading. Loss of pressure will not allow the clamp to release.

The fluid circuit should have greater pressure for unclamping because of the wedging action of the screw threads.

Up to 1½ Turns Rotation

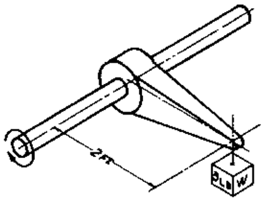
To obtain full 1 turn, or up to 1½ turns, two rotary actuators may be rigidly joined back-to-back, with one shaft anchored, the other shaft free to rotate the work. Usually a special adapter, manufactured by the customer, is required to join the two units. Connections must be made with trailing hoses.



Hydraulic Motors

Selection

When selecting a hydraulic motor you must first estimate or measure how much torque (not horsepower) is required to overcome starting (breakaway) friction of the load. Torque is defined as "rotary force," and is obtained by multiplying length of arm times the weight or force applied at the end of the arm. Sometimes a simple experimental system can be rigged up similar to the one shown in the diagram, using a spring scale or known weight to find how much torque is actually needed to get the load started.



The next step is to tentatively decide on a maximum hydraulic system pressure. Then examine the published characteristics of various motors, and select one which has about 25% more torque than actually needed. This is necessary since motors have from 15 to 25 percent less starting torque than running torque.

Electric Motor Torque

If you already know the electric motor size which will do the job, use this table to find its normal full load torque. The hydraulic motor needs to be sized about 75% above this because the electric motor (3 phase) can develop at least 50% more on starting, and the hydraulic motor may be short about 25% on starting.

Speed

To obtain maximum HP, operate the hydraulic motor at the highest practical speed, and usually not less than 500 RPM, except motors built specifically for slow speed. When reduction gearing is used the torque is multiplied in proportion.

Speed may be controlled with either a needle valve or pressure compensated flow control valve. If a by-pass type of speed control can be used rather than the series type, less heat will be generated in the oil. Motors may be obtained with reversible rotation, and the direction can be controlled with the same 4-way valve that would be used for a cylinder. CAUTION: If the motor is coupled to a high momentum load, a double cushion valve should be installed in the motor lines to absorb the shock when the 4-way valve is centered or reversed.

Oil Volume Required

To figure the GPM (gallons per minute) required for driving a fluid motor, first look up the motor displacement. This is usually given in cubic inches per revolution, GPM per 100 RPM, etc. Multiply by RPM or hundreds of RPM as the case may require, then, if necessary, convert to GPM using: 231 cubic inches = 1 U.S. gallon.

Example: A certain fluid motor has a displacement of 4.2 cubic inches per revolution. How many GPM of oil are needed to run it at 1200 RPM?

SOLUTION: 4.2 × 1200 = 5040 cubic inches. Then: 5040 ÷ 231 = 22 GPM required.

TORQUE CHART

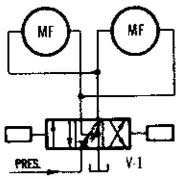
Please note that all torque values in this table are in foot pounds. Multiply by 12 to get inch pounds. For values not in the table, use one of these formulae:

Torque = HP x 5252 ÷ RPM **HP = Torque x RPM ÷ 5252**

HP	100 RPM	500 RPM	750 RPM	1000 RPM	1200 RPM	1500 RPM	1800 RPM	2400 RPM	3000 RPM	3600 RPM
¼	13.1	2.63	1.76	1.31	1.10	.876	.730	.548	.438	.365
⅓	17.5	3.50	2.34	1.75	1.46	1.17	.972	.730	.584	.486
½	26.3	5.25	3.50	2.63	2.20	1.75	1.46	1.10	.875	.730
¾	39.4	7.87	5.24	3.94	3.28	2.62	2.18	1.64	1.31	1.09
1	52.5	10.5	7.00	5.25	4.38	3.50	2.92	2.19	1.75	1.47
1½	78.8	15.7	10.5	7.88	6.56	5.26	4.38	3.28	2.63	2.19
2	105	21.0	14.0	10.5	8.76	7.00	5.84	4.38	3.50	2.92
3	158	31.5	21.0	15.8	13.1	10.5	8.76	6.57	5.25	4.38
5	263	52.5	35.0	26.3	22.0	17.5	14.6	11.0	8.75	7.30
7½	394	78.8	53.2	39.4	32.8	26.6	21.8	16.4	13.1	10.9
10	525	105	70.0	52.5	43.8	35.0	29.2	21.9	17.5	14.6
15	788	158	105	78.8	65.6	52.6	43.8	32.8	26.5	21.9
20	1050	210	140	105	87.6	70.0	58.4	43.8	35.0	29.2
25	1313	263	175	131	110	87.7	73.0	54.8	43.8	36.5
30	1576	315	210	158	131	105	87.4	65.7	52.6	43.7
40	2100	420	280	210	175	140	116	87.5	70.0	58.2
50	2626	523	350	263	220	175	146	110	87.5	72.8
60	3151	630	420	315	262	210	175	131	105	87.4

Hydraulic Motor Circuits

These are partial circuits, illustrating some of the ways hydraulic motors can be connected and controlled. Each circuit is condensed to its simplest form, to illustrate one basic idea. These circuits can be combined with other hydraulic circuitry to form a complete working hydraulic system.

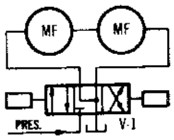


Parallel Motors

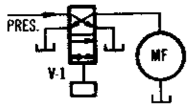
Two identical motors connected in parrallel will develop twice the torque and half the speed as one of these motors working from the same pump. Unless the motors are mechanically tied together in some way, more oil will go to the motor with the lighter load. Sometimes, flow splitting valves must be used to make the oil divide equally.

Series Motors

Two identical motors in series will run approximately the same speed regardless of the difference in load between them. They will divide the available pump pressure in proportion to the load on each. Make sure that motors used in a series circuit are capable of having both ports pressurized.



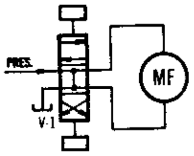
Single-Direction Rotation

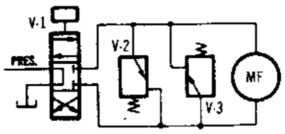


A standard 4-way, 2-position valve is used to start and stop the hydraulic motor. In the stop position, shown on the sketch, the motor can "free wheel," that is, it can coast to a stop, or can be rotated manually. Also, in the stop position the pump is unloaded.

Reversible Rotation Motor

Direction of rotation is controlled with a standard 3-position, 4-way valve used in exactly the way it would be used to control a double-acting cylinder. In center position, the pump is unloaded to tank, and the motor can "free wheel."



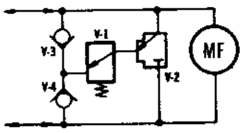


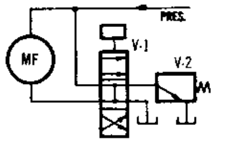
Manual Throttle Braking

Controlled braking by throttling toward center position of a manual valve. Reversible motor operation with pump unloaded in center position. V-2 and V-3 must be used to prevent rupture of hydraulic system if valve is accidentally centered abruptly.

Surge Relief Circuit

Similar to above circuit, but with single relief valve, V-1, acting as safety relief for both directions of rotation, through network of V-2 shuttle valve, and V-3 and V-4 check valves.

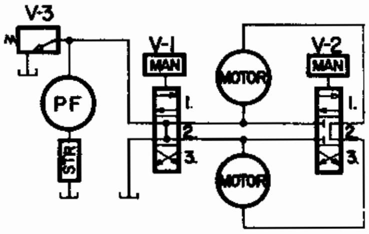




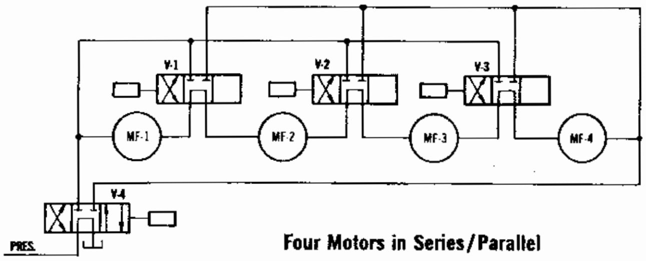
Relief Valve Braking

Single-direction rotation, with "free-wheeling" in valve top position with V-2 serving as the main circuit relief valve, and braking in valve lower position with V-2 acting as brake relief valve.

Series/Parallel Motors



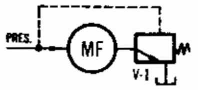
For vehicle wheel drive, two hydraulic motors may be used in series for high speed travel, and in parallel for high torque at low speed. Valve V-2 is the "speed shift", giving free wheeling in Position 1 for towing. Valve V-1 is the directional control with pump unloading in center position. Valve 3 is the regular relief valve.



Four Motors in Series/Parallel

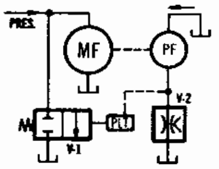
With V-1, V-2, and V-3 in center position as shown, all four motors are in series. With these three valves thrown to the left, all four motors are in parallel. With V-1 and V-3 in center position, and V-2 thrown to the left, MF-1 is in series with MF-2. MF-3, is in series with MF-4, and these two groups are in parallel with each other. The right hand position of these three valves, shown blank on the drawing, is not used in this circuitry, and may be blocked off on the valves. This gives speed and torque ratios of 1, 2, or 4 without using any mechanical transmission. Valve V-4 is the directional control for all motors.

Over-Run Limiter



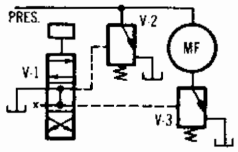
Use this circuit to prevent a hydraulic motor from over-running, as on a vehicle travelling downhill. Valve V-1 is a 2-way hydraulic by-pass valve, of spool-type construction preferably, connected for external pilot, internal drain operation.

Constant Motor Speed



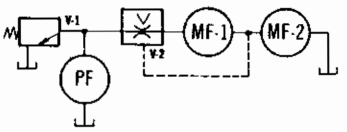
To keep motor, MF, running at constant speed, a small pump coupled to it generates pilot pressure across needle valve, V-2, to operate a 2-way by-pass valve, V-1. If the motor tends to overspeed, the higher pilot pressure generated causes V-1 to by-pass more circuit oil, slowing the motor to normal speed.

Controlling Large Motors



A small, 3-position valve, V-1, controls a very large, single-direction motor. V-2 and V-3 are pilot operated reliefs with vent connections. Center position is "free wheeling," as V-2 and V-3 are both vented. Top position is braking position, with V-2 vented and by-passing circuit oil to tank. Bottom position is running position with brake valve, V-3, vented, and V-2 as circuit relief valve.

Conveyor Drives



Modified series connection of two identical motors in which the second motor must run slightly faster than the first. Example: MF-1 driving a feeding conveyor, MF-2 driving a carry-off conveyor. V-2 is a by-pass type flow control with the excess oil, shown in dotted lines, plumbed back into the second motor inlet.

Accumulator Sizing

1. First, the cubic inches of oil needed from the accumulator on each discharge must be calculated or estimated.

2. The accumulator pressure, during discharge, will drop from a maximum fully charged pressure to a lower pressure. This means the system must be designed so sufficient cylinder force can be obtained at the lower (minimum) pressure.

3. With the data from Steps 1 and 2, use this chart to ascertain the gallonage capacity of the accumulator to meet these conditions. We suggest pre-charging the accumulator with dry (oil pumped) nitrogen to about ⅓ to ½ the maximum system pressure. The chart is for piston accumulators, but can be used for bladder accumulators, remembering that a bladder accumulator should not be completely discharged on every cycle because of ultimate damage to the bladder. Accumulator volume is specified as total oil and gas volume, or gas volume when discharged.

SELECTION CHART

This chart shows the cubic inches of oil delivered by a "1 gallon" piston accumulator in working from a maximum system pressure shown along the top of the chart to one of the lower pressures shown in the extreme left column. A "3-gallon" accumulator would deliver 3 times this amount and a "5-gallon" accumulator 5 times this amount, etc.

The chart is based on the accumulator having a gas pre-charge of one-half the maximum system pressure, as suggested in our design recommendations.

INSTRUCTIONS: Find your maximum system pressure along the top of the chart, follow down the column to the intersection with the minimum design pressure shown in the left column. This is the working oil available from a 1-gallon accumulator.

Minimum Hydraulic System Pressure	Maximum Hydraulic System Pressure — PSI									
	3000	2750	2500	2250	2000	1750	1500	1250	1000	750
2700	12									
2600	17									
2500	22	11								
2400	27	16								
2300	33	21	9							
2200	40	26	15							
2100	46	33	22	9						
2000	55	41	28	15						
1900	64	49	35	21	6					
1800	74	58	44	27	12					
1700	84	67	52	35	20	2				
1600	96	79	62	45	27	10				
1500	110	91	73	55	37	18				
1400		100	86	66	47	27	8			
1300			100	81	59	38	18			
1200				96	73	50	28			
1100					88	66	40	17		
1000						83	55	28		
900						100	75	43	13	
800							95	62	28	
700								86	47	9
600									75	28
500										66

Cubic inches of oil delivered by a "1 gallon" accumulator

Note 1: 231 cubic inches equals 1 U.S. gallon.

Note 2: The above chart is calculated on oil volumes 5% less than obtained for ideal (isothermal behavior of the gas) conditions. On applications where cycling is infrequent but is extremely rapid when it does occur, the actual oil delivered could be about 15% less than shown in the chart.

Note 3: To calculate oil volumes at other pre-charge pressures, use the formula:

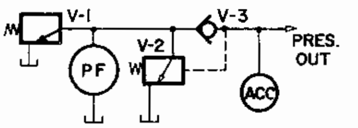
$$P_1V_1 = P_2V_2$$

where P_1 is pre-charge pressure, V_1 is total cubic inch volume of the accumulator, P_2 is the maximum hydraulic system pressure, V_2 is gas volume (in cubic inches) at pressure P_2. Solve for V_2, then subtract this from the total volume, V_1. Pressures are absolute (gauge plus 14.7).

Accumulator Circuits

See Preceding Page for Accumulator Selection

Circuits shown on these pages are stripped down to bare essentials, with accessory items such a gauges, filters, cut-out valves omitted for the sake of simplicity. Each circuit presents one basic idea for the use of accumulators, and it may be possible to combine elements of more than one circuit to obtain a total circuit with the desired characteristics.

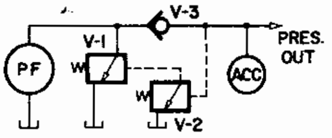


Accumulator unloading. For low volume pumps, up to 20 GPM. When accumulator has been charged up to the setting on V-2, pilot pressure (shown by dotted line) causes V-2 to dump the full pump output to tank, allowing the pump to idle at practically zero pressure. When accumulator terminal pressure falls 20 to 25%, V-2 loads the pump and starts charging the accumulator. Check valve V-3 prevents loss of accumulator oil through V-2 when the pump is idling.

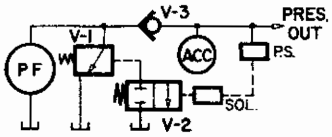
Valve V-2, accumulator unloading valve, is a snap-action valve designed specifically for this use. An ordinary by-pass or unloading valve will not work.

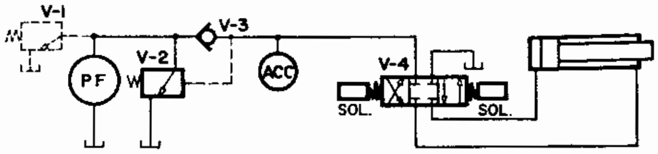
Valve 1, circuit relief valve, is optional in this circuit. It is generally included as an additional safety feature, although the unloading valve as long as it is operating properly will protect the pump.

Accumulator unloading. For high volume pumps. V-1 is a pilot operated relief valve with a remote control or venting connection. It is sized to handle the full pump volume. V-2 is a standard accumulator unloading valve of small capacity used to control the vent connection of V-1. As mentioned above, V-2 must be especially designed for this type of service.



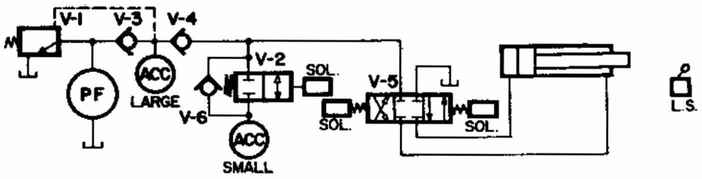
Accumulator unloading. Electrical method. Similar to circuit above except that a pressure switch and 2-way solenoid valve replace V-2. This system sometimes fits in better with circuits using other solenoid controlled valves. By choosing a pressure switch with adjustable differential, the cut-in and cut-out pressures can be adjusted to suit.



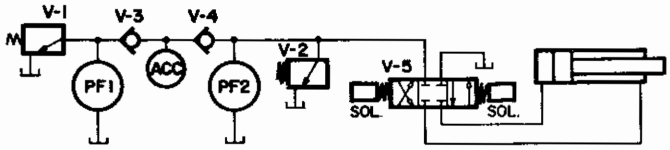


Basic accumulator circuit. For circuits in which oil flow is required intermittently, with relatively long resting periods in between. A low volume pump, running continuously, can store pressurized oil in the accumulator, to be used in large volumes for short periods. Cylinder bore diameter must be large enough so that sufficient thrust will be obtained even at the low point of the pressure cycle, just before the pump is loaded to re-charge the accumulator. Recommendations on the preceding page may be followed to determine the total accumulator volume needed for a given design. Re-charging of the accumulator may take place at any part of the cycle — loading, curing, etc. The use of relief Valve V-1 is optional for extra safety.

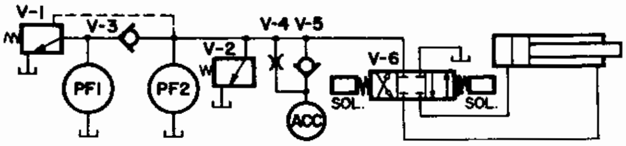
Accumulator Circuits — Cont'd.



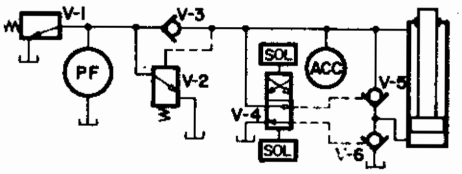
Hi-Lo Circuit. The use of one large and one small accumulator is arranged to perform much like a hi-lo pump circuit. During resting, both accumulators are charged to full system pressure. When V-5 is shifted, oil from the large accumulator provides the necessary volume for rapidly advancing the cylinder, with oil trapped and held at full pressure in the small accumulator during this phase. When the limit switch is actuated, oil from the small accumulator is released by V-2 to do the high pressure holding. This high pressure oil closes check valve V-4, and the pump starts replenishing the large accumulator during the holding cycle. At any time in the loading, pressing, or curing part of the cycle that the accumulators become fully charged, V-1, which is an accumulator unloading valve, will unload the pump.



Fast closing of the cylinder is made possible using a fairly small size traverse pump, PF-1. When cylinder bottoms out against the work load, PF-2, the low volume, high pressure pump, takes over to do the high pressure work, while PF-1 is re-charging the accumulator. V-1 may be an accumulator unloading valve if desired.



Economical circuit for long holding cycles. Rapid traverse of the cylinder is done with combined oil from both pumps and the accumulator. When cylinder bottoms out against the work load, PF2 takes over and has enough oil volume to mtaintain full work holding pressure, with the excess oil charging the accumulator through the restrictor valve, V-4. During this time PF-1 is completely unloaded with V-1, which may be any type of pilot controlled by-pass valve. A variation of this circuit would be to omit PF-1, V-1, and V-3 if cycle time is sufficiently long.



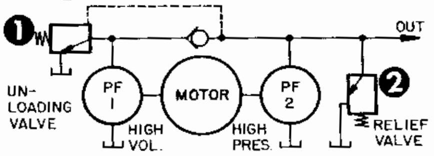
Regeneration plus accumulator oil plus pump oil combine to give very rapid advance. Pump plus accumulator plus small rod area combine to give very rapid retraction. Circuit as shown is for high speed travel at low thrust, as no valving has been provided to get full thrust on the blind end at any time. For best results, use a 2:1 ratio cylinder. V-5 and V-6 are pilot operated check valves for connecting cylinder blind end either to tank or to rod end. V-4 may be a small valve, as it only handles pilot oil for V-5 and V-6. Valve V-2 is the conventional unloading valve for the pump when accumulator becomes fully charged. This circuit works best in those applications requiring high speed, but where there is sufficient time between cycles for the accumulator to take charge. Do not use this on continuously reciprocating applications.

"High-Low" Systems

Many systems require a high volume at low pressure for rapid traverse up to the work, and then low volume, high pressure for clamping, feeding, pressing, etc. Sometimes this can be done most economically with two pumps. During rapid traverse, both forward and reverse, oil is supplied by both pumps working together. When work resistance causes the system pressure to rise to the "set" unloading pressure, the high volume pump is automatically dumped by the unloading valve which receives pilot pressure from the main pressure line.

Valve 1 is a spool-type by-pass valve with external pilot, internal drain. Valve 2 is a pilot-operated relief capable of handling combined volume of both pumps.

"High-Low" System Design



1. The "set" unloading pressure is usually at the point where the driving motor has reached full capacity driving both pumps. Then, when the large pump has been unloaded or "dumped," all the motor power is available for driving the remaining pump to a much higher pressure. Thus, motor HP must be sufficient to drive both pumps at least up to the minimum pressure required for traversing forward or reverse, whichever is greater.

2. The motor must have sufficient HP to carry the high pressure pump up to the maximum desired system pressure while the other pump is idling.

3. The combined volume output of both pumps together must be able to traverse the cylinder(s) at the desired speed.

4. A "high-low" system cannot be used where the maximum pressure is required for the full length of both forward and return stroke, although it is sometimes practical to use where full pressure is required in one direction for the entire stroke, but low pressure is sufficient for the return stroke. Examples of this would be broaching or extruding.

5. A "high-low" system is entirely automatic and, once set, requires no attention from the operator.

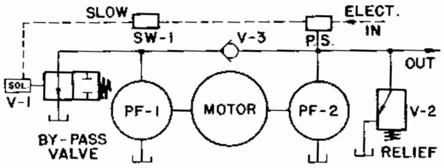
Comparison With Single-Pump System

EXAMPLE: A certain press has a cylinder with 6-inch diameter piston and 10-inch stroke. The press must make the full 10-inch stroke in 6 seconds, and must develop 2000 PSI pressure in the last part of its stroke.

SINGLE PUMP SYSTEM: The volume of oil required to fill the cylinder during the forward stroke is: 28 square inches (area of piston) x 10-inch stroke = 280 cubic inches. Required pumping rate (to close the press in 6 seconds) is 280 ÷ 0.1 (6 seconds is 1/10th minute) = 2800 cubic inches per minute, or, 2800 ÷ 231 (1 U.S. Gallon is 231 cu. ins.) = 12 gallons per minute (GPM). Refer to "Motor HP" table in this manual. Required HP is 16.8 (85% efficient pump).

"HIGH-LOW" SYSTEM: We will assume the same press cylinder as above will move most of the distance in "free travel," and will do the heavy pressing, say in the last ½ inch. Probably less than 500 PSI will be required in both forward and reverse "free travel". The HP table in this manual shows a 5 HP motor to be adequate to drive a pump with up to 15 GPM flow at 500 PSI free travel pressure. The same table shows 5 HP can drive a pump with up to 3 GPM capacity to the full 2000 PSI high pressure required for pressing. Therefore, if the 15 GPM were being delivered by two pumps of 12 and 3 GPM capacity, and if the 12 GPM pump could be automatically unloaded when work resistance builds up the pressure to exceed 500 PSI, then the "high-low," two-pump system could do as much work with a 5 HP motor as the single-pump system can do with 16.8 HP. The slow-down in speed during the pressing operation is more than offset by the increased speed during "free travel" by having 15 instead of 12 GPM.

SLOW Operation for Set-up



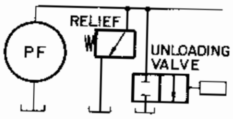
Unloading of the high volume pump, PF-1, is by the pressure switch and solenoid 2-way valve, V-1, when pressure reaches the "set" amount. If the SLOW switch, SW-1, is thrown to the open position, the high volume pump will remain unloaded, and the set-up or testing can be done at slow speed using only the low volume, high pressure pump, PF-2.

Solenoid valve V-1 should be normally open, to be fail-safe electrically. At the start of the cycle it should be energized simultaneously with main control valve.

Pump Unloading

When the hydraulic pump is not needed to develop pressure, such as during loading time, the oil flow should be diverted back to the oil tank under low, or zero, back pressure. Forcing the pump to buck the relief valve unnecessarily, causes excessive pump wear, consumes high horsepower, and most important, it creates heat which accumulates in the oil tank. The unloading circuits on this page may be modified in many ways to exactly suit a particular application.

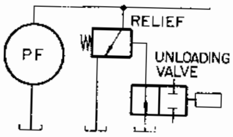
Dump Valve



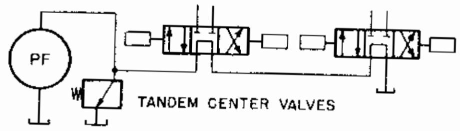
The simplest way to unload a pump is to by-pass the oil directly to tank with a 2-way manual valve. This method is undesirable for general use, as the operator may forget to shift the valve when pressure is not being used. A better way is to use a solenoid valve operated automatically by a limit switch. The switch is actuated by the machine at the end of the cycle, and breaks current to the solenoid valve.

Pilot Control Valve

Essentially the same circuit as above except a small (¼-inch) size unloading valve, either manual or solenoid, can be used to unload even the largest size pump by piloting the system relief valve. The relief valve must be of the pilot-operated type with remote control or "vent" connection. It must be of large enough size to carry the full pump volume at small residual back pressure.



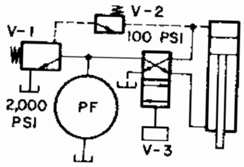
Tandem Center Control Valves



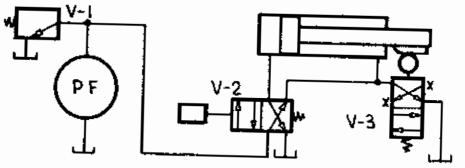
This system works equally well for either manual or solenoid valves. Any number of valves may be connected in tandem, but the pressure losses become quite high when more than 2 or 3 valves are used, and the pump must be capable of developing the extra pressure to supply these losses. CAUTION: Some manual valves must be ordered with external drain connection in order to be used in tandem. Read your valve description carefully before ordering.

Low Pressure Unloading

This circuit is often used for counterbalancing a press ram or cylinder against downward drift due to gravity. The main circuit relief valve, V-1, is set for the required working pressure. Auxiliary relief valve, V-2, works on the vent connection of V-1 to reduce the system pressure to a very low value when the 4-way control valve is shifted for the return stroke. Valve V-2 need never be larger than ¼" size. Note that the discharge port of V-2 is connected to one of the cylinder lines rather than to drain.



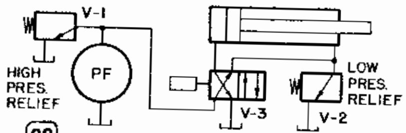
Cam Valve Unloading



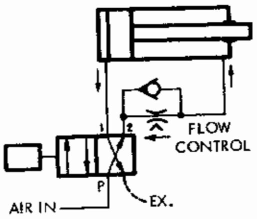
A cam is placed on the cylinder rod, or other convenient part of the moving mechanism. When cylinder reaches desired retracted position and depresses cam actuator, the pump is automatically unloaded to tank. V-3 is shown as a 4-way valve with unused ports plugged.

High Pressure Forward, Low Pressure Return

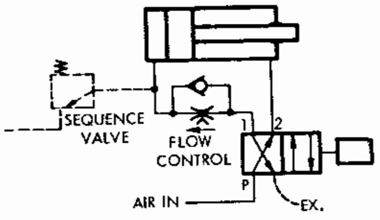
V-1 is set for high pressure forward, and V-2 for low pressure return. Pump continues to buck the low pressure relief after the cylinder bottoms out. This unloads the pump sufficiently for many systems.



Fluid Power Speed Control

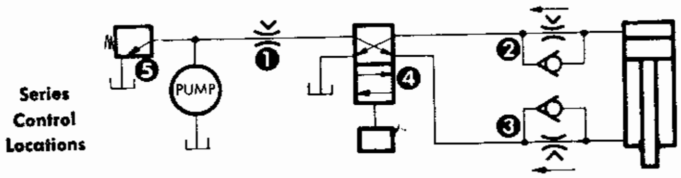


Meter-Out Control



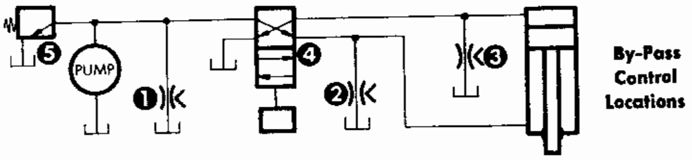
Meter-In Control

When constant volume pumps are used, the speed of an air or hydraulic cylinder is usually accomplished by metering the flow either in series with the cylinder or in shunt with the cylinder. The circuits above show series metering controls connected to meter the outgoing fluid as in the left circuit, or to meter the incoming fluid as in the right circuit. Meter-out circuits are preferred in most cases except where the build-up of pressure behind the control would interfere with proper operation of a sequence valve, a pressure switch or similar device. A flow control valve is any type of throttling valve, with or without a return check, and with or without the pressure compensating feature.



Series-Type Speed Control

Series speed controls may be installed in Positions 2 and 3, connected either as meter-in or meter-out devices, by-passed with check valves as shown. This gives individual adjustment of speed in each direction. Or, a single control at Position 1 will give meter-in control in both directions. For the same speed valve setting, hydraulic cylinder speed will be faster while retracting because of the volume occupied by the cylinder rod.

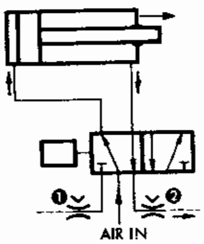


By-Pass Type Speed Control ⬆

Shunt, or by-pass, control is shown above. Unwanted oil is shunted directly back to reservoir, either with individual valves at 2 and 3 for individual control in each direction, or with one valve at 1, for controlling both directions of movement.

Dual Air Exhaust Controls ➡

Most air valves have two exhaust ports, one for each direction of motion of the cylinder. A very simple and effective control of speed is obtained by screwing a needle valve into each of these ports, for individual control of speed in each direction.



Vehicle Drive Calculations

The force necessary to drive a vehicle is composed of the sum of: (1) road resistance, (2) force necessary to negotiate the grade, (3) force needed to accelerate to final velocity in the allowable time, (4) force to overcome air resistance, on fast moving vehicles. Each of these can be calculated or estimated from the formulae on this page, then added together. In selecting an engine, allow enough extra horsepower to make up for losses in the mechanical transmission system, including gear-shift boxes, clutches, differentials, chain or belt drives.

TRAVEL SPEED in MPH (miles per hour) is found by multiplying wheel RPM x wheel circumference.

$$MPH = \frac{RPM \times d}{336}$$

$$RPM = \frac{336 \times MPH}{d}$$

d = Wheel diameter in inches

AXLE TORQUE for driving the vehicle is found by multiplying drawbar pull (or push) times the wheel radius.

$$T = F \times r \qquad \text{or} \qquad F = \frac{T}{r}$$

T = Inch lbs. axle torque
F = Drawbar pull in lbs.
r = Wheel radius in inches

DRAWBAR PULL ta keep the vehicle in steady motion on level ground depends on the road surface. The following values, published by Clark Equipment Co. are typical. Values are lbs. drawbar pull per 1000 lbs. of vehicle weight.

Concrete	10 to 20 lbs.
Asphalt	12 to 22 lbs.
Macadam	15 to 37 lbs.
Cobbles	55 to 85 lbs.
Snow	25 to 37 lbs.
Dirt	25 to 37 lbs.
Mud	37 to 150 lbs.
Sand	60 to 300 lbs.

HORSEPOWER required on vehicle wheels is torque x RPM.

$$HP = \frac{T \times RPM}{63024}$$

T = Inch lbs torque

NOTE: Additional horsepower is required at the engine to overcome transmission system losses.

CONVERSION formula between torque and horsepower.

$$T = \frac{HP \times 63024}{RPM}$$

MOMENTUM of a vehicle is equivalent to that constant force which would bring it to rest in one second by resisting its movement.

$$\text{Momentum} = \frac{\text{Weight} \times V}{g}$$

Weight is in lbs.
V = Velocity in feet per second
g = Gravity acceleration = 32.16

ACCELERATION of a vehicle is expressed in this formula involving weight, accelerating force, and time.

$$F = \frac{V \times W}{g \times T}$$

F = Accelerating force in lbs.
V = Final velocity- ft. per sec.
W = Vehicle weight in lbs.
g = Gravity acceleration = 32.16
T = Time in seconds that force acts

Please note that the use of the symbol "g" in this equation is to convert weight into mass.

GRADE. In mobile work, grade is usually expressed in percentage rather than by angular measurement. For example, a 10% grade has a rise of 10 feet in a distance of 100 feet, etc.

GRADE RESISTANCE is the drawbar pull necessary to keep the vehicle in constant motion up the grade. This is in addition to the drawbar pull to overcome road resistance, as expressed in another formula.

$$F = GR \times W$$

F = Drawbar pull in lbs.
GR = Per Cent grade (i. e. 10% = .10)
W = Vehicle gross weight in lbs.

AIR RESISTANCE will be important only on fast moving vehicles. (over 30 MPH).

$$F = FA \times (MPH)^2$$

F = Additional force necessary to overcome air resistance
FA = Frontal area of vehicle in square feet
MPH = Vehicle speed in miles per hour

AXLES and drive shafts must have a diameter large enough to transmit the torque without excessive deflection. The angle of deflection for a solid, round axle, may be calculated from this formula:

$$A = \frac{583.6 \times T \times L}{D^4 \times E}$$

A = Angle of deflection in degrees
T = Applied torque in inch lbs.
L = Shaft length in inches
E = Modulus of elasity (12,000,000 for steel)
D = Shaft diameter in inches

Some authorities say that a steel shaft should be limited to an angular deflection of .08 degrees per foot of length, to avoid failure.

Overheating in Hydraulic Systems

HEAT GENERATION

Heat is generated in a hydraulic system whenever oil dumps from a higher to a lower pressure without doing mechanical work. Typical examples are: oil bucking a relief valve; pressure losses from oil flowing through piping, valving, etc. At the point where mechanical work is being done, such as in the cylinder, fluid motor, etc., most of the energy is going into work, and very little heat is being generated.

When designing a hydraulic system, an estimate must be made of the heat which will be generated. An oil reservoir of suitable size must be used to dissipate this heat, or some type of oil cooler must be added to the system. Oil temperature should be held to 120°F. for best results, and should never be allowed to exceed 150°F. At high temperatures, oxidation of the oil is accelerated, shortening its useful life, by producing acids and sludge which corrode metal parts, clog valve orifices, and cause rapid wear of moving parts.

Oil reservoir temperature should be checked occasionally, since overheating tends to get worse as the system ages. Oil leaking past cylinder pistons, and slippage in the pump and valves, produces a quite appreciable amount of heat which accumulates in the oil tank. And of course the danger of overheating is much greater in hot weather.

HEAT GENERATION FORMULAE

Heat generation across an orifice or pressure relief valve may be easily calculated if the pressure drop across the device and the oil flow through it is known or can be measured. Use the formulae:

$$HP = PSI \times GPM \div 1714$$

$$\text{or, } BTU/hr = 1\tfrac{1}{2} \times PSI \times GPM$$

in which PSI is the net pressure across the orifice or valve in pounds/square inch, and GPM is the oil flow through the orifice or valve in gallons/minute.

Heat may be converted into other units by means of conversion factors:

$$1\ HP = 2545\ BTU/hr = 42.4\ BTU/min.$$

$$= 33{,}000\ ft.lbs./min. = 746\ watts.$$

$$1\ BTU/hr = .0167\ BTU/min. = .00039\ HP.$$

Example: 12 GPM by-passing a relief valve at a pressure drop of 500 PSI generates 3½ HP of heat, most of which is carried by the oil back into the tank.

NOTE: Heat is generated only when the energy is not going into mechanical power.

ESTIMATING HEAT BUILD-UP

In most circuits the prime heat generator is the pressure relief valve. Usually this valve is in action for only a portion of the cycle. Therefore, it is necessary to find the maximum rate of heat generation by using the formulae above, then arrive at an AVERAGE for the entire cycle.

Example: In a certain system the relief valve is generating heat at the rate of 3 HP while in action. However, the oil is bucking the relief valve on an average of about ⅓ of the time. This average includes idle time between cycles plus regular time, and is over a 1-hour period. The AVERAGE rate of heat generation is, then, 1 HP.

Another major cause of heat generation is a pressure compensated flow control valve used to obtain variable speed of a cylinder or fluid motor. If this valve is piped in series with the load, then a portion of the oil is always forced to by-pass through the system relief valve. When the flow control is throttled down to obtain slow speed operation, a major portion of the pump volume is usually generating heat in the relief valve. If the flow control can be used in a "by-pass" circuit to shunt off unwanted oil, heat generation can sometimes be reduced.

Pressure reducing valves should also be checked if they are passing large volume flows or working at a high pressure drop from inlet to outlet.

In addition to the major heat generating points already listed, assume that an additional 25% of the pump horsepower will go into heat. This will usually take care of the many hard-to-figure points of heat generation such as fluid friction in pipes and miscellaneous valves, mechanical friction and slippage in cylinders, fluid motors, or pumps. It is good practice to provide connections, usually in the main oil return line or in the relief valve return line, for installing a heat exchanger.

COOLING CAPACITY OF STEEL RESERVOIRS

After estimating the HP or BTU heat generation in your hydraulic system, the next step is to see whether the heat can be radiated from the oil tank, or whether an oil cooler is required.

For average work, assume heat radiation from the sides and top of the reservoir. The surface area of external plumbing may also be counted as radiating surface. Do not figure the bottom of the reservoir unless it is exposed to free air circulation. Cooling capacity of the reservoir will increase in proportion to the square footage radiating surface, and also in proportion to the difference between oil temperature and surrounding air temperature. For steel oil tanks the following formula will give

approximately correct results:

HP (heat generation) = 0.001 x TD x A

A = square footage of radiating surface.
TD = temperature difference between oil and surrounding air.
HP = cooling capacity expressed in horsepower which is usually the most convenient working unit.

There should be a reasonable degree of free air circulation around the tank. A forced blast of air directed on the side of the tank can increase the heat radiating capacity as much as 50% or even more.

For converting HP to other heat units, refer to the conversion formulae on the opposite page.

COOLING CAPACITY OF STANDARD TANKS

This table shows the aproximate heat radiation that can be expected of standard steel oil reservoirs. Figures are approximate, as the actual shape of the tank will influence the radiation. Values of radiation are figured very conservatively, being based on an ambient temperature of 100°F, with a maximum oil temperature of 150°F. Radiating capacity will increase at lower ambients. Oftentimes a considerable amount of heat is radiated from plumbing, valves, and cylinders.

Total Gallon Volume	Sq. Ft. Radiating Surface	Heat Radation, BTU/hr.	Heat Radiation HP
6½	4½	575	.23
12	6¾	850	.34
17	8¼	1000	.40
33	13¼	1700	.67

Total Gallon Volume	Sq. Ft. Radiating Surface	Heat Radiation, BTU/hr.	Heat Radiation HP
80	24	3000	1.18
120	30	4000	1.60
250	50	6000	2.70
500	80	10,200	4.00

TO REDUCE HEAT BUILD-UP

1. Unload the pump, if possible, when pressure is not required. Several methods of doing this are diagrammed and explained elsewhere in this manual.

2. On presses where a high static pressure must be held for a long time, an accumulator may be used to hold pressure while the pump is unloaded.

3. On molding, bonding, or laminating presses, use an air-driven pressure intensifier to maintain pressure without heat generation.

4. In systems where heat buildup may be a problem, use a generous reservoir size, with a large surface area.

5. Pressure compensated flow control valves, if used, should be connected in a "by-pass" arrangement if this is possible.

6. Set the main pressure relief valve for the least amount of pressure which will do the work.

7. Locate oil reservoirs out in the open. Putting the tank in a small enclosed compartment or console surrounded with a dead air space will not permit maximum heat radiation. Shading the oil tank from the direct rays of the sun may help in marginal cases.

Compressibility of Water and Oil

These values for compressibility will be found useful in calculating the amount of fluid that must be released from a cylinder in order to decompress to a desired lower pressure. Or to calculate the amount of additional fluid which must be pumped into a pressure vessel which has been pre-filled completely with fluid, to bring the test pressure up to a certain level.

PSI	Oil	Water	PSI	Oil	Water	PSI	Oil	Water
1000	0.56%	0.33%	6000	2.67%	1.83%	11,000	4.44%	3.19%
2000	1.04%	0.67%	7000	3.04%	2.13%	12,000	4.80%	3.48%
3000	1.47%	0.94%	8000	3.40%	2.38%	13,000	5.13%	3.77%
4000	1.89%	1.25%	9000	3.77%	3.68%	14,000	5.46%	4.06%
5000	2.30%	1.54%	10,000	4.11%	2.90%	15,000	5.79%	4.35%

RULE OF THUMB. For hydraulic oil, average ½% reduction in volume per 1000 PSI. For water, average ⅓% reduction in volume per 1000 PSI.

Motor Horsepower . . .

TO DRIVE A HYDRAULIC PUMP

The chart is based on a pump efficiency of 85%. and is calculated from the formula: HP = GPM × PSI ÷ (1714 × .85). Above 2000 PSI, multiply chart values by a suitable factor. Example: to get HP at 3000 PSI multiply chart values in 1500 PSI column by 2, etc.

GPM	100 PSI	200 PSI	250 PSI	300 PSI	400 PSI	500 PSI	750 PSI	1000 PSI	1250 PSI	1500 PSI	2000 PSI
½	.04	.07	.09	.11	.14	.18	.26	.35	.44	.53	.70
1	.07	.14	.18	.21	.28	.35	.52	.70	.88	1.05	1.40
1½	.10	.21	.26	.31	.41	.52	.77	1.03	1.29	1.55	2.06
2	.14	.28	.35	.42	.56	.70	1.04	1.40	1.76	2.10	2.80
2½	.17	.34	.43	.51	.69	.86	1.29	1.72	2.15	2.58	3.44
3	.21	.42	.53	.63	.84	1.05	1.56	2.10	2.64	3.15	4.20
3½	.24	.48	.60	.72	.96	1.20	1.80	2.40	3.00	3.60	4.80
4	.28	.56	.70	.84	1.12	1.40	2.08	2.80	3.52	4.20	5.60
5	.35	.70	.88	1.05	1.40	1.75	2.60	3.50	4.40	5.25	7.00
6	.42	.84	1.05	1.26	1.68	2.10	3.12	4.20	5.28	6.30	8.40
7	.49	.98	1.23	1.47	1.96	2.45	3.64	4.90	6.16	7.35	9.80
8	.56	1.12	1.40	1.68	2.24	2.80	4.16	5.60	7.04	8.40	11.2
9	.62	1.24	1.55	1.86	2.48	3.10	4.65	6.18	7.73	9.28	12.4
10	.70	1.40	1.75	2.10	2.80	3.50	5.20	7.00	8.80	10.5	14.0
11	.77	1.54	1.93	2.31	3.08	3.85	5.72	7.70	9.68	11.5	15.4
12	.84	1.68	2.10	2.52	3.36	4.20	6.24	8.40	10.5	12.6	16.8
13	.89	1.78	2.23	2.67	3.56	4.45	6.68	8.92	11.2	13.4	17.8
14	.96	1.92	2.40	2.88	3.84	4.80	7.20	9.60	12.0	14.4	19.2
15	1.05	2.10	2.63	3.15	4.20	5.25	7.80	10.5	13.2	15.7	21.0
16	1.10	2.20	2.75	3.30	4.40	5.50	8.25	11.0	13.8	16.5	22.0
17	1.17	2.34	2.93	3.51	4.68	5.85	8.78	11.7	14.6	17.6	23.4
18	1.26	2.52	3.15	3.78	5.04	6.30	9.35	12.6	15.8	18.9	25.2
19	1.30	2.60	3.25	3.90	5.20	6.50	9.75	13.0	16.3	19.5	26.0
20	1.40	2.80	3.50	4.20	5.60	7.00	10.4	14.0	17.6	21.0	28.0
25	1.75	3.50	4.38	5.25	7.00	8.75	13.1	17.5	21.9	26.2	35.0
30	2.10	4.20	5.25	6.30	8.40	10.5	15.6	21.0	26.4	31.5	42.0
35	2.45	4.90	6.13	7.35	9.80	12.2	18.4	24.5	30.6	36.7	49.0
40	2.80	5.60	7.00	8.40	11.2	14.0	20.8	28.0	35.2	42.0	56.0
45	3.15	6.30	7.87	9.45	12.6	15.8	23.6	31.5	39.4	47.3	63.0
50	3.50	7.00	8.75	10.5	14.0	17.5	26.0	35.0	44.0	52.5	70.0
55	3.85	7.70	9.63	11.6	15.4	19.3	28.6	38.5	48.4	57.8	77.0
60	4.20	8.40	10.5	12.6	16.8	21.0	31.2	42.0	52.8	63.0	84.0
65	4.55	9.10	11.4	13.6	18.2	22.8	33.8	45.5	57.2	68.2	91.0

Overloading Electric Motors . . .

The most common electric motor used for driving hydraulic pumps is the 3-phase, squirrel-cage, Design B motor. It has a service factor of 10%, which means it can be operated continuously at 10% above its nameplate rating at normal room temperatures.

In addition, this motor will deliver up to 300% of its nameplate horsepower for short periods before stalling. From a standstill, it can produce up to 150% rated torque for starting. However, it is not good practice to overload a motor to this extent because the line current tends to approach infinity on heavy overloads.

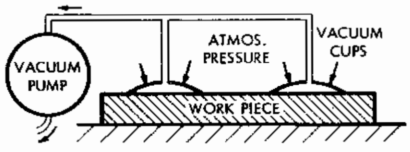
It is good practice to overload the motor to some extent if the hydraulic load is intermittent, as long as the AVERAGE hydraulic loading does not exceed nameplate rating, for the entire cycle.

It is best to limit the overloading to about 125 to 140% with the motor running, and to start it under no load by designing the hydraulic circuit to have the pump unloaded while the electric motor is being started.

EXAMPLE: Find the required motor horsepower to operate a 16 GPM pump on the following press cycle: 10 seconds to traverse forward at 300 PSI pressure, 10 seconds to press at 1000 PSI, and 10 seconds to return traverse at 400 PSI.

SOLUTION: From the chart, 3.3 HP is required on forward traverse, 11.0 HP on pressing, and 4.4 HP for return traverse. A 10 HP motor, if used, would be overloaded about 10% on pressing, but would be operating at less than half power during two thirds of the cycle. Thus, the average loading would be considerably less than 100%. This is a good design if economy of first cost is important.

Vacuum Applications



Vacuum pads are used for handling concrete slabs, metal or paper sheets, cloth, leather, etc. Commonly used in printing and in packaging machines to grip or transfer paper, cardboard, or cellophane.

The figures in the body of this chart are lift capacities in pounds for a straight vertical lift. Vacuum pads can easily slip sideways when worked on slick surfaces; therefore, use care when gripping a load which has a tendency to shear away from the pad.

Lifting Force of Vacuum Pads

	Circular Pad				Square Pad				
Pad Dia, Ins.	Pad Area, Sq.Ins.	Lift at 14" Vac.	Lift at 20" Vac.	Lift at 26" Vac.	Pad Side Dim.	Pad Area, Sq.Ins.	Lift at 14" Vac.	Lift at 20" Vac.	Lift at 26" Vac.
1"	.785	5.5	7.8	10	1"	1.0	7.0	10	13
2	3.14	22	31	40	2	4.0	28	40	52
3	7.07	50	71	92	3	9.0	63	90	117
4	12.57	88	126	160	4	16.0	112	160	208
5	19.63	137	196	255	5	25.0	175	250	325
6	28.27	200	283	370	6	36.0	252	360	470
8	50.27	350	503	650	8	64.0	448	640	830
10	78.50	550	785	1000	10	100	700	1000	1300
12	113.1	800	1131	1460	12	144	1008	1440	1875

Conversion of Inches Mercury to PSI

"Hg	30	28	26	24	22	20	18	16	14	12	10	8
PSI	14.73	13.74	12.77	11.78	10.80	9.83	8.84	7.86	6.88	5.90	4.91	3.93

For "rule-of-thumb" figuring 1 PSI equals 2" Hg.

Atmospheric Pressure at Various Altitudes

Since "vacuum force" is developed by weight of the atmosphere above, an allowance may have to be made in a vacuum system which is to be operated more than a few thousand feet above sea level.

Altitude	1000	2000	4000	6000	8000	10,000	12,000	14,000	16,000
"Hg Pres.	29.0	27.8	25.8	24.0	22.2	20.5	19.0	17.5	16.2
PSI Pres.	14.2	13.6	12.7	11.8	10.9	10.1	9.3	8.6	7.8

Evacuation Time For Large Tank

This chart is set up for attaining a 20" Hg. vacuum, which is the value used in many industrial applications using large vacuum storage tanks. For higher vacuums the time would be greater than shown in the chart.

To use the chart it is necessary to know the free-running air displacement of your vacuum pump, and the volume of the tank to be evacuated. Displacement information is obtained from the manufacturers catalog, or in some cases can be calculated from the physical dimensions of the pump and the rotational speed.

Tank Vol, Gals. →	20	30	50	75	100	150	200	300	500
Tank Vol, Cu.Ft. →	2.7	4	6.7	10	13	20	27	40	67
Pump CFM, Running Free			Time in Minutes						
2 CFM	2.34	3.50	5.85	8.77	11.7	17.5	23.4	35.0	58.5
3	1.56	2.34	3.90	5.86	7.80	11.7	15.6	23.4	39.0
4	1.17	1.75	2.93	4.39	5.85	8.77	11.7	17.5	29.3
6	.78	1.17	1.95	2.93	3.90	5.83	7.80	11.7	19.5
10	.47	.70	1.17	1.77	2.34	3.51	4.68	7.02	11.7
15	.31	.47	.78	1.17	1.50	2.34	3.12	4.68	7.80
20	.23	.35	.58	.88	1.17	1.75	2.34	3.50	5.85
30	.16	.23	.39	.59	.78	1.17	1.56	2.34	3.90
50	....	.14	.23	.35	.47	.70	.94	1.40	2.34

Hydraulic Pipe Table

Working Pressure in PSI

Size	O.D.	I.D.	Area	Strength S.F.* = 6	Strength S.F.* = 8	Strength S.F.* = 10
STANDARD PIPE						
⅛	.405	.269	.06	2820	1705	1364
¼	.540	.364	.10	2172	1629	1303
⅜	.675	.493	.19	1797	1348	1078
½	.84	.622	.30	1731	1298	1038
¾	1.05	.824	.53	1434	1076	860
1	1.32	1.049	.86	1348	1011	808
1¼	1.66	1.380	1.49	1124	843	674
1½	1.90	1.610	2.03	1017	763	610
EXTRA HEAVY PIPE						
⅛	.405	.215	.036	3977	2983	2386
¼	.540	.302	.071	2937	2203	1762
⅜	.675	.423	.141	2488	1866	1492
½	.84	.546	.231	2333	1750	1400
¾	1.05	.742	.425	1954	1716	1172
1	1.32	.957	.710	1814	1611	1088
1¼	1.66	1.278	1.271	1533	1150	920
1½	1.90	1.500	1.753	1403	1052	841
DOUBLE EXTRA HEAVY PIPE						
½	.84	.252	.047	4666	3500	2800
¾	1.05	.434	.140	3910	2933	2346
1	1.32	.599	.271	3629	2722	2177
1¼	1.66	.896	.615	3068	2301	1840
1½	1.90	1.100	.930	2807	2105	1684

*S.F. = Safety Factor

Safe Working Pressure

Seamless Steel Tubing

This chart is based on steel tubing with tensile strength of 75,000 PSI, and includes a safety factor of 5. Order "hydraulic grade" tubing which is annealed to a soft conditon after drawing. This yields a perfect flare with much less effort. For higher pressures, hard-drawn mechanical or stainless steel tubing may be used, giving pressure ratings higher than shown in the chart.

WALL →	.020"	.025"	.028"	.032"	.035"	.042"	.049"	.058"	.065"	.072"	.083"
OD ↓											
⅛"	4,800	6,000	6,700	7,680	8,430	9,920	11,750				
3/16"	3,200	4,000	4,470	5,100	5,610	6,750	7,850				
¼"	2,400	2,980	3,350	3,830	4,200	5,040	5,890	7,000	7,850		
5/16"	2,000	2,410	2,690	3,080	3,370	4,040	4,710	5,610	6,290	6,950	8,010
⅜"	1,600	1,990	2,240	2,560	2,800	3,360	3,920	4,660	5,240	5,790	6,660
7/16"	1,370	1,715	1,920	2,190	2,400	2,880	3,360	4,000	4,500	4,960	5,720
½"	1,200	1,490	1,675	1,920	2,100	2,520	2,940	3,500	3,925	4,340	5,000
9/16"	1,065	1,330	1,490	1,705	1,870	2,240	2,620	3,110	3,490	3,860	4,450
⅝"	1,000	1,200	1,340	1,535	1,680	2,020	2,350	2,800	3,140	3,470	4,000
11/16"	870	1,090	1,220	1,395	1,530	1,830	2,135	2,530	2,835	3,140	3,620
¾"	800	1,000	1,120	1,280	1,400	1,640	1,960	2,320	2,600	2,880	3,320
⅞"	685	860	960	1,090	1,200	1,440	1,680	1,985	2,225	2,465	2,845
1"	600	750	840	960	1,050	1,260	1,470	1,740	1,950	2,160	2,490

Table of Equivalents

Units in the first column can be converted to units of the second column by multiplying them by the factor in the third column. Example: Given 5 atmospheres, how many feet of water is this? Solution: 5 × 33.9 = 169.5 feet of water.

Or, units in the second column can be converted to equivalent units of the first column by dividing by the factor in the third column. Example: Given 100 feet of water, how many atmospheres is this? Solution: 100 ÷ 33.9 = 3.69 atmospheres.

TO CONVERT ➤	INTO ➤	MULTIPLY BY
INTO ◄	**TO CONVERT ➤**	**DIVIDE BY**
Atmospheres	Feet of Water	33.9
Atmospheres	Inches of Mercury (Hg.)	29.92
Atmospheres	PSI (Lbs./Sq. In.)	14.7
BTU	Foot Lbs.	778.3
BTU/Hr.	Watts	0.2931
BTU/Min.	Horsepower	0.02356
Centigrade	Fahrenheit	°C x 1.8 + 32
Centimeters	Inches	0.3937
Cubic Centimeters	Gallons (U. S. Liq.)	0.0002642
Cubic Centimeters	Liters	0.001
Cubic Feet	Cubic Inches	1,728
Cubic Feet	Gallons (U. S. Liq.)	7.48052
Cubic Inches	Cubic Feet	0.0005787
Cubic Inches	Gallons (U. S. Liq.)	0.004329
Days	Seconds	86,400
Degrees (Angle)	Radians	0.01745
Feet	Meters	0.3048
Feet	Miles	0.0001894
Feet of Water	Atmospheres	0.0295
Feet of Water	Inches of Mercury (Hg.)	0.8826
Feet of Water	PSI (Lbs./Sq. In.)	0.4335
Feet/Min.	Miles/Hr.	0.01136
Feet/Sec.	Miles/Hr.	0.6818
Foot Lbs.	BTU	0.001286
Foot Lbs./Min.	Horsepower	0.0000303
Foot Lbs./Sec.	Horsepower	0.001818
Gallons (U. S. Liq.)	Cubic Feet	0.1337
Gallons (U. S. Liq.)	Cubic Inches	231
Gallons of Water	Pounds of Water	8.3453
Horsepower	BTU/Min.	42.44
Horsepower	Foot Lbs./Min.	33,000
Horsepower	Foot Lbs./Sec.	550
Horsepower	Watts	745.7
Hours	Days	0.04167
Hours	Weeks	0.005952
Inches	Centimeters	2.54
Inches of Mercury (Hg.)	Atmospheres	0.03342
Inches of Mercury (Hg.)	Feet of Water	1.133
Inches of Mercury (Hg.)	PSI (Lbs./Sq. In.)	0.4912
Inches of Water	PSI (Lbs./Sq. In.)	0.03613
Liters	Cubic Centimeters	1,000
Liters	Gallons (U. S. Liq.)	0.2642
Micron	Inches	0.00004
Miles (Statute)	Feet	5,280
Miles/Hr. (MPH)	Feet/Min.	88
Miles/Hr. (MPH)	Feet/Sec.	1.467
Ounces (Weight)	Pounds	0.0625
Ounces (Fluid)	Cubic Inches	1.805
Pints (Fluid)	Quarts (Fluid)	0.5
Pounds	Grains	7,000
Pounds	Grams	453.5924
Pounds	Ounces	16
PSI (Lbs./Sq. In.)	Atmospheres	0.06804
PSI (Lbs./Sq. In.)	Feet of Water	2.307
PSI (Lbs./Sq. In.)	Inches of Mercury (Hg.)	2.036
Quarts	Gallons	0.25
Square Feet	Square Inches	144
Temperature (°F) −32	Temperature (°C)	0.5555
Tons (Short)	Pounds	2,000
Watts	Horsepower	0.001341

Circumference & Area of Circles

For circle diameters not shown in this table, use these formulae:
Circumference = πD, or, $2\pi r$
Area = $\pi D^2 \div 4$, or, πr^2

Diam.	Circum.	Area
1/64	.04909	.00019
1/32	.09818	.00077
3/64	.14726	.00173
1/16	.19635	.00307
3/32	.29452	.00690
1/8	.39270	.01227
5/32	.49087	.01917
3/16	.58905	.02761
7/32	.68722	.03758
1/4	.78540	.04909
9/32	.88357	.06213
5/16	.98175	.07670
11/32	1.0799	.09281
3/8	1.1781	.11045
13/32	1.2763	.12962
7/16	1.3744	.15033
15/32	1.4726	.17257
1/2	1.5708	.19635
17/32	1.6690	.22166
9/16	1.7671	.24850
19/32	1.8653	.27688
5/8	1.9635	.30680
21/32	2.0617	.33824
11/16	2.1598	.37122
23/32	2.2580	.40574
3/4	2.3562	.44179
25/32	2.4544	.47937
13/16	2.5525	.51849
27/32	2.6507	.55914
7/8	2.7489	.60132
29/32	2.8471	.64504
15/16	2.9452	.69029
31/32	3.0434	.73708
1	3.1416	.7854
1/16	3.3379	.8866
1/8	3.5343	.9940
3/16	3.7306	1.1075
1/4	3.9270	1.2272
5/16	4.1233	1.3530
3/8	4.3197	1.4849
7/16	4.5160	1.6230
1/2	4.7124	1.7671
9/16	4.9087	1.9175
5/8	5.1051	2.0739
11/16	5.3014	2.2365
3/4	5.4978	2.4053
13/16	5.6941	2.5802
7/8	5.8905	2.7612
15/16	6.0868	2.9483
2	6.2832	3.1416
1/16	6.4795	3.3410
1/8	6.6759	3.5466
3/16	6.8722	3.7583
1/4	7.0686	3.9761
5/16	7.2649	4.2000
3/8	7.4613	4.4301
7/16	7.6576	4.6664
1/2	7.8540	4.9087
9/16	8.0503	5.1572
5/8	8.2467	5.4119
11/16	8.4430	5.6727
3/4	8.6394	5.9396
13/16	8.8357	6.2126
7/8	9.0321	6.4918
15/16	9.2284	6.7771
3	9.4248	7.0686
1/16	9.6211	7.3662
1/8	9.8175	7.6699
3/16	10.014	7.9798

Diam.	Circum.	Area
1/4	[illegible]	[illegible]
5/16	[illegible]	[illegible]
3/8	[illegible]	[illegible]
7/16	[illegible]	[illegible]
1/2	[illegible]	[illegible]
9/16	[illegible]	[illegible]
5/8	[illegible]	[illegible]
11/16	[illegible]	[illegible]
3/4	[illegible]	[illegible]
13/16	[illegible]	[illegible]
7/8	[illegible]	[illegible]
15/16	[illegible]	[illegible]
4	[illegible]	[illegible].566
1/16	[illegible]	[illegible]
1/8	[illegible]	[illegible].364
3/16	[illegible]	[illegible]
1/4	[illegible]	[illegible].156
5/16	[illegible]	[illegible].507
3/8	[illegible]	[illegible].033
7/16	[illegible]	[illegible].466
1/2	[illegible]	[illegible].904
9/16	[illegible]	[illegible].349
5/8	[illegible]	[illegible].800
11/16	[illegible]	[illegible].257
3/4	[illegible]	[illegible].728
13/16	[illegible]	[illegible].190
7/8	[illegible]	[illegible].665
15/16	[illegible]	[illegible].147
5	[illegible]	[illegible].635
1/16	[illegible]	[illegible].129
1/8	[illegible]	[illegible].629
3/16	[illegible]	[illegible].135
1/4	[illegible]	[illegible].648
5/16	[illegible]	22.166
3/8	[illegible]	[illegible].691
7/16	[illegible]	[illegible].221
1/2	[illegible]	[illegible].758
9/16	[illegible]	[illegible].301
5/8	[illegible]	[illegible].850
11/16	[illegible]	[illegible].406
3/4	[illegible]	[illegible].967
13/16	[illegible]	26.535
7/8	[illegible]	[illegible].109
15/16	[illegible]	[illegible].688
6	[illegible]	28.274
1/8	[illegible]	29.465
1/4	[illegible]	30.680
3/8	[illegible]	31.919
1/2	[illegible]	33.183
5/8	[illegible]	34.472
3/4	[illegible]	35.785
7/8	[illegible]	37.122
7	[illegible]	38.485
1/8	22.384	39.871
1/4	22.776	41.282
3/8	23.159	42.718
1/2	23.562	44.179
5/8	23.955	45.664
3/4	24.347	47.173
7/8	24.740	48.707
8	25.133	50.265
1/8	25.525	51.849
1/4	25.918	53.456
3/8	26.311	55.088
1/2	26.704	56.745
5/8	27.096	58.426
3/4	27.489	60.132
7/8	27.882	61.862
9	28.274	63.617

Diam.	Circum.	Area
1/8	28.667	65.397
1/4	29.060	67.201
3/8	29.452	69.029
1/2	29.845	70.882
5/8	30.238	72.760
3/4	30.631	74.662
7/8	31.023	76.589
10	31.416	78.540
1/4	32.201	82.516
1/2	32.987	86.590
3/4	33.772	90.763
11	34.558	95.033
1/4	35.343	99.402
1/2	36.128	103.87
3/4	36.914	108.43
12	37.699	113.10
1/4	38.485	117.86
1/2	39.270	122.72
3/4	40.055	127.68
13	40.841	132.73
1/4	41.626	137.89
1/2	42.412	143.14
3/4	43.197	148.49
14	43.982	153.94
1/4	44.768	159.48
1/2	45.553	165.13
3/4	46.338	170.87
15	47.124	176.71
1/4	47.909	182.65
1/2	48.695	188.69
3/4	49.480	194.83
16	50.265	201.06
1/4	51.051	207.39
1/2	51.836	213.82
3/4	56.622	220.35
17	53.407	226.98
1/4	54.192	233.71
1/2	54.978	240.53
3/4	55.763	247.45
18	56.549	254.47
1/4	57.334	261.59
1/2	58.119	268.80
3/4	58.905	276.12
19	59.690	283.53
1/4	60.476	291.04
1/2	61.261	298.65
3/4	62.046	306.35
20	62.832	314.16
1/2	64.403	330.06
21	65.973	346.36
1/2	67.544	363.05
22	69.115	380.13
1/2	70.686	397.61
23	72.257	415.48
1/2	73.827	433.74
24	75.398	452.39
1/2	76.969	471.44
25	78.540	490.87
1/2	80.111	510.71
26	81.681	530.93
1/2	83.252	551.55
27	84.823	572.56
1/2	86.394	593.96
28	87.965	615.75
1/2	89.535	637.94
29	91.106	660.52
1/2	92.677	683.49
30	94.248	706.86
1/2	95.819	730.62

ASA Graphic Symbols

For Use On Fluid Power Drawings

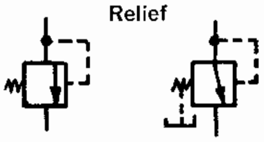
Popular symbols are shown. A brochure with complete listings may be obtained for $2.00 per copy from the Fluid Power Society, P. O. Box 49, Thiensville, Wis.

On this page, new symbols are shown on the left and the old symbols which they replaced are shown on the right.

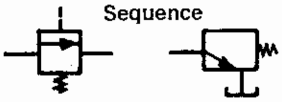
PRESSURE CONTROL VALVES

New Symbol **Old Symbol**

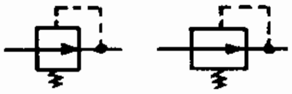
Relief



Sequence



Reducing



FLOW CONTROL VALVES

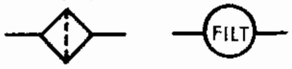
Fixed

Variable

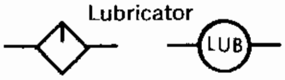
Variable — Pressure Comp

FLUID CONDITIONERS

Filter or Strainer



Lubricator

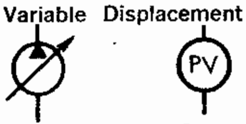


FLUID PUMPS

New Symbol **Old Symbol**

Fixed Displacement

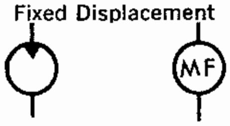
Variable Displacement



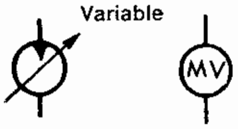
Variable - Pres. Comp.

FLUID MOTORS

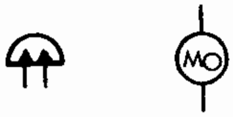
Fixed Displacement



Variable



FLUID OSCILLATORS



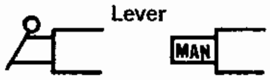
ELECTRIC MOTORS

VALVE ACTUATORS

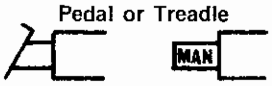
New Symbol **Old Symbol** **New Symbol** **Old Symbol**

Manual

Lever

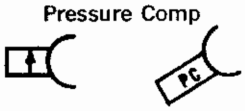


Pedal or Treadle



Mechanical

Pressure Comp



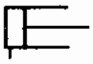
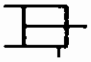
Solenoid

Pilot — Spring Centered

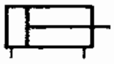
Solenoid and Pilot

THE FOLLOWING SYMBOLS ARE UNCHANGED

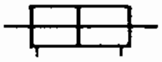
CYLINDERS

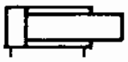
SINGLE-ACTING
Push Pull



Single Rod



DOUBLE-ACTING
Double Rod



Large Rod

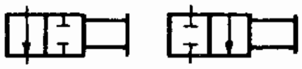
MISCELLANEOUS VALVES

STANDARD
CHECK

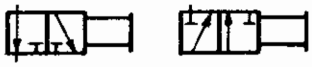

PILOT-OPERATED CHECKS
Pilot-to-open Pilot-to-close

2-WAY
MANUAL

DIRECTIONAL CONTROL VALVES



2-WAY VALVES



3-WAY VALVES

2-Position

4-WAY VALVES
Closed Center

Open Center

Tandem Center

4-WAY VALVES
Float Center

Cylinder Vent

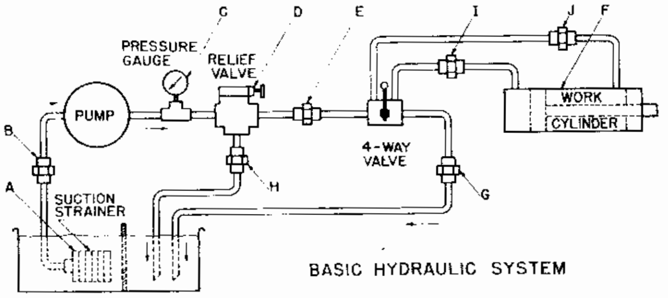
Troubleshooting

YOUR HYDRAULIC SYSTEM

Most of the failures in a hydraulic system have essentially the same result — a gradual or sudden loss of high pressure, with consequent loss of power in the work cylinder, causing it to stall out under light loads, or to move more slowly than normal. Any one of the system components may be at fault, and by following the step-by-step instructions on these 2 pages, the trouble can usually be pinpointed in a short time.

These instructions are intended for spotting troubles in a system which previously has been working properly. They are not intended for diagnosing a new system which has not been properly designed.

A reasonably accurate pressure gauge must be in the circuit at position C, and the correct pump working pressure and GPM flow must be known at least approximately.



BASIC HYDRAULIC SYSTEM

Step 1. Pump Suction Strainer . . .

This is the No. 1 source of trouble and should always be checked first, particularly if the system has developed unusual noise. If the strainer is not located in the suction line it will be found immersed in the tank, as at A. Remove and clean it whether it looks dirty or not. Use a compressed air hose blowing from the inside out. If it is damaged or does not clean up well, replace it with a new strainer.

When replacing the strainer, inspect all joints in the plumbing for possible air leaks, particularly at union joints. There must be no air leaks in the suction line. Check the oil level in the reservoir; it should be at least 2 inches above the top of the strainer. If not, a vortex may form over the intake which may allow the top of the strainer to be exposed to air intake when the pump is started up.

Step 2. Pump and Relief Valve . . .

If cleaning the pump suction strainer does not correct the trouble, proceed to test as follows:

Disconnect the plumbing at Point E so that only the pump, relief valve, and pressure gauge are in the pressure circuit. Plug both exposed ends of the piping. Start the pump and test for pressure by screwing down the knob of the relief valve. If full pressure can now be developed, obviously the pump and relief valve are OK. If full pressure cannot be developed in this test, continue with Step 3.

Step 3. Pump or Relief Valve? . . .

If high pressure cannot be developed in Step 2, proceed as follows: Disconnect the discharge line of the relief valve at Point H and, with the pump running and with the relief valve screwed down tight, see whether a full stream of oil is being discharged. A good way to observe this is to connect a hydraulic hose to the relief valve connection. Hold the open end of the hose over the filler hole in the oil reservoir, where the rate of discharge can easily be observed. If possible, measure the oil flow by letting it run into an open container for, say 15 seconds, then multiply the number of gallons collected by 4 to get the pump flow in gallons per minute (GPM). If the discharge is approximately equal to the pump rating, the relief valve is at fault and should be tested as per Step 5. If there is almost no discharge from the relief valve, and high pressure still cannot be developed, then the trouble lies in the pump circuit.

Step 4. Pump . . .

If the pump has been spotted in Step 3 as the source of trouble, check the following:

The direction of rotation should agree with the arrow stamped on case on the pump. Accidental interchange of any 2 wires of a 3-phase system will cause the electric motor to change direction of rotation.

Also check for slipping belts, sheared shaft key, broken shaft, broken coupling, or loosened set screw.

Step 5. Relief Valve . . .

If Step 3 has shown the relief valve, D, to be at fault, disassemble, inspect, and clean the internal parts. Run a small wire through all internal orifices.

IMPORTANT: Quite often a relief valve spool will stick because the pipe connections have been screwed into it too tightly, distorting the body and binding the spool. When inspecting the valve, test the main spool for sticking before unscrewing the connections, if this is possbile.

Step 6. Cylinder . . .

Assuming the results of Step 2 show the pump and relief valve are working properly, re-connect Junction E and proceed to test the cylinder. Oil leaking past a worn piston packing, F, cannot be seen externally but can cause a high loss of circuit pressure and piston rod force.

Piston blow-by can be detected by disconnecting one cylinder line, I or J, and applying pump pressure to the opposite end of the cylinder. A small leakage, equivalent to a slowly dripping faucet, can sometimes be tolerated. Piston packings should be replaced if there is a continuous stream of leakage oil. Test for leakage in both directions. When breaking Joint I or J, plug the open pipe end leading to the valve, as the residual tank line pressure in some circuits may cause an unnecessary loss of hydraulic fluid from the open line.

Sometimes a cylinder loses force at one particular point in its travel, indicating a defect in the barrel at that point. It may be necessary to mechanically block the piston rod at that point while making the above test.

Step 7. Control Valve . . .

An excessively worn 4-way control valve can cause loss of high pressure. If the results of Step 2 testing show the pump and relief valve to be correctly functioning, test for excessive control valve leakage as follows:

Disconnect the cylinder lines and plug both cylinder ports on the valve. When the valve handle is thrown to either extreme position, it should be possible to build up full pump pressure on the gauge by adjusting the relief valve knob.

Alternate Test Method . . .

The 3 most likely trouble points in the system are: pump suction strainer, A; piston blow-by, at F; and relief valve sticking, in that order. Always inspect and clean the pump suction strainer as per Step 1 regardless of what other troubles may be turned up. Leakage past the cylinder piston can be detected at Point G by first allowing the cylinder to bottom out, then leaving the valve handle in the thrown position.

Relief valve sticking can be detected during the same test by observing flow at Point H. Full flow at this point indicates malfunctioning relief valve provided the cylinder is not stalled.

CHANGING HYDRAULIC OIL

All hydraulic oil has a definite, useful life span, and when it has deteriorated to near the danger point, it should be discarded.

One major cause of short oil life is operation at too high a temperature. This speeds up the oxidation process, which forms acids and sludge in the oil, causing rapid wear and corrosion to moving parts in the system. An oil temperature of from 120 to 130°F. is ideal for a hydraulic system. Check your oil temperature occasionally with a thermometer, or by placing your hand on the outside of the tank below the oil level. At 120°F. it is uncomfortable to leave your hand on the tank for more than a few seconds. If the tank is too hot to be touched at all, check it with a thermometer, and add an oil cooler if necessary.

Make a visual inspection of your oil once in a while. Compare the color and body with an unused sample of the same oil. A slight darkening is usually not serious, but a deep, dark color or a noticeable thickening may indicate a serious deterioration. Feel a smudge of oil between your fingers to detect small pieces of grit.

On large volume systems, consult your oil company representative about having a sample tested. On small volume systems it is cheaper to discard the used oil if there is any doubt about its purity or cleanliness.

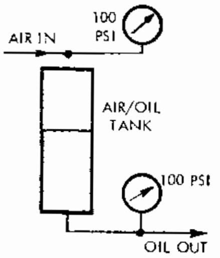
Combination Air/Oil Applications

Air/Oil Principle

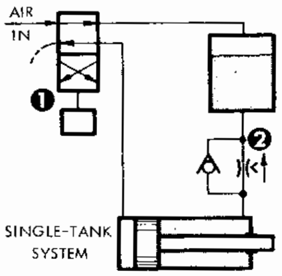
Air-over-oil systems are used where an air cylinder would normally be used but where better metering control is required.

Air pressure is applied to the top of a closed oil tank, and this develops an equal oil pressure which can be handled with low pressure hydraulic components such as check, flow control, and directional control valves.

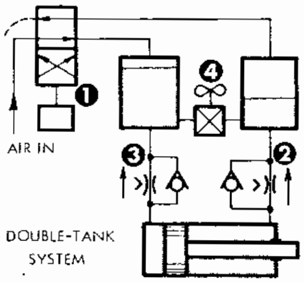
Advantages are: (1), the cylinder can be throttled more accurately, (2), better control over lunge if the tool breaks through the work, (3), the cylinder can be stopped more accurately in mid-stroke by shutting off the oil.



Basic Air/Oil Circuits



SINGLE-TANK SYSTEM



DOUBLE-TANK SYSTEM

A single-tank system is used for controlled slow feeding in one direction, with rapid cylinder return. The arrangement shown on the left is for slow feeding on the extension stroke with flow control Valve 2. For slow feed on the retraction stroke, the cylinder connections would be reversed.

The double-tank system is used for controlled motion in both directions. There may be a gradual interchange of oil from one tank to the other over a long period of time. To restore the balance of oil between the tanks, Valve 4 may be opened when the air pressure is on the fuller tank, and this will transfer oil to the depleted tank.

Design Considerations

The tank should be mounted in an elevated position above the cylinder to allow air to bleed out of the oil. Cylinders should have very tight piston seals to minimize transfer of oil. Plumbing lines should be oversize to those used in an ordinary hydraulic system, to keep pressure losses low, and to realize acceptable speed. In any air/oil system there will be a small amount of oil blown out the air valve exhaust. The exhaust may be piped to the outside or discharged into a container to collect the blow-by. Oil tanks should be checked regularly.

Construction of Tanks

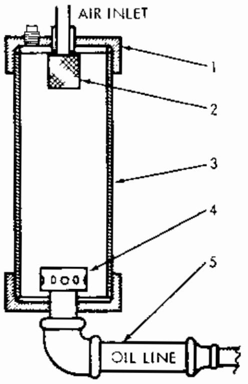
(1). The top must be removable or with a filler plug, so level can be checked and replenished.

(2). Use an air diffuser inside the tank to prevent excessive oil turbulence from a high velocity air stream. An air muffler works well.

(3). Tank configuration should be "tall and thin" to keep air and oil connections well separated.

(4). Use an oil diffuser at the oil inlet to break up the high velocity oil stream, to prevent transfer into the air circuit.

(5). Reduce the oil velocity before it enters the tank by using extra large diameter plumbing for about 12 inches adjacent to the tank.



Fluid Power Formulae

Pressure loss per foot length of pipe:

$P = V \times Q \div 18{,}300D^4$

P is loss in PSI per foot.

V is SSU viscosity at the operating temperature.

Q is GPM flow.

D is inside diameter of pipe, inches.

Thrust of an air or hydraulic cylinder:

$T = A \times PSI$

Thrust is in pounds.

A is piston or "net" area, sq. ins.

PSI is gauge pressure.

Force for piercing or shearing sheet metal:

$F = P \times T \times PSI$

Force is in pounds.

P is hole perimeter, in inches.

T is metal thickness, in inches.

PSI is shear strength of the material, in pounds per square inch.

Circle formulae:

Area $= \pi r^2$ or, $\pi D^2 \div 4$

Circumference $= 2\pi r$ or, πD

r is radius, D is diameter; π is 3.14

Torque/Horsepower relations:

$T = HP \times 5252 \div RPM$

$HP = T \times RPM \div 5252$

Torque is in foot pounds.

Hydraulic (fluid power) horsepower:

$HP = 0.000583 \times GPM \times PSI$

Heat equivalent of fluid power:

BTU/hr. $= 1\frac{1}{2} \times PSI \times GPM$

Usable force developed on a machine motion by cylinder pushing at an angle.

$T = F \times \sin A$

T is usable force, lbs.

F is cylinder thrust, lbs.

A is angle between cylinder axis and machine travel direction.

Velocity of oil flow in a pipe:

$V = GPM \times 0.3208 \div A$

Velocity is in feet per second.

A is inside square inch area.

Heat radiation capacity of a steel reservoir:

$HP = 0.001 \times A \times TD$

HP is radiating capacity in horsepower.

A is surface area, in square feet.

TD is temperature difference, in degrees F, between oil and surrounding air temperature.

Behavior of gases:

Boyles law: $P_1V_1 = P_2V_2$

Charles law: $T_1P_2 = T_2P_1$ or, $T_1V_2 = T_2V_1$

P_1, V_1 T_1 are initial values, P_2, V_2, T_2 are final values of absolute pressure, absolute temperature, and volume.

Hydraulic cylinder travel speed:

$S = CIM \div A$

Speed is in inches per minute.

CIM is cubic inches/minute flow.

A is piston or "net" area in square inches.

Effective force of a cylinder working at an angle to the direction of the load:

$F = T \times \sin A$

F is total cylinder force, in pounds.

T is effective thrust, in pounds.

A is least angle, in degrees between cylinder direction and load direction.

Formulae found on preceding pages of this booklet:

Toggle mechanism	Page 9
Vehicle drive	Page 31
Heat formulae	Page 32
Internal pres., tubing	Page 37
Conversion formulae	Page 40