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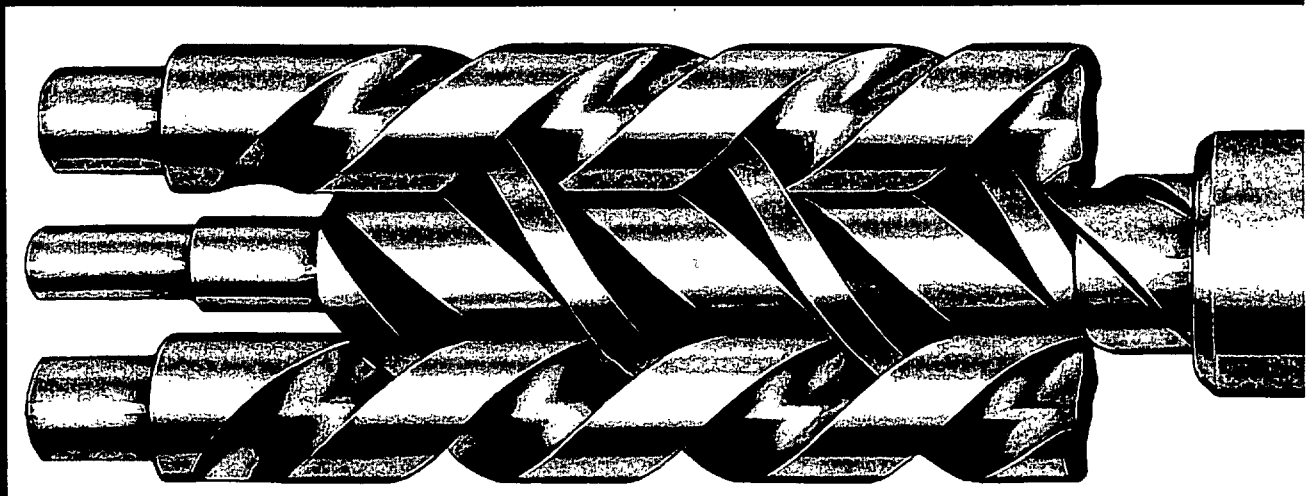
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DE LAVAL

IMO®

PUMP DATA BOOK

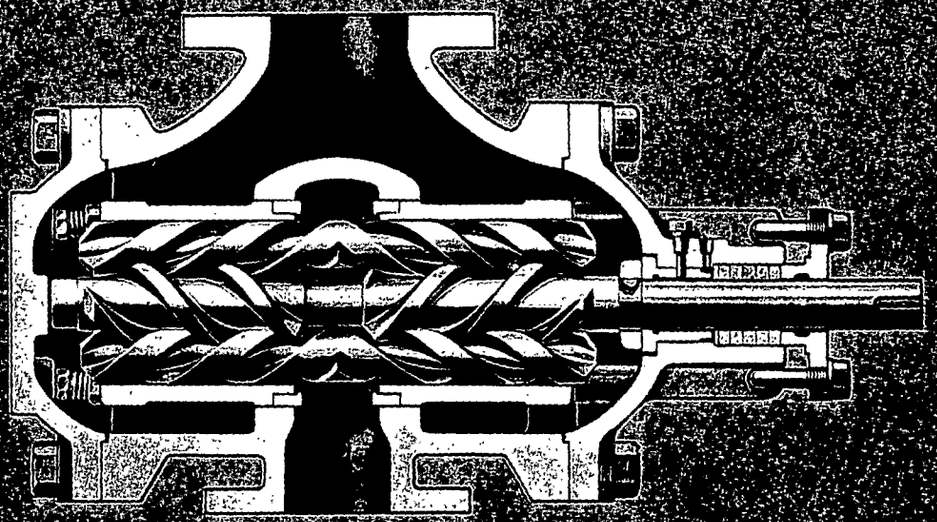


DE LAVAL TURBINE INC.
POWER AUXILIARIES DIVISION
TRENTON, NEW JERSEY 08802

DE LAVAL IMO®

Rotary, Constant-Displacement Pumps and Hydraulic Motors

The trade name IMO is derived from the names of the Swedish developers of the De Laval IMO pump.



The De Laval IMO® pump is a constant-displacement, rotary, screw type. The illustration shows the basic elements of the IMO pump design — three screws with thread surfaces shaped so they form a tight seal relative to each other and to the surrounding housing or cylinder. As the screws rotate, closures are formed sealed from each other by the threads. These closures transform rotational motion into continuous and uniform axial motion.

Thorough pioneer mathematical analysis led to the design of the screws for the IMO pump. This resulted in (1) the theoretical principle for complete sealing (2) the determination of kinematically correct thread proportions (3) the selection of the correct number of threads and screws.

In the standardized construction of the IMO pump design, there are three fully meshing screws — one power screw (rotor) and two idler screws (rotors) placed symmetrically. Proportions of the three double-threaded screws allow a one-to-one ratio between the rotors so that the idler rotors roll on the power rotor. The mathematical analysis of screw thread proportions proved that fluid pressure could be used to drive the idler or sealing rotors. Thus, the entire work is delivered or absorbed by the power rotor. The idlers, which rotate in the opposite direction, serve merely as sealing elements. Consequently, in the unique IMO pump design, the screws need not be connected by power or timing gears.

The threads of the IMO pump screws are shaped so that the volume enclosed by the threads remains constant during the entire pumping cycle. There are no tooth or end clearance volumes to pocket the liquid pumped. Thus, no destructive hydraulic compression or fluid shock forces occur between the threads; no internal leakage canals are needed to relieve such forces and diminish capacity.

The IMO pump is exceptionally quiet in operation because of the natural, smooth rolling action of the cooperating screws, elimination of timing gears and absence of hydraulic hammer. The velocity of the liquid being pumped is inherently low because of the theoretical correctness of the design and arrangement of the IMO pump screws. Thus, even at direct-connected motor or turbine speeds, De Laval IMO pumps perform at high operating efficiencies.

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ROTARY PUMPS
Basic Considerations In Their Applications

In the application of rotary pumps there are certain factors which must be considered in order to insure a successful installation. These are fundamentally the same regardless of the fluids to be handled or the pumping conditions. Although the primary purpose of this paper is to acquaint you with these factors and their relation to rotary pumps it might be well to first briefly consider the various basic type of pumps.

In general, power pumps may be put into three classes: centrifugal, reciprocating and rotary.

The centrifugal pump develops its pressure as a result of centrifugal force, has constant delivery characteristics and is mostly used where large volumes at relatively low pressures are required. Pumps of this type operate at comparatively high speed usually direct connected to their driver. They are not, however, self priming except in the case of some very special designs and the delivered capacity will vary considerably with any change in discharge pressure. Centrifugal pumps are not particularly adapted to handling viscous fluids; although they are occasionally used for applications of this nature, the efficiency usually drops off quite rapidly if viscosities above 500 to 1000 SSU are encountered.

The reciprocating pump has positive pressure characteristics and is used principally to handle small volumes at relatively high pressures. Due to its reciprocating motion and the inertia effect of the parts, speeds are relatively low. This type of pump is self priming and the delivered capacity is practically constant regardless of discharge pressure. It can handle with uniformly high volumetric efficiency practically all types of fluids.

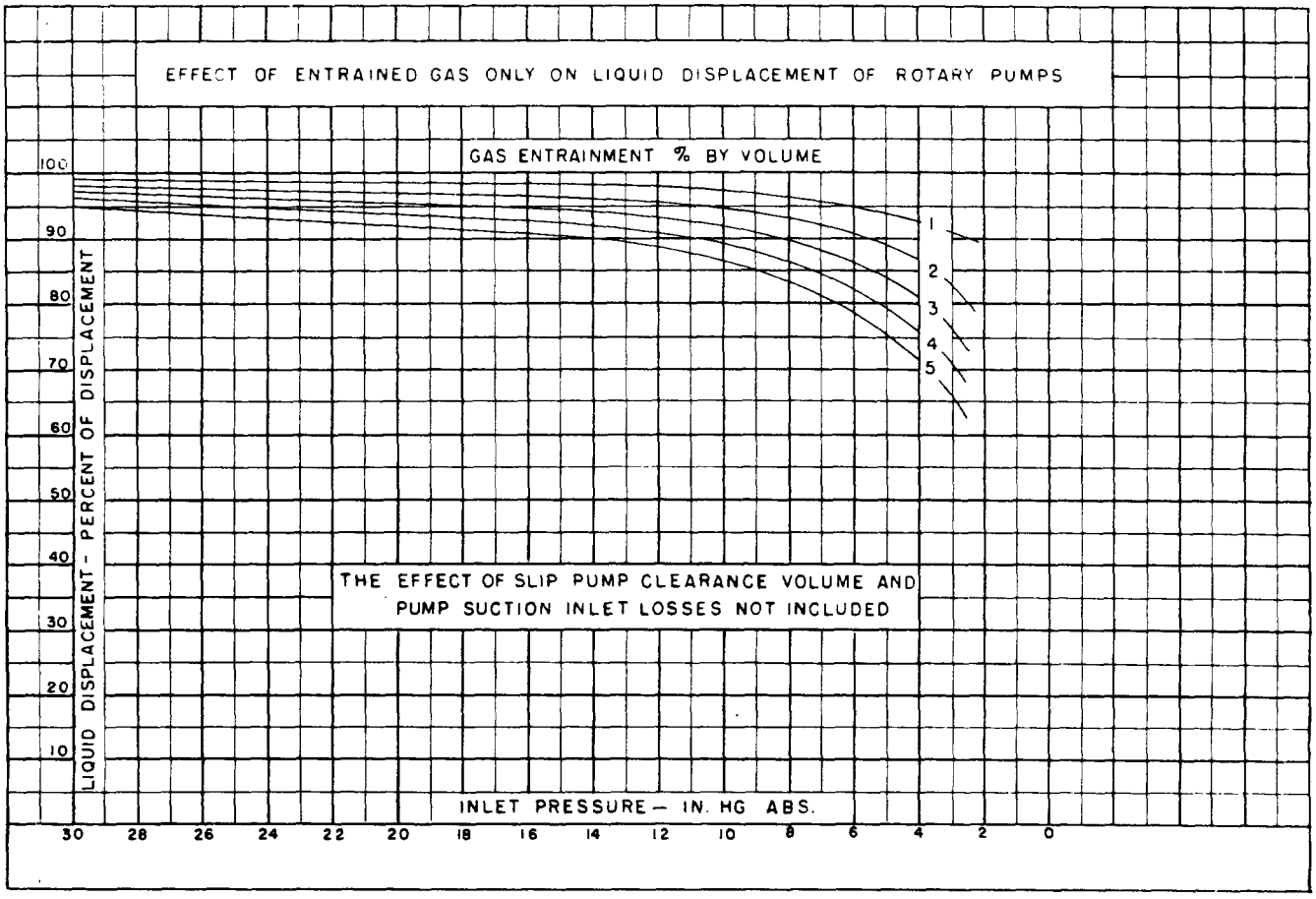
The rotary pump combines the rotary motion of the centrifugal with the positive pressure characteristics of the reciprocating pump. Like the reciprocating pump it is a positive displacement device that delivers with each revolution a given quantity of fluid, is self priming, and gives practically constant delivered capacity regardless of pressure. Speeds are much higher, however, than normally found in reciprocating pumps with the result that in probably the majority of cases direct connected motors can be used. Rotaries are available for pumping practically any fluid that will flow, and their greatest field is in the handling of viscous fluids. The rotary pump is known as the "work horse of industry" and is today, keeping full the pipelines of many processes, that otherwise would be unable to function.

By definition of the standards of Hydraulic Institute, a rotary pump is a positive displacement pump consisting of a fixed casing containing gears, cams, screws, plungers or similar elements actuated by rotation of the drive shaft.

Pumps in this classification commonly fall into six basic types, the gear, the vane, the screw, the cam and piston, the shuttle block, and the multiple piston, practically all of which can be broken down into further subdivisions.

The gear type which is perhaps the most common, consists of the external gear (fig. 1) including spur, helical and herringbone teeth, the lobular or impeller type, (fig. 2) and the internal gear, (fig. 3). Depending on the design and service, gear type pumps may be furnished either with or without timing gears. All have constant displacement characteristics.

The vane type pumps consist essentially of the sliding vane (fig. 4) and swinging vane, or bucket types, (fig. 5). Although the majority are designed for constant displacement, there are designs available in the sliding vane type, wherein the displacement is accomplished by incorporating a device



which governs the amount of vane movement.

Screw pumps may have one, two or three rotors with threads of various contours, depending on the design. The majority of two rotor pumps require timing gears whereas the other types do not. All are of the constant displacement type and are illustrated by Figures 6, 7 and 8.

The cam and piston type (fig. 9) is exactly what its name implies. It combines the rotary motion of a cam in combination with a piston. Usually two sets are utilized 180° apart so that a fairly steady flow results. This type of pump is particularly applicable on high vacuum service.

The shuttle block pump (fig. 10) is essentially a piston type pump. It contains a slotted rotor having a pin set eccentric to it in such a manner that it will cause the pistons to move in and out as the rotor is turned. By varying the position of the pin differences in stroke can be obtained and as a result, variable displacement.

Multiple piston pumps may be of the radial or axial type (fig. 11 & 12). In either, a series of pistons are actuated by a rotor mounted off center in the casing. Pumps of this type have very high volumetric efficiencies and are used a great deal in the hydraulic field,. they are particularly well suited where variable displacement is desired.

The pumping action of all rotary pumps is essentially the same As the pumping elements are rotated they open on the inlet side creating a void. The fluid, forced by atmospheric pressure, flows in to fill this space. The continued turning of the rotors encloses the fluid between the rotating parts themselves, or between these parts and the pump casing. They are at this point under inlet pressure and continue so until the enclosed portion opens into the outlet chamber.

Since the pump is a positive displacement device it is apparent that with every revolution of the shaft a definite quantity of fluid passes through. If no clearance existed this quantity, called "theoretical capacity," would be dependent only on the physical dimensions of the pumping elements and the speed. Clearances, however, do exist with the result that whenever a pressure differential occurs, there will always be internal leakage from the outlet side to the inlet side. This leakage, commonly known as "slip" may be relatively large or small depending on the type of pump, the amount of clearance, the viscosity of the fluid handled, and the differential between outlet and inlet pressures. For any given set of conditions it is usually unaffected by speed. The delivered or net capacity, therefore, is the theoretical capacity less the slippage.

The theoretical capacity of any pump can readily be calculated providing all the essential dimensions are known. Slip can likewise be calculated but is usually based on empirical values developed as a result of tests. This data is a part of the "stock in trade" of every pump manufacturer and capacities can be accurately predicted for any given set of conditions.

The selection of a pump for a specific application is not difficult if all of the operating conditions are known. It is

tation, it is common practice to use the term "Thixotropy" to cover both.

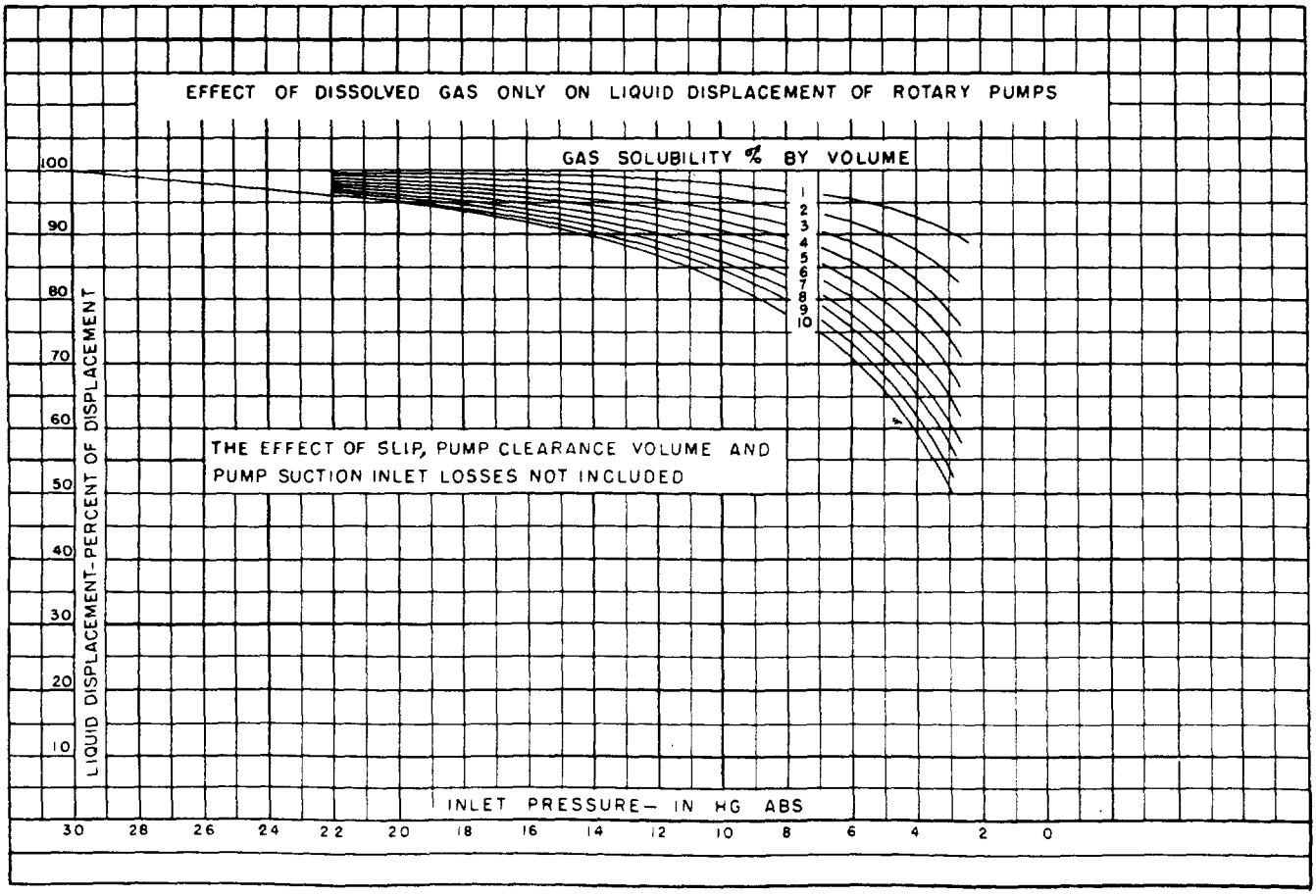
If a liquid is known to be thixotropic the expected viscosity under actual pumping conditions should be determined since it can vary quite widely from the viscosity under static conditions. One instance comes to mind concerning the handling of a cellulose product where the viscosity was given as 20,000 SSU which was its actual static, or apparent, viscosity. It later developed that under actual pumping conditions the viscosity was approximately 500 SSU. No serious harm was done, but a large low speed pump was installed where a smaller, cheaper, higher speed unit could have been used.

The grease manufacturing industry is very familiar with the thixotropic properties of its products, as evidenced by the numerous curves which had been published wherein "apparent viscosity" is plotted against "rate of shear." The occasion is rare, however, when one is able to obtain accurate information as to viscosity, when it is necessary to select a pump for handling this material.

When a pump application engineer receives an inquiry for the handling of a very viscous fluid and is unable to ascertain the viscosity he will generally attempt to visualize what the material might be like. In the case of grease, for example, he probably will recall the five gallon container which he recently saw in his favorite gasoline service station, which was so viscous that the container could be turned over and the grease would not flow. As a result he is likely to decide that the material is too viscous for any pump and refuse to quote or else the offering will be a very large, slow speed and expensive unit, with what seems to be a senseless request for exceptionally high suction head. Actually, if the truth were known, the source of the material in question very likely would be a mixing kettle or some similar device where the material was being agitated at a fairly high temperature, with a resulting low viscosity.

It is understood that it is practically impossible in most instances to give the viscosity of grease in the terms most familiar to the pump manufacture, i.e.; Saybolt Seconds Universal or Saybolt Seconds Furol, but if only a rough approximation could be given it would be of great help. Grease penetration test results are sometimes given, but since there is no relation between these values and the conditions under which the material must be pumped they do not mean too much.

For applications of this type the information, if available, which would be of the greatest value to a pump engineer is data taken from a similar installation. Such information should consist of type, size, capacity, and speed of already installed pumps, suction pressure, and temperature at the pump inlet flange, total working suction head, and above all the pressure drop in a specified length of piping. From



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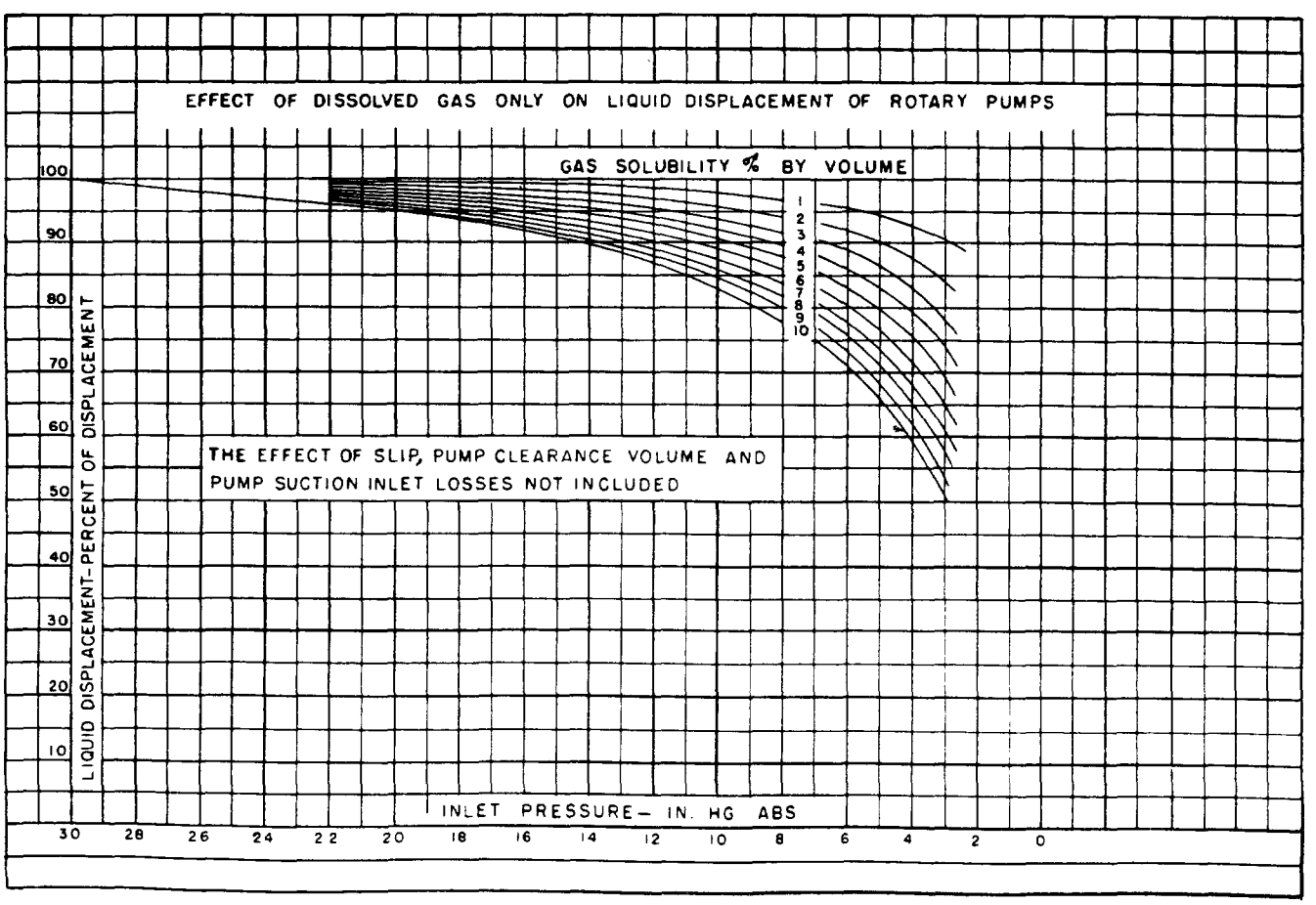
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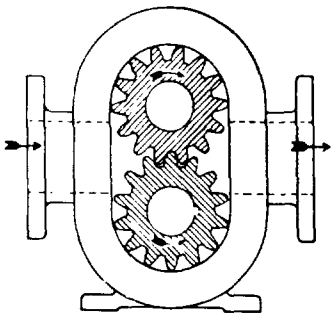


Figure 1
External Gear Pump

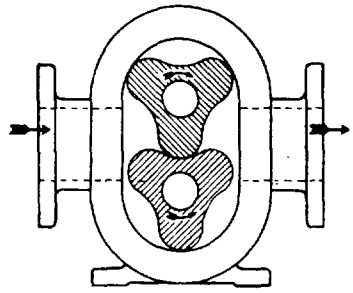


Figure 2
Three Lobe Pump

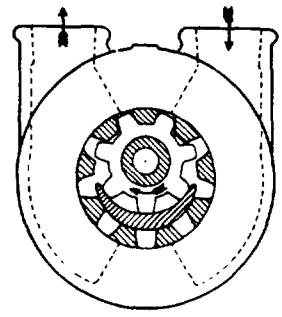


Figure 3
Internal Gear Pump

the latter an excellent approximation of viscosity under actual operating conditions can be obtained.

Suction lift occurs where the total suction head at the pump inlet is below atmospheric pressure. It is normally the result of a static lift and pipe friction. Although rotary pumps are capable of producing a high vacuum it is not this vacuum that forces the fluid to flow. As previously explained, it is atmospheric pressure that forces the fluid into the pump. Since atmospheric pressure at sea-level corresponds to 14.7 psia or 30" Hg. this is the maximum amount of pressure available for moving the fluid and suction lift cannot exceed these figures. Actually it must be somewhat less, since there are always pump inlet losses which must be taken into account. It is considered the best practice to keep suction lifts just as low as possible.

The majority of rotary pumps operate with suction lifts of approximately 5 to 15" Hg. Lifts corresponding to 24-25" Hg. are not uncommon and there are numerous installations operating continuously and satisfactorily where the absolute suction pressure is within one half inch of the barometer. In the latter cases, however, the pumps are usually taking the fluid from tanks under vacuum and no entrained or dissolved air or gases are present. Great care must be taken in selecting pumps for these applications since the inlet losses can very easily exceed the net suction head available for moving the fluid into the pumping elements.

These are known instances of successful installations where pumps were properly selected for the suction conditions. There are also, unfortunately, many other installations with equally high suction lifts which are not so satisfactory. This is due to the fact that proper consideration was not given, at the time the installations were made, to the actual suction conditions at the pump inlet. Frequently, suction conditions are given as "flooded" simply because the source feeding the pump is above the inlet flange. Absolutely no consideration is given to outlet losses from the tank or pipe friction and these can be exceptionally high when dealing with extremely viscous fluids.

An outstanding and almost unbelievable example in this respect is one concerning an inquiry relative to the handling of glue at 20,000 SSU. Although the inquiry stated that the suction would be flooded, a request was made that the customer submit additional information concerning the piping layout on the suction side. When the information arrived it was found by calculating friction losses that although the tanks were elevated above the pump, and the total length of the suction line was only approximately 20 feet, the small size of the line plus a veritable maze of fittings and valves resulted in a total friction loss of approximately 350 pounds. This obviously is an impossible pumping condition.

Where it is desired to pump extremely viscous fluids such as grease, chilled shortening, cellulose preparations and the

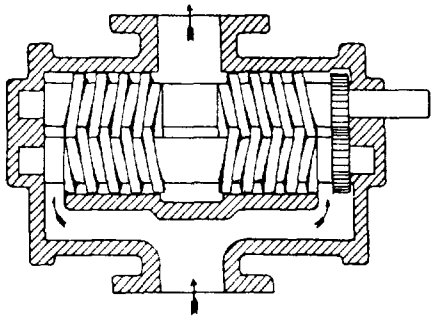


Figure 7
Two Screw Pump

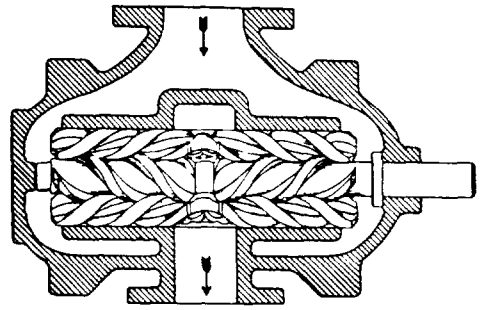


Figure 8
Three Screw Pump

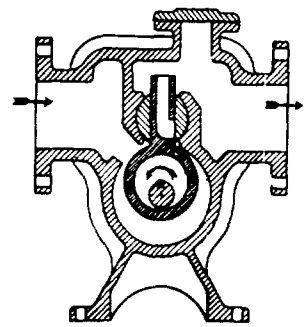


Figure 9
Cam and Piston Pump

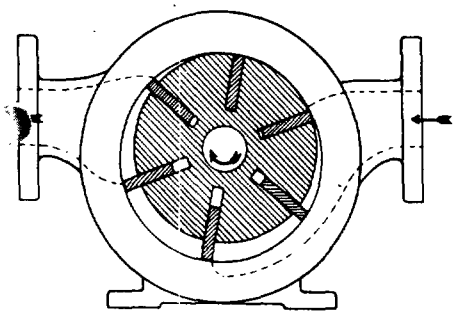


Figure 4
Sliding Vane Pump

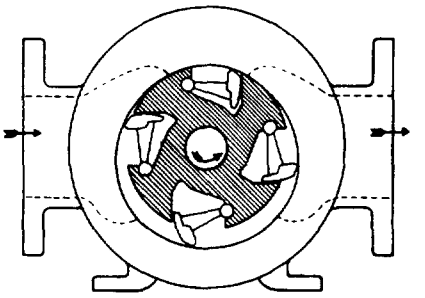


Figure 5
Swinging Vane Pump

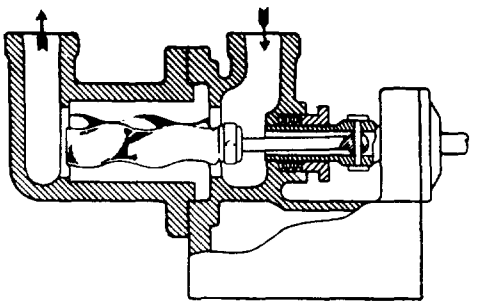


Figure 6
Single Screw Pump

like, care should be taken to use the largest possible size of suction piping, eliminate all unnecessary fittings and valves, and to place the pump just as closely as possible to the source of supply. In addition, it may be found necessary to supply the fluid to the pump under some pressure, which may be supplied by elevation, air pressure, or mechanical means such as a screw conveyor.

It was previously stated that viscosity and speed are closely tied together and it is impossible to consider that one without the other. Although rotative speed is the ultimate outcome, the basic speed which the manufacturer must consider is the velocity of the fluid going through the pump; this is a function of pump type and design. Certain types such as gear and vane pumps carry the fluid around the periphery of the pumping elements and as a result the velocity of the fluid through the pump can become quite high unless relatively low rotative speeds are used. On the other hand, in screw type pumps, the flow is axial and fluid speeds are relatively lower with the result that higher rotative speeds can be used. Based on handling light fluids, say 100 to 500 SSU, gear or vane type pumps rarely exceed a rotative speed of 1200 RPM except in the case of a very small unit or special designs for a particular use, such as for aircraft purposes. Screw pumps, however, where timing gears are not required commonly operate without difficulty at speeds up to 5000 RPM and there are instances where special designs are operating at speeds of approximately 24,000 RPM taking oil under lift from a sump tank located below the pump.

Although rotative speeds are relative and dependent on the pump type, they usually should be reduced when handling fluids of high viscosity. This is due not only to the difficulty of filling the pumping elements, but also to the mechanical losses which result from the shearing action of these parts on the fluid handled. The reduction of these losses is frequently of more importance than relatively high speeds, even though the latter might be possible due to positive inlet conditions.

No direct comparison can be made as to mechanical loss between the various types of pumps, most manufacturers have established their own data on a basis of tests made under closely controlled operating conditions and these are so-called "trade secrets." In general, the losses for a given size and type of pump vary with the viscosity and rotative speed and may or may not be affected by pressure, depending on the type of pump under consideration. These losses, however, must always be based on the maximum viscosity to be handled, since they will be highest at this point. If this is not done the resultant calculated horsepower will be too low and a motor selected on this basis will probably be overloaded.

The other factor that determines the horsepower required

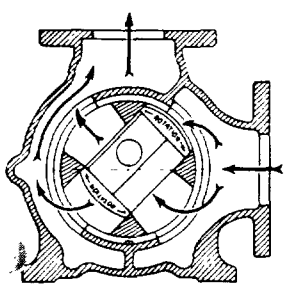


Figure 10
Shuttle Block Pump

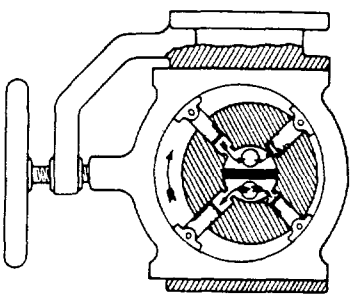


Figure 11
Radial Plunger Pump

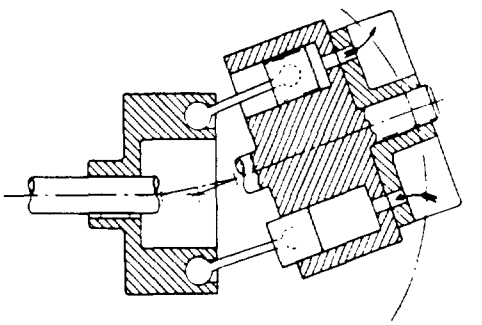


Figure 12
Axial Plunger Pump

to drive the pump is the actual work done in converting the fluid from its pressure condition on the inlet side to its corresponding condition on the outlet side. Since this work is done on all of the fluid (this must be so because slip does not exist until a pressure differential occurs), the volume which must be considered is not the delivered capacity but the actual displacement volume, or theoretical capacity, of the pump. Any number of formulae can be devised for calculating this value, known as the "theoretical liquid horsepower," depending on the units in which capacity and head, or pressure, are expressed. It should be noted that the theoretical liquid horsepower is independent of viscosity and is concerned only with the physical dimensions of the pumping elements, the rotative speed, and the total pressure.

The brake horsepower required is the sum of the theoretical liquid horsepower and the mechanical loss.

Rotary pumps are available in a wide variety of pressure ranges. Although the great majority are sold for pressures less than 150 psi, which is within the capabilities of almost any type of pump, a good size field exists for a great number up to 500 psi and literally thousands of precision pumps are built for hydraulic service on machine tools, and the like, where pressures up to 3000 psi may be encountered.

Rotary pumps do not in themselves create pressure, they simply transfer a quantity of fluid from the inlet to the outlet side. The pressure developed on the outlet side is solely the result of resistance to flow in the discharge line. If, for example, a pump were to be set up and run without a discharge line, a gauge placed at the pump outlet flange would register zero no matter how fast or how long it was run.

Resistance usually consists of differences of elevation, fixed resistances such as orifices, and pipe friction. Nothing much can be done about the first two since these are the basic reasons for using a pump, something however, can be done about pipe friction. Literally millions of dollars are thrown away annually due to the use of piping that is too small for the job. To be sure, all pipe friction cannot be eliminated as long as fluids must be handled in this manner but every effort should be made to use the largest pipe that is economically feasible. Numerous tables are available from which friction losses in any combination of piping may be calculated, among the most recent of which are those published by the Hydraulic Institute.

Before any new installation is made the cost of larger size piping which will result in lower pump pressures, should be carefully balanced against the cost of a less expensive pump, smaller motor, and a saving in horsepower over the expected life of the system. The larger piping may cost a little more in the beginning but the ultimate savings in power will often many times offset the original cost. These facts are particularly true of the handling of extremely viscous fluids and although most engineers dealing with fluids of this type are conscious of what can be done, it is surprising the number of installations encountered where considerable savings could have been made if a little more study had been made initially.

The question is frequently asked "What is the efficiency of a rotary pump?" This is rather difficult to answer because of the many variables. Efficiency is dependent on practically all of the factors already discussed such as size of pump, rotative speed, viscosity, and pressure differential. It can be calculated for any specified set of conditions but will vary with any change in these conditions.

Sometimes an attempt is made to calculate the efficiency on the basis of guaranteed capacity and brake horsepower, this is satisfactory provided that both were figured on the same viscosity. Normally, however, where rotary pumps are used a viscosity range has been specified or assumed, the capacity guaranteed on the minimum viscosity and the brake horsepower on the maximum. If these guarantee figures are used in an attempt to calculate efficiency a false value will result.

Good efficiencies can be obtained if a pump is properly selected to meet one set of conditions only. For example, medium size pumps of about 50 to 100 GPM capacity operating against 100 psi discharge pressure, normally have efficiencies of about 70% to 80%. Higher pressure pumps of about the same capacity, selected for optimum conditions, will give as high as 85% or 87%.

There is one other point that we have not as yet touched and that is the handling of fluids containing abrasives. Due to the fact that rotary pumps depend upon close clearances for proper pumping action the handling of abrasive fluids will usually cause rapid wear. Some progress has been made in the use of harder and more abrasive resistant materials for the pumping elements so that a fair job can be done in some instances. It cannot be said, however, that performance is too satisfactory when handling fluids of this character. On the whole, rotary pumps should not be used for handling fluids containing abrasives unless frequent replacements are acceptable.

Over the last twenty-year period, great progress has been made in the design and application of rotary pumps and further progress is being made constantly. Pumps of this type are available for the handling of fluids ranging from aviation gasoline and water to semi-plastics such as cellulose products, chilled shortening and greases. Where required, extremely high vacuum, pressures, and speeds are possible and practical.

If complete information is given as to suction conditions, viscosity, air or gas entrainment, discharge pressure, etc. so that a proper pump selection can be made, and the proper attention is paid to installation features particularly on the suction side, there is absolutely no reason why a satisfactory installation will not result.

In this day and age of specialization, pump specialists are available and can help you with your problems. Practically all manufacturers have sales and application engineers available who are familiar with the capabilities and limitations of their own products and usually of their competitor's as well. To be sure, their prime object is to sell their equipment, but very few of them desire to misapply that equipment or run the chance of a "trouble" installation. All have a background of successful installations and can be of invaluable help in assisting you.

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VISCOSITY

Viscosity is that property of any fluid (liquid or gas) which tends to resist a shearing force. It is important to fluid flow because nearly all fluid motion is accompanied by shearing forces.

The two basic viscosity parameters are the DYNAMIC (or ABSOLUTE) VISCOSITY, μ, having the dimension, *force x time/(length)*^{2}, and the KINEMATIC VISCOSITY, ν, having the dimension, *(length)*^{2}*/time*. The parameters are related through the mass density of the fluid, ρ, such that, $\nu = \mu/\rho = \mu g/\gamma$, where γ is the specific weight and g is the acceleration of gravity.

The unit of dynamic viscosity in English measure is the *pound-second per square foot* which is numerically identical with the *slug per foot-second*. The unit of dynamic viscosity in Metric measure is the *dyne-second per square centimeter*, called the POISE, which is numerically identical with the *gram per centimeter-second*. Numerical values generally are expressed in CENTIPOISES, a unit which is one hundredth of a Poise. A unit called the REYN, equal to one *pound-second per square inch*, is used in lubrication problems.

The unit of kinematic viscosity in English measure is the *square foot per second*. The unit of kinematic viscosity in Metric measure is the *square centimeter per second*, called the STOKE. Numerical values generally are expressed in CENTISTOKES, a unit which is one hundredth of a Stoke.

Widespread use of the Saybolt viscosimeter has led to the use of the time of efflux in seconds, for 60 cc. of liquid, as an arbitrary unit of kinematic viscosity. The term SSU (Seconds Saybolt Universal) refers to the smaller, and the term SSF (Seconds Saybolt Furol) refers to the larger of two orifices with which the instrument may be equipped. Other empirical measures of kinematic viscosity may be converted to basic units by the chart on page 78.

The dynamic viscosity of any fluid is a function of temperature and pressure. The dynamic viscosity of most liquids increases with increase of pressure but, fortunately, the changes may be neglected for the ranges of pressure usually encountered in engineering problems. The dynamic viscosity of gases is virtually independent of pressure except at extremely high or low pressures. Pressure is very important in determining the kinematic viscosity of a gas due to its influence on the mass density. The kinematic viscosities of many fluids are shown in the charts on pages 80 to 83 inclusive. Viscosities of air and water at 68°F. and atmospheric pressure are as follows:

Fluid	Dynamic Viscosity μ		Kinematic Viscosity ν		
	Poises	lb-sec/sq ft	Stokes	Sq ft/sec	SSU
Air	180.8×10^{-6}	0.3369×10^{-6}	0.1501	161.6×10^{-6}	——
Water	0.010087	21.067×10^{-6}	0.010105	10.877×10^{-6}	30.1

VISCOSITY CONVERSION FACTORS AND FORMULAS

Multiply	by	to obtain
Poises	100	centipoises
pound-seconds/sq. ft.	47,880.1	centipoises
Reyns	6.89473×10^{6}	centipoises
centipoises	2.08855×10^{-5}	pound-seconds/sq. ft.
centipoises	1.45038×10^{-5}	Reyns
Stokes	100	centistokes
sq. ft./second	92,903.4	centistokes
centistokes	1.07639×10^{-5}	sq. ft./second
sq. ft./second	$1488.16 \times \gamma$ in $\frac{\text{pounds}}{\text{cu. ft.}}$	centipoises
centipoises	$\frac{6.71970 \times 10^{-4}}{\gamma \text{ in pounds/cu. ft.}}$	sq. ft./second

SAYBOLT VISCOSIMETER CONVERSION FORMULAS

μ, in centistokes $= 0.226 \times$ SSU $- 195/$SSU — for SSU $\leqslant 100$

μ, in centistokes $= 0.220 \times$ SSU $- 130/$SSU — for SSU > 100

μ, in centistokes $= 2.24 \times$ SSF $- 184/$SSF — for $25 \leqslant$ SSF $\leqslant 40$

μ, in centistokes $= 2.16 \times$ SSF $- 60/$SSF — for SSF > 40

VISCOSITY CONVERSION TABLE

The following table will give a comparison of various viscosity ratings so that if the viscosity is given in terms other than Saybolt Universal, it can be translated quickly by following horizontally to the Saybolt Universal column.

Seconds Saybolt Universal ssu	Kinematic Viscosity Centistokes *	Seconds Saybolt Furol ssf	Seconds Redwood 1 (Standard)	Seconds Redwood 2 (Admiralty)	Degrees Engler	Degrees Barbey	Seconds Parlin Cup #7	Seconds Parlin Cup #10	Seconds Parlin Cup #15	Seconds Parlin Cup #20	Seconds Ford Cup #3	Seconds Ford Cup #4
31	1.00	-	29	-	1.00	6200	-	-	-	-	-	-
35	2.56	-	32.1	-	1.16	2420	-	-	-	-	-	-
40	4.30	-	36.2	5.10	1.31	1440	-	-	-	-	-	-
50	7.40	-	44.3	5.83	1.58	838	-	-	-	-	-	-
60	10.3	-	52.3	6.77	1.88	618	-	-	-	-	-	-
70	13.1	12.95	60.9	7.60	2.17	483	-	-	-	-	-	-
80	15.7	13.70	69.2	8.44	2.45	404	-	-	-	-	-	-
90	18.2	14.44	77.6	9.30	2.73	348	-	-	-	-	-	-
100	20.6	15.24	85.6	10.12	3.02	307	-	-	-	-	-	-
150	32.1	19.30	128	14.48	4.48	195	-	-	-	-	-	-
200	43.2	23.5	170	18.90	5.92	144	40	-	-	-	-	-
250	54.0	28.0	212	23.45	7.35	114	46	-	-	-	-	-
300	65.0	32.5	254	28.0	8.79	95	52.5	15	6.0	3.0	30	20
400	87.60	41.9	338	37.1	11.70	70.8	66	21	7.2	3.2	42	28
500	110.0	51.6	423	46.2	14.60	56.4	79	25	7.8	3.4	50	34
600	132	61.4	508	55.4	17.50	47.0	92	30	8.5	3.6	58	40
700	154	71.1	592	64.6	20.45	40.3	106	35	9.0	3.9	67	45
800	176	81.0	677	73.8	23.35	35.2	120	39	9.8	4.1	74	50
900	198	91.0	762	83.0	26.30	31.3	135	41	10.7	4.3	82	57
1000	220	100.7	896	92.1	29.20	28.2	149	43	11.5	4.5	90	62
1500	330	150	1270	138.2	43.80	18.7	-	65	15.2	6.3	132	90
2000	440	200	1690	184.2	58.40	14.1	-	86	19.5	7.5	172	118
2500	550	250	2120	230	73.0	11.3	-	108	24	9	218	147
3000	660	300	2540	276	87.60	9.4	-	129	28.5	11	258	172
4000	880	400	3380	368	117.0	7.05	-	172	37	14	337	230
5000	1100	500	4230	461	146	5.64	-	215	47	18	425	290
6000	1320	600	5080	553	175	4.70	-	258	57	22	520	350
7000	1540	700	5920	645	204.5	4.03	-	300	67	25	600	410
8000	1760	800	6770	737	233.5	3.52	-	344	76	29	680	465
9000	1980	900	7620	829	263	3.13	-	387	86	32	780	520
10000	2200	1000	8460	921	292	2.82	-	430	96	35	850	575
15000	3300	1500	13700	-	438	2.50	-	650	147	53	1280	860
20000	4400	2000	18400	-	584	1.40	-	860	203	70	1715	1150

* Kinematic Viscosity (in centistokes) $= \dfrac{\text{Absolute viscosity (in centipoises)}}{\text{Specific Gravity}}$

Above 250 SSU, use the following approximate conversion

SSU = Centistokes x 4.62

Above the range of this table and within the range of the viscosimeter, multiply their rating by the following factors to convert to SSU.

Viscosimeter	Factor
Saybolt Furol	10.
Redwood Standard	1.095
Redwood Admiralty	10.87
Engler - Degrees	34.5

Viscosimeter	Factor
Parlin cup #15	98.2
Parlin cup #20	187.0
Ford cup # 4	17.4

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VISCOSITY CONVERSION TABLE

The following table will give a comparison of various viscosity ratings so that if the viscosity is given in terms other than Saybolt Universal, it can be translated quickly by following horizontally to the Saybolt Universal column.

Seconds Saybolt Universal ssu	Kinematic Viscosity Centistokes*	Approx. Seconds Mac Michael	Approx. Gardner Holt Bubble	Seconds Zahn Cup #1	Seconds Zahn Cup #2	Seconds Zahn Cup #3	Seconds Zahn Cup #4	Seconds Zahn Cup #5	Seconds Demmler Cup #1	Seconds Demmler Cup #10	Approx. Seconds Stormer 100 gm Load	Seconds Pratt and Lambert "F"
31	1.00	–	–	–	–	–	–	–	–	–	–	–
35	2.56	–	–	–	–	–	–	–	–	–	–	–
40	4.30	–	–	–	–	–	–	–	1.3	–	–	–
50	7.40	–	–	–	–	–	–	–	2.3	–	2.6	–
60	10.3	–	–	–	–	–	–	–	3.2	–	3.6	–
70	13.1	–	–	–	–	–	–	–	4.1	–	4.6	–
80	15.7	–	–	–	–	–	–	–	4.9	–	5.5	–
90	18.2	–	–	–	–	–	–	–	5.7	–	6.4	–
100	20.6	125	–	38	18	–	–	–	6.5	–	7.3	–
150	32.1	145	–	47	20	–	–	–	10.0	1.0	11.3	–
200	43.2	165	A	54	23	–	–	–	13.5	1.4	15.2	–
250	54.0	198	A	62	26	–	–	–	16.9	1.7	19	–
300	65.0	225	B	73	29	–	–	–	20.4	2.0	23	–
400	87.0	270	C	90	37	–	–	–	27.4	2.7	31	7
500	110.0	320	D	–	46	–	–	–	34.5	3.5	39	8
600	132	370	F	–	55	–	–	–	41	4.1	46	9
700	154	420	G	–	63	22.5	–	–	48	4.8	54	9.5
800	176	470	–	–	72	24.5	–	–	55	5.5	62	10.8
900	198	515	H	–	80	27	18	–	62	6.2	70	11.9
1000	220	570	I	–	88	29	20	13	69	6.9	77	12.4
1500	330	805	M	–	–	40	28	18	103	10.3	116	16.8
2000	440	1070	Q	–	–	51	34	24	137	13.7	154	22
2500	550	1325	T	–	–	63	41	29	172	17.2	193	27.6
3000	660	1690	U	–	–	75	48	33	206	20.6	232	33.7
4000	880	2110	V	–	–	–	63	43	275	27.5	308	45
5000	1100	2635	W	–	–	–	77	50	344	34.4	385	55.8
6000	1320	3145	X	–	–	–	–	65	413	41.3	462	65.5
7000	1540	3670	–	–	–	–	–	75	481	48	540	77
8000	1760	4170	Y	–	–	–	–	86	550	55	618	89
9000	1980	4700	–	–	–	–	–	96	620	62	695	102
10000	2200	5220	Z	–	–	–	–	–	690	69	770	113
15000	3300	7720	Z2	–	–	–	–	–	1030	103	1160	172
20000	4400	10500	Z3	–	–	–	–	–	1370	137	1540	234

$$\text{*Kinematic Viscosity (in centistokes)} = \frac{\text{Absolute viscosity (in centipoises)}}{\text{Specific Gravity}}$$

Above 250 SSU, use the following approximate conversion

$$\text{SSU} = \text{Centistokes} \times 4.62$$

Above the range of this table and within the range of the viscosimeter, multiply their rating by the following factors to convert to SSU.

Viscosimeter	Factor
Mac Michael	1.92 (approx.)
Demmler #1	14.6
Demmler #10	146.
Stormer	13. (approx.)

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VISCOSITY OF COMMON LIQUIDS

Liquid	*Sp Gr at 60 F	VISCOSITY		At F
		SSU	Centistokes	
Freon	1.37 to 1.49 @ 70 F		.27–.32	70
Glycerine (100%)	1.26 @ 68 F	2,950 813	648 176	68.6 100
Glycol:				
Propylene	1.038 @ 68 F	240.6	52	70
Triethylene	1.125 @ 68 F	185.7	40	70
Diethylene	1.12	149.7	32	70
Ethylene	1.125	88.4	17.8	70
Hydrochloric Acid (31.5%)	1.05 @ 68 F		1.9	68
Mercury	13.6		.118 .11	70 100
Phenol (Carbolic Acid)	.95 to 1.08	65	11.7	65
Silicate of Soda	40 Baumé 42 Baumé	365 637.6	79 138	100 100
Sulfuric Acid (100%)	1.83	75.7	14.6	68
FISH AND ANIMAL OILS:				
Bone Oil	.918	220 65	47.5 11.6	130 212
Cod Oil	.928	150 95	32.1 19.4	100 130
Lard	.96	287 160	62.1 34.3	100 130
Lard Oil	.912 to .925	190 to 220 112 to 128	41 to 47.5 23.4 to 27.1	100 130
Menhadden Oil	.933	140 90	29.8 18.2	100 130
Neatsfoot Oil	.917	230 130	49.7 27.5	100 130
Sperm Oil	.883	110 78	23.0 15.2	100 130
Whale Oil	.925	163 to 184 97 to 112	35 to 39.6 19.9 to 23.4	100 130
MINERAL OILS:				
Automobile Crankcase Oils (Average Midcontinent Paraffin Base):				
SAE 10	**.880 to .935	165 to 240 90 to 120	35.4 to 51.9 18.2 to 25.3	100 130
SAE 20	**.880 to .935	240 to 400 120 to 185	51.9 to 86.6 25.3 to 39.9	100 130
SAE 30	**.880 to .935	400 to 580 185 to 255	86.6 to 125.5 39.9 to 55.1	100 130

*Unless otherwise noted.

**Depends on origin or percent and type of solvent.

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VISCOSITY OF COMMON LIQUIDS

Liquid	*Sp Gr at 60 F	VISCOSITY		At F
		SSU	Centistokes	
SAE 40	**.880 to .935	580 to 950 255 to 80	125.5 to 205.6 55.1 to 15.6	100 130 210
SAE 50	**.880 to .935	950 to 1,600 80 to 105	205.6 to 352 15.6 to 21.6	100 210
SAE 60	**.880 to .935	1,600 to 2,300 105 to 125	352 to 507 21.6 to 26.2	100 210
SAE 70	**.880 to .935	2,300 to 3,100 125 to 150	507 to 682 26.2 to 31.8	100 210
SAE 10W	**.880 to .935	5,000 to 10,000	1,100 to 2,200	0
SAE 20W	**.880 to .935	10,000 to 40,000	2,200 to 8,800	0
Automobile Transmission Lubricants: SAE 80	**.880 to .935	100,000 max	22,000 max	0
SAE 90	**.880 to .935	800 to 1,500 300 to 500	173.2 to 324.7 64.5 to 108.2	100 130
SAE 140	**.880 to .935	950 to 2,300 120 to 200	205.6 to 507 25.1 to 42.9	130 210
SAE 250	**.880 to .935	Over 2,300 Over 200	Over 507 Over 42.9	130 210
Crude Oils: Texas, Oklahoma	.81 to .916	40 to 783 34.2 to 210	4.28 to 169.5 2.45 to 45.3	60 100
Wyoming, Montana	.86 to .88	74 to 1,215 46 to 320	14.1 to 263 6.16 to 69.3	60 100
California	.78 to .92	40 to 4,840 34 to 700	4.28 to 1,063 2.4 to 151.5	60 100
Pennsylvania	.8 to .85	46 to 216 38 to 86	6.16 to 46.7 3.64 to 17.2	60 100
Diesel Engine Lubricating Oils (Based on Average Midcontinent Paraffin Base): Federal Specification No. 9110	**.880 to .935	165 to 240 90 to 120	35.4 to 51.9 18.2 to 25.3	100 130
Federal Specification No. 9170	**.880 to .935	300 to 410 140 to 180	64.5 to 88.8 29.8 to 38.8	100 130
Federal Specification No. 9250	**.880 to .935	470 to 590 200 to 255	101.8 to 127.8 43.2 to 55.1	100 130
Federal Specification No. 9370	**.880 to .935	800 to 1,100 320 to 430	173.2 to 238.1 69.3 to 93.1	100 130
Federal Specification No. 9500	**.880 to .935	490 to 600 92 to 105	106.1 to 129.9 18.54 to 21.6	130 210

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VISCOSITY OF COMMON LIQUIDS

Liquid	*Sp Gr at 60 F	VISCOSITY		At F
		SSU	Centistokes	
Diesel Fuel Oils:				
No. 2 D	**.82 to .95	32.6 to 45.5	2 to 6	100
		39	1 to 3.97	130
No. 3 D	**.82 to .95	**45.5** to **65**	6 to 11.75	100
		39 to 48	3.97 to 6.78	130
No. 4 D	**.82 to .95	**140 max**	29.8 max	100
		70 max	13.1 max	130
No. 5D	**.82 to .95	**400 max**	86.6 max	122
		165 max	35.2 max	160
Fuel Oils:				
No. 1	**.82 to .95	34 to 40	2.39 to 4.28	70
		32 to 35	2.69	100
No. 2	**.82 to .95	36 to 50	3.0 to 7.4	70
		33 to 40	2.11 to 4.28	100
No. 3	**.82 to .95	**35 to 45**	2.69 to .584	100
		32.8 to 39	2.06 to 3.97	130
No. 5A	**.82 to .95	**50 to 125**	7.4 to 26.4	100
		42 to 72	4.91 to 13.73	130
No. 5B	**.82 to .95	125 to	26.4 to	100
		400	86.6	122
		72 to 310	13.63 to 67.1	130
No. 6	**.82 to .95	**450 to 3,000**	97.4 to 660	122
		175 to 780	37.5 to 172	160
Fuel Oil — Navy Specification	**.989 max	**110 to 225**	23 to 48.6	122
		63 to 115	11.08 to 23.9	160
Fuel Oil — Navy II	1.0 max	**1,500 max**	324.7 max	122
		480 max	104 max	160
Gasoline	.68 to .74		.46 to .88	60
			.40 to .71	100
Gasoline (Natural)	76.5 degrees API		.41	68
Gas Oil	28 degrees API	73	13.9	70
		50	7.4	100
Insulating Oil:				
Transformer, switches and		115 max	24.1 max	70
circuit breakers		**65 max**	11.75 max	100
Kerosene	.78 to .82	35	2.69	68
		32.6	2	100
Machine Lubricating Oil (Average Pennsylvania Paraffin Base):				
Federal Specification No. 8	**.880 to .935	112 to 160	23.4 to 34.3	100
		70 to 90	13.1 to 18.2	130

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VISCOSITY OF COMMON LIQUIDS

Liquid	*Sp Gr at 60 F	VISCOSITY		At F
		SSU	Centistokes	
Federal Specification No. 10	**.880 to .935	160 to 235 **90 to 120**	34.3 to 50.8 18.2 to 25.3	100 130
Federal Specification No. 20	**.880 to .935	235 to 385 **120 to 185**	50.8 to 83.4 25.3 to 39.9	100 130
Federal Specification No. 30	**.880 to .935	385 to 550 185 to 255	83.4 to 119 **39.9** to 55.1	100 130
Mineral Lard Cutting Oil: Federal Specification Grade 1		**140 to 190** 86 to 110	29.8 to 41 17.22 to 23	100 130
Federal Specification Grade 2		**190 to 220** 110 to 125	41 to 47.5 23 to 26.4	100 130
Petrolatum	.825	100 77	20.6 14.8	130 160
Turbine Lubricating Oil: Federal Specification (Penn Base)	.91 Average	400 to 440 **185 to 205**	86.6 to 95.2 39.9 to 44.3	100 130
VEGETABLE OILS: Castor Oil	.96 @ 68 F	1,200 to 1,500 **450** to 600	259.8 to 324.7 97.4 to 129.9	100 130
China Wood Oil	.943	**1,425** 580	308.5 125.5	69 100
Cocoanut Oil	.925	**140 to 148** 76 to 80	29.8 to 31.6 14.69 to 15.7	100 130
Corn Oil	.924	135 **54**	28.7 8.59	130 212
Cotton Seed Oil	.88 to .925	**176** 100	37.9 20.6	100 130
Linseed Oil, Raw	.925 to .939	**143** 93	30.5 18.94	100 130
Olive Oil	.912 to .918	**200** 115	43.2 24.1	100 130
Palm Oil	.924	**221** 125	47.8 26.4	100 130
Peanut Oil	.920	**195** 112	42 23.4	100 130
Rape Seed Oil	.919	**250** 145	54.1 31	100 130
Rosin Oil	.980	**1,500** 600	324.7 129.9	100 130

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VISCOSITY OF COMMON LIQUIDS

Liquid	*Sp Gr at 60 F	VISCOSITY		At F
		SSU	Centistokes	
Rosin (Wood)	1.09 Avg.	500 to **20,000** 1,000 to 50,000	108.2 to 4,400 216.4 to 11,000	200 190
Sesame Oil	.923	**184** 110	39.6 23	100 130
Soja Bean Oil	.927 to .98	**165** 96	35.4 19.64	100 130
Turpentine	.86 to .87	33 32.6	2.11 2.0	60 100
SUGAR, SYRUPS, MOLASSES, ETC.				
Corn Syrups	1.4 to 1.47	5,000 to 500,000 1,500 to 60,000	1,100 to 110,000 324.7 to 13,200	100 130
Glucose	1.35 to 1.44	35,000 to 100,000 4,000 to 11,000	7,700 to 22,000 880 to 2,420	100 150
Honey (Raw)		340	73.6	100
Molasses "A" (First)	1.40 to 1.46	1,300 to 23,000 700 to 8,000	281.1 to 5,070 151.5 to 1,760	100 130
Molasses "B" (Second)	1.43 to 1.48	6,400 to 60,000 3,000 to 15,000	1,410 to 13,200 660 to 3,300	100 130
Molasses "C" (Blackstrap or final)	1.46 to 1.49	17,000 to 250,000 6,000 to 75,000	2,630 to 5,500 1,320 to 16,500	100 130
Sucrose Solutions (Sugar Syrups):				
60 Brix	1.29	230 92	49.7 18.7	70 100
62 Brix	1.30	310 111	67.1 23.2	70 100
64 Brix	1.31	440 148	95.2 31.6	70 100
66 Brix	1.326	650 195	140.7 42.0	70 100
68 Brix	1.338	1,000 275	216.4 59 5	70 100
70 Brix	1.35	1,650 400	364 86.6	70 100
72 Brix	1.36	2,700 640	595 138.6	70 100
74 Brix	1.376	5,500 1,100	1,210 238	70 100
76 Brix	1.39	10,000 2,000	2,200 440	70 100

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VISCOSITY OF COMMON LIQUIDS

Liquid	*Sp Gr at 60 F	VISCOSITY		At F
		SSU	Centistokes	
TARS:				
Tar-Coke Oven	1.12+	**3,000 to 8,000**	600 to 1,760	71
		650 to 1,400	140.7 to 308	100
Tar-Gas House	1.16 to 1.30	**15,000 to 300,000**	3,300 to 66,000	70
		2,000 to 20,000	440 to 4,400	100
Road Tar:				
Grade RT-2	1.07+	**200 to 300**	43.2 to 64.9	122
		55 to 60	8.77 to 10.22	212
Grade RT-4	1.08+	**400 to 700**	86.6 to 154	122
		65 to 75	11.63 to 14.28	212
Grade RT-6	1.09+	**1,000 to 2,000**	216.4 to 440	122
		85 to 125	16.83 to 26.2	212
Grade RT-8	1.13+	**3,000 to 8,000**	660 to 1,760	122
		150 to 225	31.8 to 48.3	212
Grade RT-10	1.14+	**20,000 to 60,000**	4,400 to 13,200	122
		250 400	53.7 to 86.6	212
Grade RT-12	1.15+	114,000 to 456,000	25,000 to 75,000	122
		500 to 800	108.2 to 173.2	212
Pine Tar	1.06	2,500	559	100
		500	108.2	132
MISCELLANEOUS				
Corn Starch Solutions:				
22 Baumé	1.18	150	32.1	70
		130	27.5	100
24 Baumé	1.20	600	129.8	70
		440	95.2	100
25 Baumé	1.21	1400	303	70
		800	173.2	100
Ink—Printers	1.00 to 1.38	2,500 to 10,000	550 to 2,200	100
		1,100 to 3,000	238.1 to 660	130
Tallow	.918 Avg.	56	9.07	212
Milk	1.02 to 1.05		1.13	68
Varnish — Spar	.9	1425	313	68
		650	143	100
Water — Fresh	1.0		1.13	60
			.55	130

*Unless otherwise noted.

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Detailed requirements for fuel oils [a]

Grade of fuel oil [b]		Flash point, °F	Pour point, °F	Water and sediment, %	Carbon residue on 10% residuum, %	Ash, %	Distillation temperatures, °F			Viscosity								Gravity, °API
										Saybolt				Kinematic centistokes at—				
										Universal at 100° F		Furol at 122° F		100° F		122° F		
Number	Description						10% point	90% point	End point									
		min	*max*	*max*	*max*	*max*	*max*	*max*	*max*	*max*	*min*	*max*	*min*	*max*	*min*	*max*	*min*	*min*
1	Distillate oil intended for vaporizing pot-type burners and other burners requiring this grade. [c]	100 or legal	0	Trace	0.15	------	420	------	625	------	------	------	------	2.2	1.4	------	------	35
2	Distillate oil for general purpose domestic heating for use in burners not requiring No. 1.	100 or legal	[d] 20	0.10	0.35	------	(e)	675	------	40	------	------	------	(4.3)	------	------	------	26
4	Oil for burner installations not equipped with preheating facilities.	130 or legal	20	.50	------	0.10	------	------	------	125	45	------	------	(26.4)	(5.8)	------	------	------
5	Residual-type oil for burner installations equipped with preheating facilities.	130 or legal	------	1.00	------	.10	------	------	------	------	150	40	------	------	(32.1)	(81)	------	------
6	Oil for use in burners equipped with preheaters permitting a high-viscosity fuel.	150 or legal	------	[f] 2.00	------	------	------	------	------	------	------	300	45	------	------	(638)	(92)	------

[a] Recognizing the necessity for low-sulfur fuel oils used in connection with heat treatment, nonferrous metal, glass and ceramic furnaces, and other special uses, a sulfur requirement may be specified in accordance with the following table:

Grade of fuel oil	Sulfur, max, percent
No. 1	0.5
No. 2	1.0
Nos. 4, 5, and 6	No limit

Other sulfur limits may be specified only by mutual agreement between the buyer and seller.

[b] It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

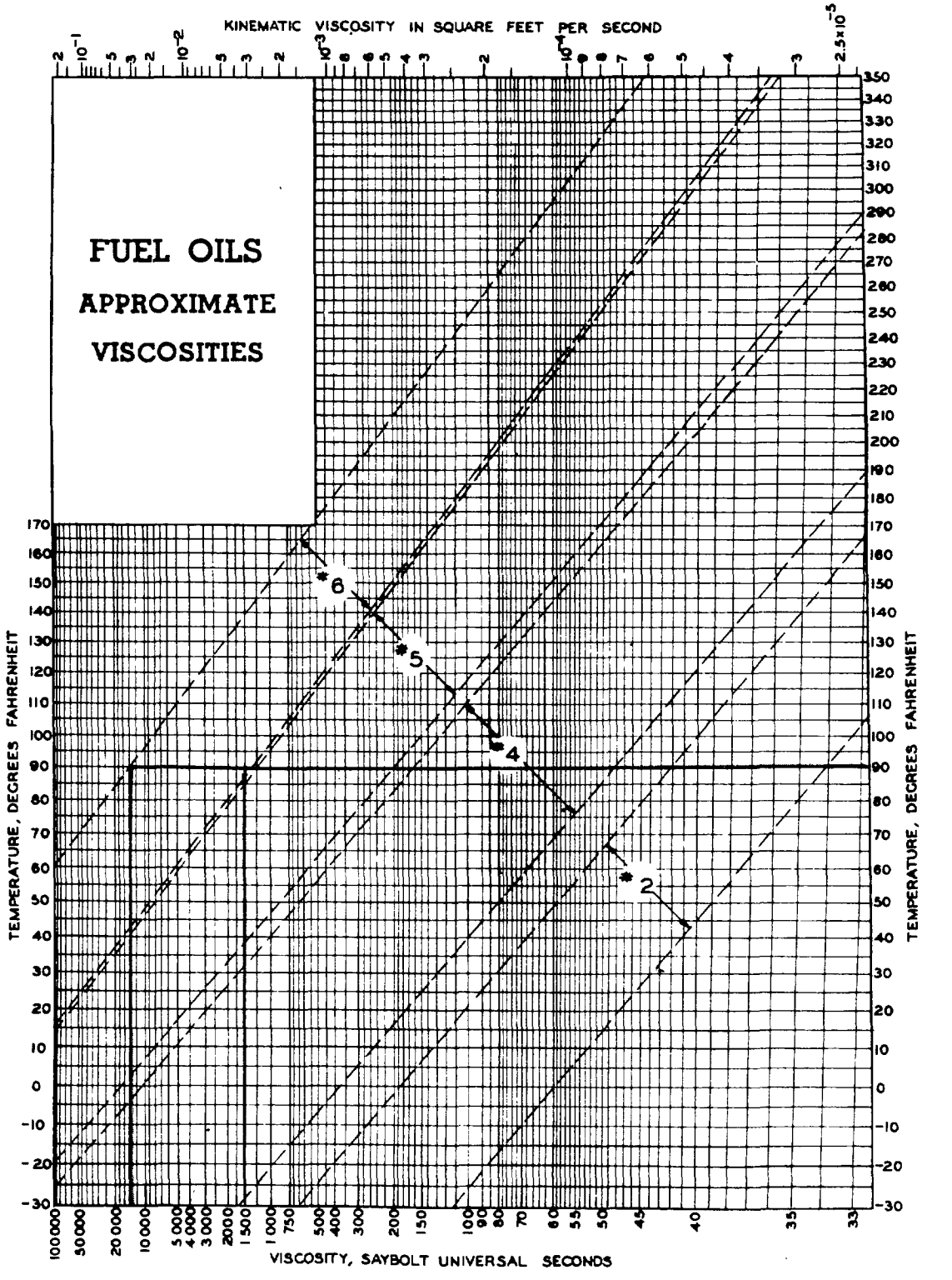
[c] No. 1 oil shall be tested for corrosion in accordance with par. 15 for 3 hours at 122° F. The exposed copper strip shall show no gray or black deposit.

[d] Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0° F under any conditions.

[e] The 10-percent point may be specified at 440° F. Maximum for use in other than atomizing burners.

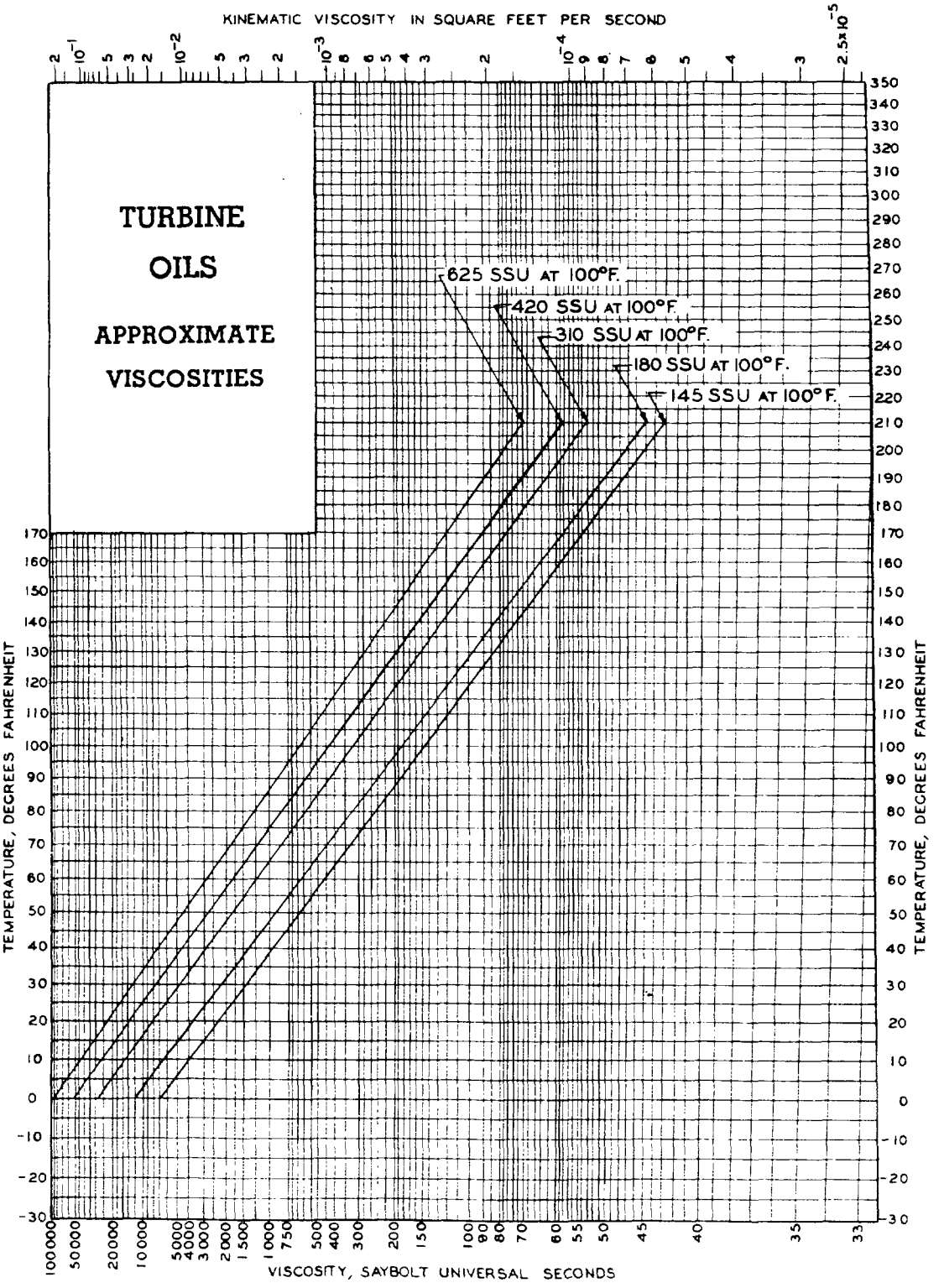
[f] The amount of water by distillation plus the sediment by extraction shall not exceed 2.00 percent. The amount of sediment by extraction shall not exceed 0.50 percent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 percent.

EFFECT OF TEMPERATURE ON FUEL OIL VISCOSITY



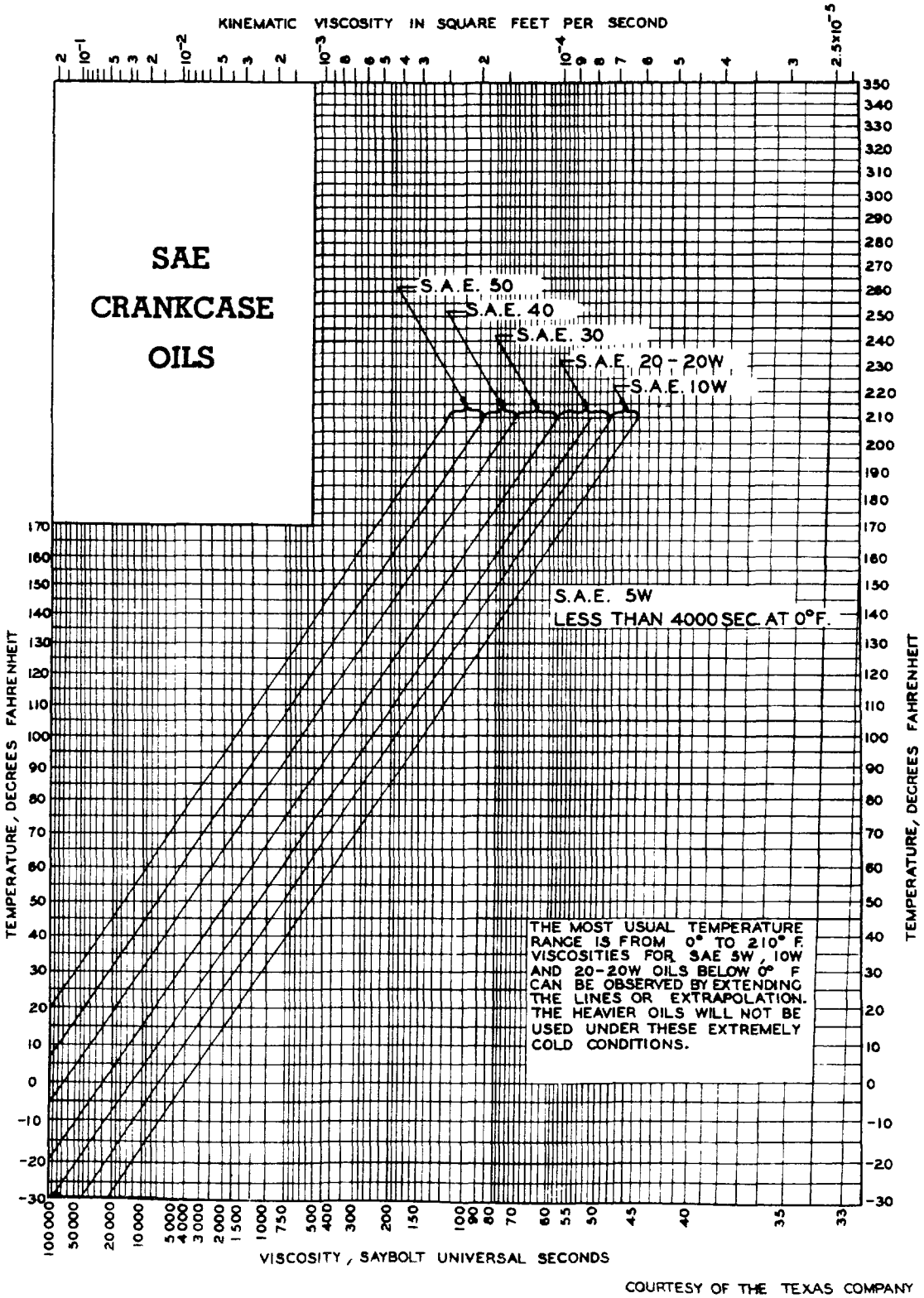
COURTESY OF THE TEXAS COMPANY

EFFECT OF TEMPERATURE ON TURBINE OIL VISCOSITY



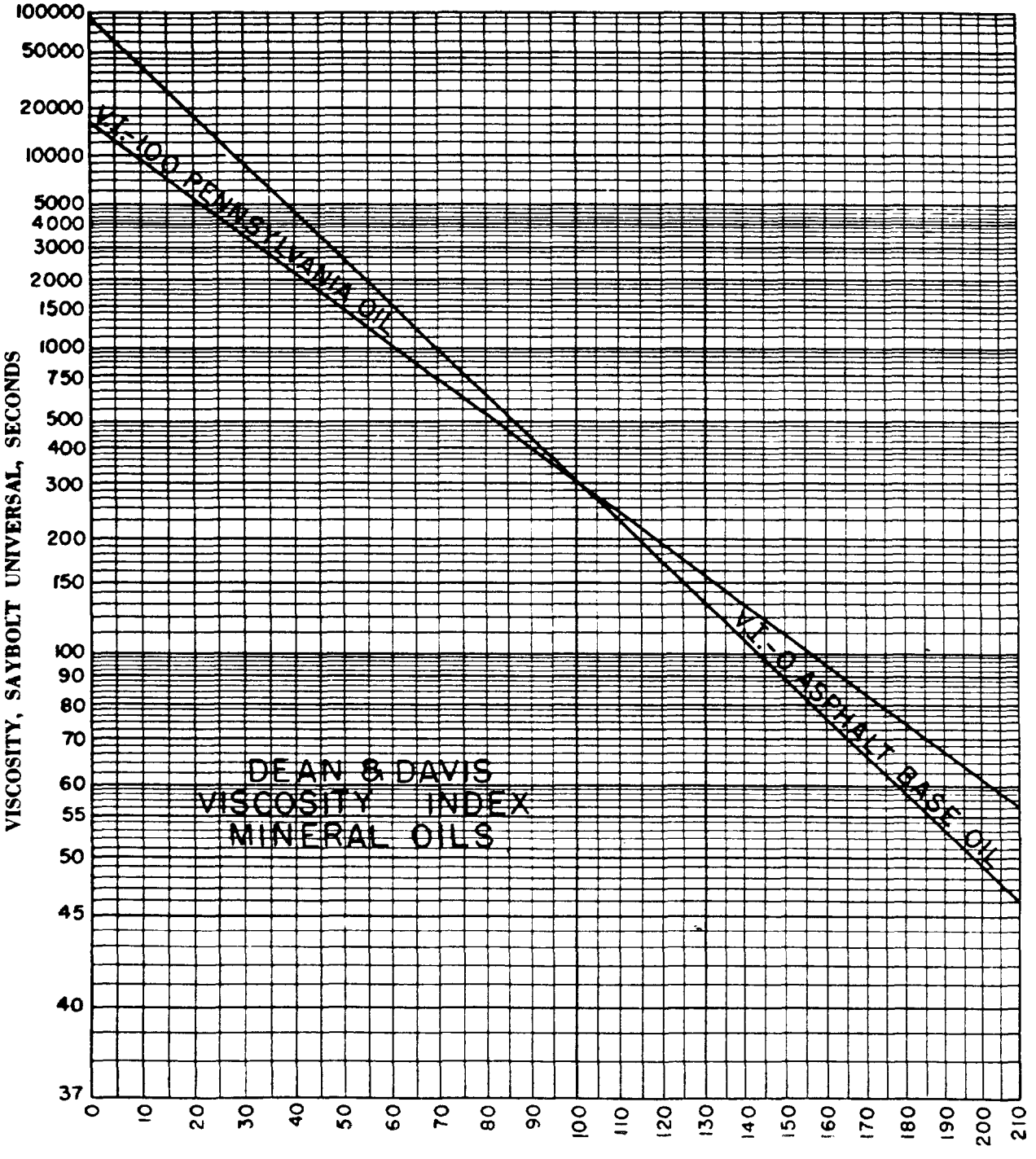
COURTESY OF THE TEXAS COMPANY

EFFECT OF TEMPERATURE ON SAE CRANKCASE OIL VISCOSITY



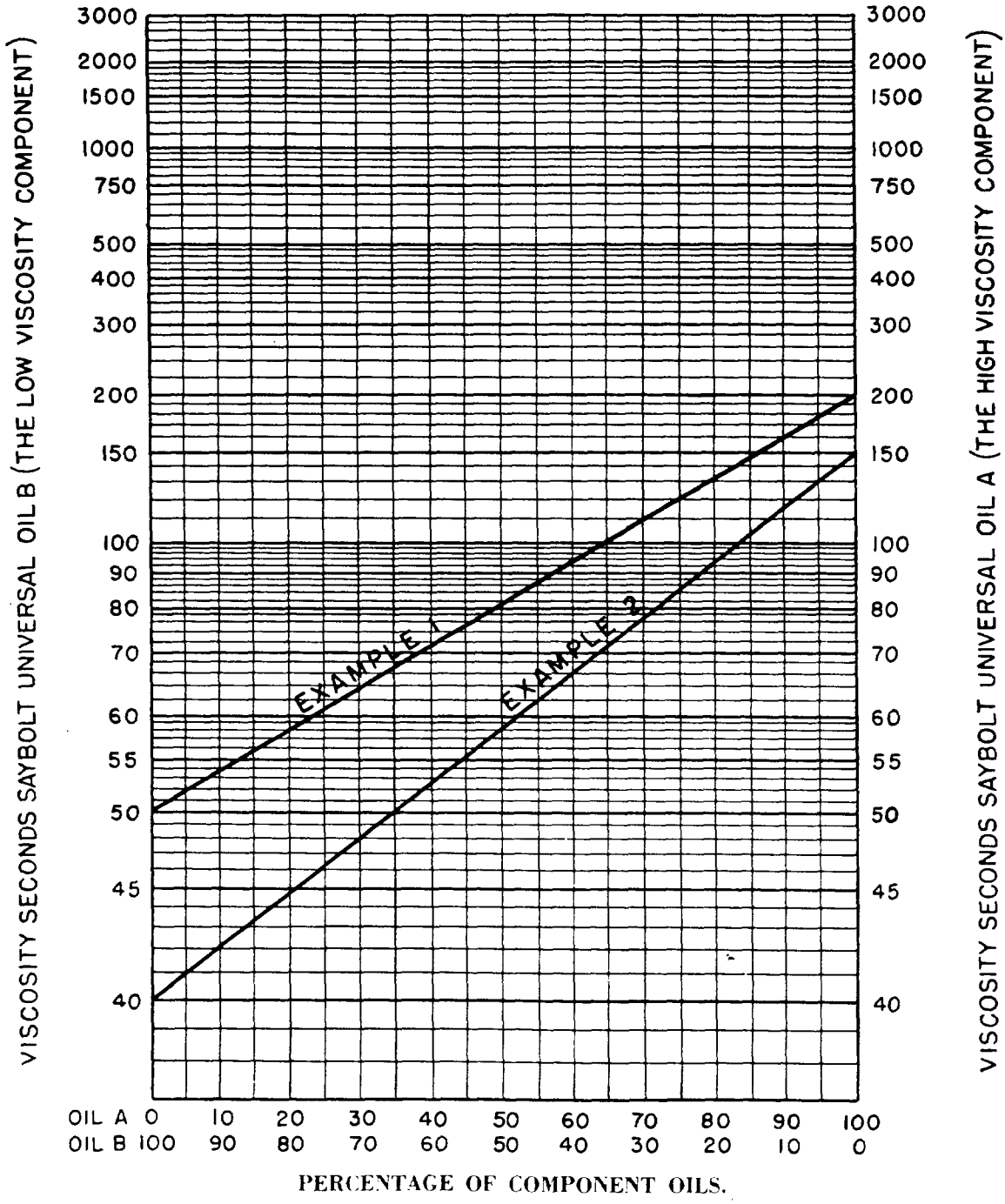
COURTESY OF THE TEXAS COMPANY

VISCOSITY INDEX – MINERAL OILS



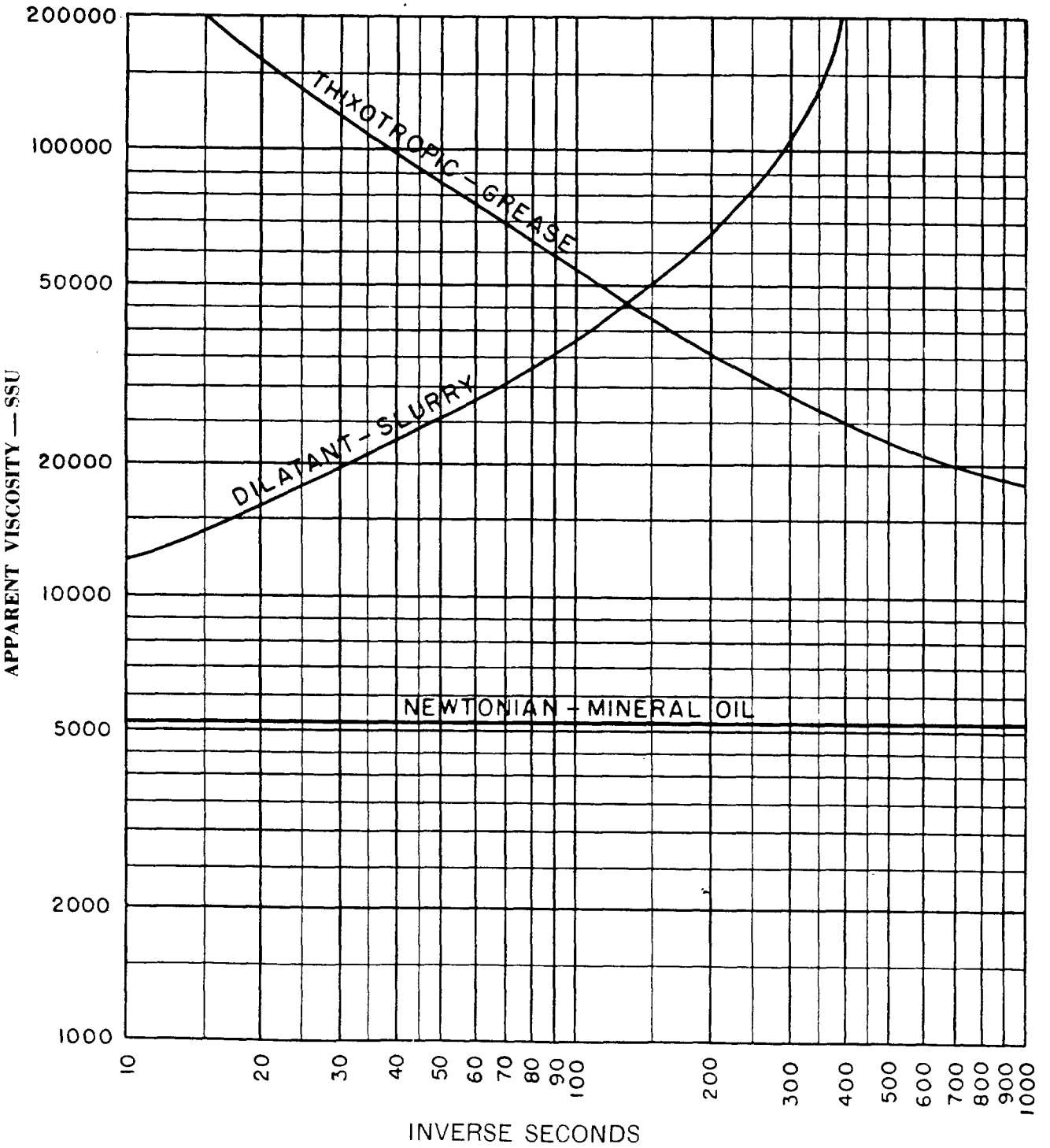
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VISCOSITY BLENDING CHART



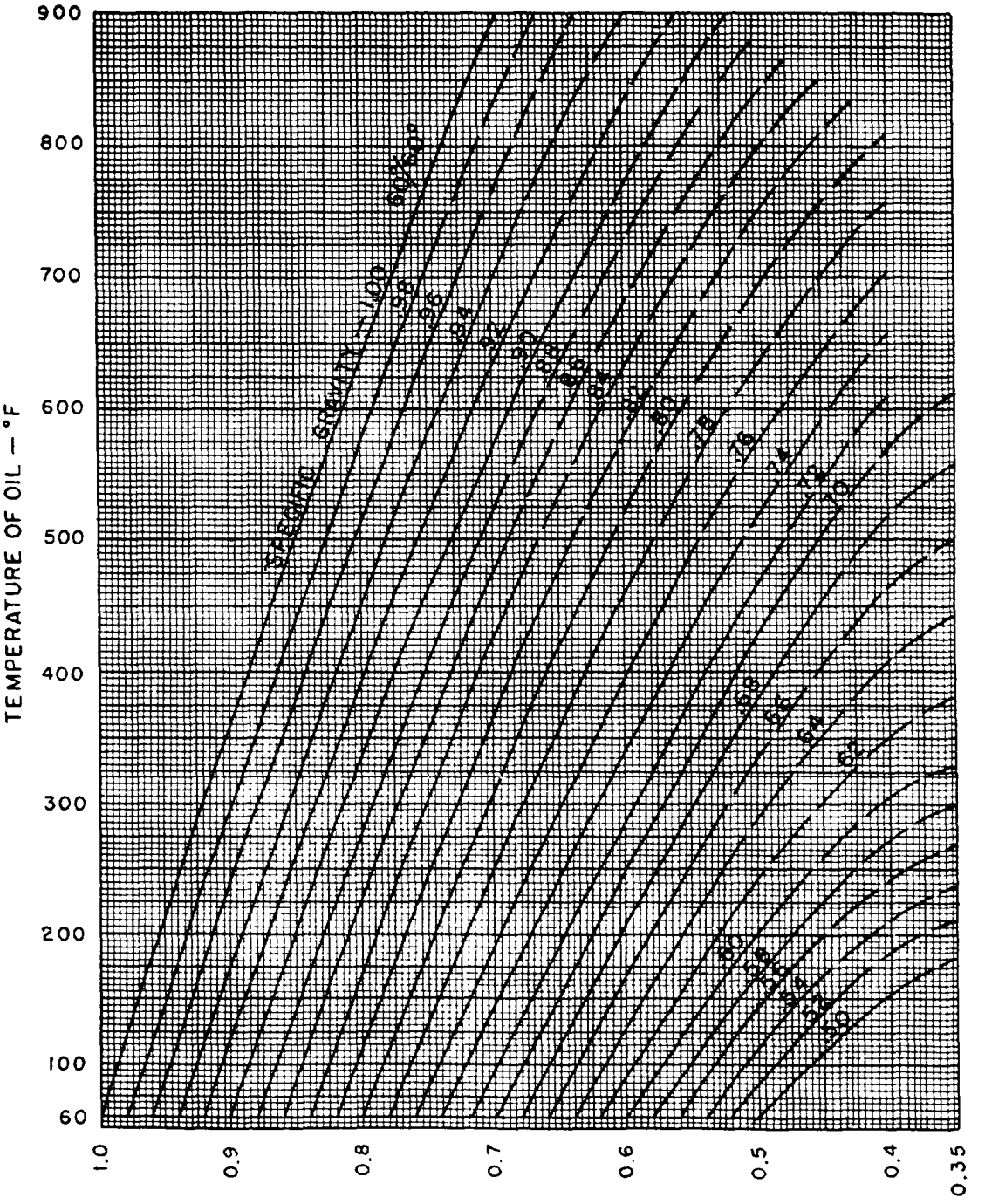
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TYPICAL SHEAR RATE CURVES



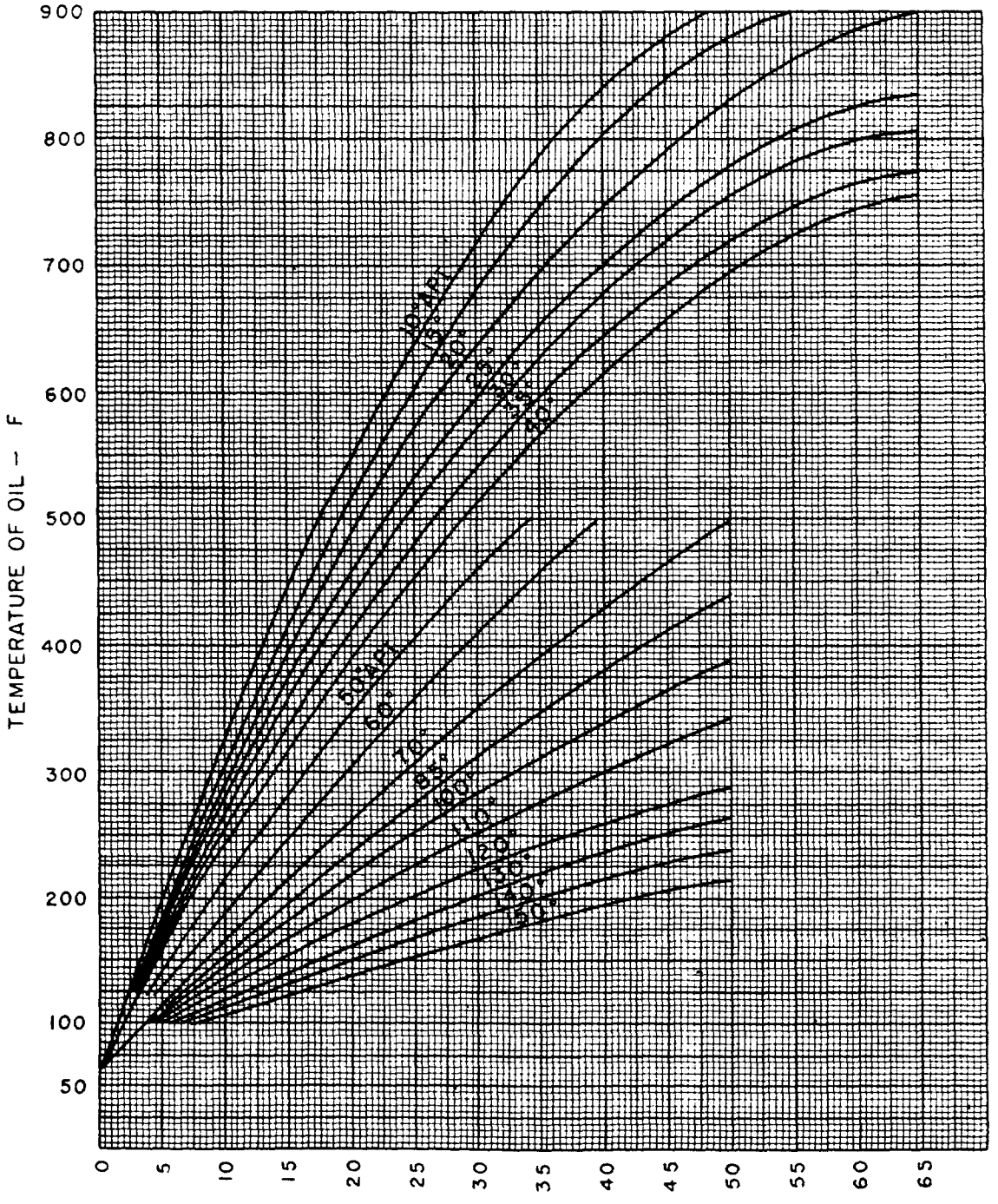
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PETROLEUM TEMPERATURE GRAVITY RELATIONS



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PETROLEUM TEMPERATURE VOLUME RELATIONS



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DIMENSIONS OF PIPE BY SCHEDULE NUMBERS FOR WALL THICKNESS

Instead of using the terms "Standard," "Extra Strong," and "Double Extra Strong," a system using Schedule Numbers is preferable in designating pipe wall thickness. The Schedule Numbers represent a rational relationship between wall thickness and pipe diameter. The Schedule Number is an approximation of the value of the expression:

$$\text{Schedule Number} = 1000 \times \frac{p}{s}$$

where: p = internal pressure in psig

s = allowable fiber stress in psi

Where the dimensions of the older designations have been retained, they are indicated as follows:

★ Standard; + Extra Strong. "Double Extra Strong" is not a part of the new standard and has not been included, although it is commercially available.

Properties of Wrought Iron and Wrought Steel Pipe

Nominal Pipe Size (in.)	Schedule No.	Outside Diameter (in.)	Wall Thickness (in.)	Inside Diameter (in.)	Inside Area (sq in.)	Weight Per Foot (lb)	Weight of Water (lb/ft)	External Surface (sq ft/ft)
1/8	40 ★	0.405	0.068	0.269	0.0568	0.244	0.025	0.106
	80 +	0.405	0.095	0.215	0.0364	0.314	0.016	
1/4	40 ★	0.540	0.088	0.364	0.1041	0.424	0.045	0.141
	80 +	0.540	0.119	0.302	0.0716	0.535	0.031	
3/8	40 ★	0.675	0.091	0.493	0.1910	0.567	0.083	0.177
	80 +	0.675	0.126	0.423	0.1405	0.738	0.061	
	40 ★	0.840	0.109	0.622	0.3040	0.850	0.132	
1/2	80 +	0.840	0.147	0.546	0.2340	1.087	0.101	0.220
	160	0.840	0.187	0.466	0.1706	1.310	0.074	
	40 ★	1.050	0.113	0.824	0.5330	1.130	0.214	
3/4	80 +	1.050	0.154	0.742	0.4330	1.473	0.1875	0.275
	160	1.050	0.218	0.614	0.2961	1.940	0.130	
	40 ★	1.315	0.133	1.049	0.8640	1.678	0.374	
1	80 +	1.315	0.179	0.957	0.7190	2.171	0.311	0.344
	160	1.315	0.250	0.815	0.5217	2.850	0.230	
	40 ★	1.660	0.140	1.380	1.495	2.272	0.649	
1-1/4	80 +	1.660	0.191	1.278	1.283	3.000	0.555	0.435
	160	1.660	0.250	1.160	1.057	3.764	0.460	

From American Standards Association "Wrought Steel and Wrought Iron Pipe," ASA B36.10 — 1950, and data of Walworth Co.

DIMENSIONS OF PIPE BY SCHEDULE NUMBERS FOR WALL THICKNESS

Properties of Wrought Iron and Wrought Steel Pipe								
Nominal Pipe Size (in.)	Schedule No.	Outside Diameter (in.)	Wall Thickness (in.)	Inside Diameter (in.)	Inside Area (sq in.)	Weight Per Foot (lb)	Weight of Water (lb/ft)	External Surface (sq ft/ft)
	40 *	1.900	0.145	1.610	2.036	2.717	0.882	0.497
1-1/2	80 +	1.900	0.200	1.500	1.767	3.631	0.765	
	160	1.900	0.281	1.337	1.405	4.862	0.610	
	40 *	2.375	0.154	2.067	3.355	3.652	1.454	0.622
2	80 +	2.375	0.218	1.939	2.953	5.022	1.28	
	160	2.375	0.343	1.689	2.240	7.450	0.97	
	40 *	2.875	0.203	2.469	4.788	5.79	2.073	0.753
2-1/2	80 +	2.875	0.276	2.323	4.238	7.66	1.83	
	160	2.875	0.375	2.125	3.547	10.01	1.54	
	40 *	3.500	0.216	3.068	7.393	7.57	3.201	0.916
3	80 +	3.500	0.300	2.900	6.605	10.25	2.86	
	160	3.500	0.437	2.626	5.416	14.30	2.35	
3-1/2	40 *	4.000	0.226	3.548	9.888	9.11	4.287	1.047
	80 +	4.000	0.318	3.364	8.891	12.51	3.847	
	40 *	4.500	0.237	4.026	12.73	10.79	5.516	1.178
4	80 +	4.500	0.337	3.826	11.50	14.98	4.98	
	120	4.500	0.437	3.626	10.33	19.00	4.47	
	160	4.500	0.531	3.438	9.28	22.51	4.02	
	40 *	5.563	0.528	5.047	20.01	14.62	8.674	1.456
5	80 +	5.563	0.375	4.813	18.19	20.78	7.875	
	120	5.563	0.500	4.563	16.35	27.04	7.100	
	160	5.563	0.625	4.313	14.61	32.96	6.33	
	40 *	6.625	0.280	6.065	28.89	18.97	12.52	1.734
6	80 +	6.625	0.432	5.761	26.07	28.57	11.29	
	120	6.625	0.562	5.501	23.77	36.40	10.3	
	160	6.625	0.718	5.189	21.15	45.30	9.16	

DIMENSIONS OF PIPE BY SCHEDULE NUMBERS FOR WALL THICKNESS

Properties of Wrought Iron and Wrought Steel Pipe								
Nominal Pipe Size (in.)	Schedule No.	Outside Diameter (in.)	Wall Thickness (in.)	Inside Diameter (in.)	Inside Area (sq in.)	Weight Per Foot (lb)	Weight of Water (lb/ft)	External Surface (sq ft/ft)
	20	8.625	0.250	8.125	51.8	22.36	22.47	2.258
	30	8.625	0.277	8.071	51.2	24.70	22.17	
	40 *	8.625	0.322	7.981	50.0	28.55	21.68	
	60	8.625	0.406	7.813	47.9	35.64	20.77	
8	80 +	8.625	0.500	7.625	45.6	43.39	19.79	
	100	8.625	0.593	7.439	43.5	50.90	18.83	
	120	8.625	0.718	7.189	40.6	60.70	17.60	
	140	8.625	0.812	7.001	38.5	67.80	16.68	
	160	8.625	0.906	6.813	36.4	74.70	15.80	
	20	10.750	0.250	10.250	82.5	28.03	35.76	2.817
	30	10.750	0.307	10.136	80.7	34.24	35.00	
	40 *	10.750	0.365	10.020	78.9	40.48	34.16	
	60 +	10.750	0.500	9.750	74.7	54.74	32.35	
10	80	10.750	0.593	9.564	71.8	64.40	31.13	
	100	10.750	0.718	9.314	68.1	77.00	29.53	
	120	10.750	0.843	9.064	64.5	89.20	28.00	
	140	10.750	1.000	8.750	60.1	104.20	26.06	
	160	10.750	1.125	8.500	56.7	116.00	24.59	
	20	12.750	0.250	12.250	117.9	33.4	51.07	3.338
	30	12.750	0.330	12.090	114.8	43.8	49.74	
	40	12.750	0.406	11.938	111.9	53.6	48.50	
	60	12.750	0.562	11.626	106.2	73.2	46.00	
12	80	12.750	0.687	11.376	101.6	88.5	44.04	
	100	12.750	0.843	11.064	96.1	107.2	41.66	
	120	12.750	1.000	10.750	90.8	125.5	39.33	
	140	12.750	1.125	10.500	86.6	140.0	37.52	
	160	12.750	1.312	10.126	80.5	160.3	34.89	

AMERICAN STANDARD STEEL PIPE FLANGES

Nominal Pipe Size	Flange Outside Diameter	Flange Thickness	Bolt Circle Diameter	Bore	Number of Bolts	Bolt Dia.
300 LB. STANDARD*						
1/2	3-3/4	9/16	2-5/8	0.90	4	1-/2
3/4	4-5/8	5/8	3-1/4	1.11	4	5/8
1	4-7/8	11/16	3-1/2	1.38	4	5/8
1-1/4	5-1/4	3/4	3-7/8	1.72	4	5/8
1-1/2	6-1/8	13/16	4-1/2	1.96	4	3/4
2	6-1/2	7/8	5	2.46	8	5/8
2-1/2	7-1/2	1	5-7/8	2.97	8	3/4
3	8-1/4	1-1/8	6-5/8	3.60	8	3/4
4	10	1-1/4	7-7/8	4.60	8	3/4
5	11	1-3/8	9-1/4	5.66	8	3/4
6	12-1/2	1-7/16	10-5/8	6.72	12	3/4
8	15	1-5/8	13	8.78	12	7/8
10	17-1/2	1-7/8	15-1/4	10.91	16	1
12	20-1/2	2	17-3/4	12.90	16	1-1/8
14 O.D.	23	2-1/8	20-1/4	14.17	20	1-1/8
16 O.D.	25-1/2	2-1/4	22-1/2	16.19	20	1-1/4
18 O.D.	28	2-3/8	24-3/4	18.19	24	1-1/4
20 O.D.	30-1/2	2-1/2	27	20.19	24	1-1/4
24 O.D.	36	2-3/4	32	24.19	24	1-1/2
600 LB. STANDARD**						
1/2	3-3/4	9/16	2-5/8	0.90	4	1/2
3/4	4-5/8	5/8	3-1/4	1.11	4	5/8
1	4-7/8	11/16	3-1/2	1.38	4	5/8
1-1/4	5-1/4	13/16	3-7/8	1.72	4	5/8
1-1/2	6-1/8	7/8	4-1/2	1.96	4	3/4
2	6-1/2	1	5	2.46	8	5/8
2-1/2	7-1/2	1-1/8	5-7/8	2.97	8	3/4
3	8-1/4	1-1/4	6-5/8	3.60	8	3/4
4	10-3/4	1-1/2	8-1/2	4.60	8	7/8
5	13	1-3/4	10-1/2	5.66	8	1
6	14	1-7/8	11-1/2	6.72	12	1
8	16-1/2	2-3/16	13-3/4	8.78	12	1-1/8
10	20	2-1/2	17	10.91	16	1-1/4
12	22	2-5/8	19-1/4	12.90	20	1-1/4
14 O.D.	23-3/4	2-3/4	20-3/4	14.17	20	1-3/8
16 O.D.	27	3	23-3/4	16.19	20	1-1/2
18 O.D.	29-1/4	3-1/4	25-3/4	18.19	20	1-5/8
20 O.D.	32	3-1/2	28-1/2	20.19	24	1-5/8
24 O.D.	37	4	33	24.19	24	1-7/8

* Flange Thickness Includes 1/16" Raised Face.
** Flange Thickness Does Not Include 1/4" Raised Face.
From American Standard Association "Steel Pipe Flanges and Flanged Fittings", B16e1939.

AMERICAN STANDARD STEEL PIPE FLANGES

Nominal Pipe Size	Flange Outside Diameter	Flange Thickness	Bolt Circle Diameter	Bore	Number of Bolts	Bolt Dia.
900 LB. STANDARD*						
1/2	4-3/4	7/8	3-1/4	0.90	4	3/4
3/4	5-1/8	1	3-1/2	1.11	4	3/4
1	5-7/8	1-1/8	4	1.38	4	7/8
1-1/4	6-1/4	1-1/8	4-3/8	1.72	4	7/8
1-1/2	7	1-1/4	4-7/8	1.96	4	1
2	8-1/2	1-1/2	6-1/2	2.46	8	7/8
2-1/2	9-5/8	1-5/8	7-1/2	2.97	8	1
3	9-1/2	1-1/2	7-1/2	3.60	8	7/8
4	11-1/2	1-3/4	9-1/4	4.60	8	1-1/8
5	13-3/4	2	11	5.66	8	1-1/4
6	15	2-3/16	12-1/2	6.72	12	1-1/8
8	18-1/2	2-1/2	15-1/2	8.78	12	1-3/8
10	21-1/2	2-3/4	18-1/2	10.91	16	1-3/8
12	24	3-1/8	21	12.90	20	1-3/8
14 O.D.	25-1/4	3-3/8	22	14.17	20	1-1/2
16 O.D.	27-3/4	3-1/2	24-1/4	16.19	20	1-5/8
18 O.D.	31	4	27	18.19	20	1-7/8
20 O.D.	33-3/4	4-1/4	29-1/2	20.19	20	2
24 O.D.	41	5-1/2	35-1/2	24.19	20	2-1/2
1500 LB. STANDARD*						
1/2	4-3/4	7/8	3-1/4	0.90	4	3/4
3/4	5-1/8	1	3-1/2	1.11	4	3/4
1	5-7/8	1-1/8	4	1.38	4	7/8
1-1/4	6-1/4	1-1/8	4-3/8	1.72	4	7/8
1-1/2	7	1-1/4	4-7/8	1.96	4	1
2	8-1/2	1-1/2	6-1/2	2.46	8	7/8
2-1/2	9-5/8	1-5/8	7-1/2	2.97	8	1
3	10-1/2	1-7/8	8	3.60	8	1-1/8
4	12-1/4	2-1/8	9-1/2	4.60	8	1-1/4
5	14-3/4	2-7/8	11-1/2	5.66	8	1-1/2
6	15-1/2	3-1/4	12-1/2	6.72	12	1-3/8
8	19	3-5/8	15-1/2	8.78	12	1-5/8
10	23	4-1/4	19	10.91	12	1-7/8
12	26-1/2	4-7/8	22-1/2	12.90	16	2
14 O.D.	29-1/2	5-1/4	25	14.17	16	2-1/4
16 O.D.	32-1/2	5-3/4	27-3/4	16.19	16	2-1/2
18 O.D.	36	6-3/8	30-1/2	18.19	16	2-3/4
20 O.D.	38-3/4	7	32-3/4	20.19	16	3
24 O.D.	46	8	39	24.19	16	3-1/2

* Flange Thickness Does Not Include 1/4" Raised Face.
From American Standard Association "Steel Pipe Flanges and Flanged Fittings", B16e-1939.

AMERICAN STANDARD CAST IRON PIPE FLANGES

Nominal Pipe Size	Flange Outside Diameter	Flange Thickness	Bolt Circle Diameter	Number of Bolts	Bolt Dia.
125 LB. STANDARD					
1	4-1/4	7/16	3-1/8	4	1/2
1-1/4	4-5/8	1/2	3-1/2	4	1/2
1-1/2	5	9/16	3-7/8	4	1/2
2	6	5/8	4-3/4	4	5/8
2-1/2	7	11/16	5-1/2	4	5/8
3	7-1/2	3/4	6	4	5/8
4	9	15/16	7-1/2	8	5/8
5	10	15/16	8-1/2	8	3/4
6	11	1	9-1/2	8	3/4
8	13-1/2	1-1/8	11-3/4	8	3/4
10	16	1-3/16	14-1/4	12	7/8
12	19	1- 1/4	17	12	7/8
14 O.D.	21	1-3/8	18-3/4	12	1
16 O.D.	23-1/2	1-7/16	21-1/4	16	1
18 O.D.	25	1-9/16	22-3/4	16	1-1/8
20 O.D.	27-1/2	1-11/16	25	20	1-1/8
24 O.D.	32	1-7/8	29-1/2	20	1-1/4
30 O.D.	38-3/4	2-1/8	36	28	1-1/4
36 O.D.	46	2-3/8	42-3/4	32	1-1/2
42 O.D.	53	2-5/8	49-1/2	36	1-1/2
48 O.D.	59-1/2	2-3/4	56	44	1-1/2
54 O.D.	66-1/4	3	62-3/4	44	1-3/4
60 O.D.	73	3-1/8	69-1/4	52	1-3/4

From American Standard Association "Cast Iron Pipe Flanges and Flanged Fittings," B16.1 — 1948.

AMERICAN STANDARD CAST IRON PIPE FLANGES

Nominal Pipe Size Inches	Flange Outside Diameter	Flange Thickness	Bolt Circle Diameter	Number of Bolts	Bolt Dia.
250 LB. STANDARD					
1	4-7/8	11/16	3-1/2	4	5/8
1-1/4	5-1/4	3/4	3-7/8	4	5/8
1-1/2	6-1/8	13/16	4-1/2	4	3/4
2	6-1/2	7/8	5	8	5/8
2-1/2	7-1/2	1	5-7/8	8	3/4
3	8-1/4	1-1/8	6-5/8	8	3/4
3-1/2	9	1-3/16	7-1/4	8	3/4
4	10	1-1/4	7-7/8	8	3/4
5	11	1-3/8	9-1/4	8	3/4
6	12-1/2	1-7/16	10-5/8	12	3/4
8	15	1-5/8	13	12	7/8
10	17-1/2	1-7/8	15-1/4	16	1
12	20-1/2	2	17-3/4	16	1-1/8
14 O.D.	23	2-1/8	20-1/4	20	1-1/8
16 O.D.	25-1/2	2-1/4	22-1/2	20	1-1/4
18 O.D.	28	2-3/8	24-3/4	24	1-1/4
20 O.D.	30-1/2	2-1/2	27	24	1-1/4
24 O.D.	36	2-3/4	32	24	1-1/2
30 O.D.	43	3	39-1/4	28	1-3/4
36 O.D.	50	3-3/8	46	32	2
42 O.D.	57	3-11/16	52-3/4	36	2
48 O.D.	65	4	60-3/4	40	2

Flange Thickness Includes 1/16" Raised Face.

From American Standard Association "Cast Iron Pipe Flanges and Flanged Fittings", B16b-1944.

PRESSURE LOSS DUE TO PIPE FRICTION FOR STANDARD STEEL PIPE

The pressure loss curves reproduced in the following pages have been compiled from various references. Each chart covers a single pipe size and is based on standard pipe and sp gr. of 1. At the top of each chart is given the velocity in feet per second of the corresponding flow of gallons per minute given at the bottom of the chart. At the left of each chart is given the pressure loss in pounds per square inch, and diagonally on the face of the chart are given reference curves for various viscosities.

For example, see the chart on page 18. As shown by the dotted lines; to find pressure loss for ½" pipe for .5 gallons per minute of a liquid with 500 SSU viscosity, enter the chart at the bottom at .5 gallons per minute and note at the intersection of the .5 gallons per minute vertical line with the 500 SSU diagonal reference line a reading can be found at the left margin of 10 lb/□" per 100 ft. of length At the top of the chart directly above the .5 gallons per minute, a reading of approximately .52 ft. per second can be found, which is the velocity for .5 gallons per minute flow in a ½" pipe. All of the other charts for other pipe sizes are in the same form and may be read in the same manner.

In arriving at the pressure loss in any piping system, an addition must be made for the loss in fittings and valves. The factors in the table below will give an approximate value which can be used as an allowance for the fittings and valves as indicated.

As no pipe friction loss data can be considered exact for other than the particular conditions from which it was compiled, any pipe friction data, including that furnished herewith, must be used only as a reasonably accurate approximation.

FRICTION LOSS IN STANDARD VALVES AND FITTINGS

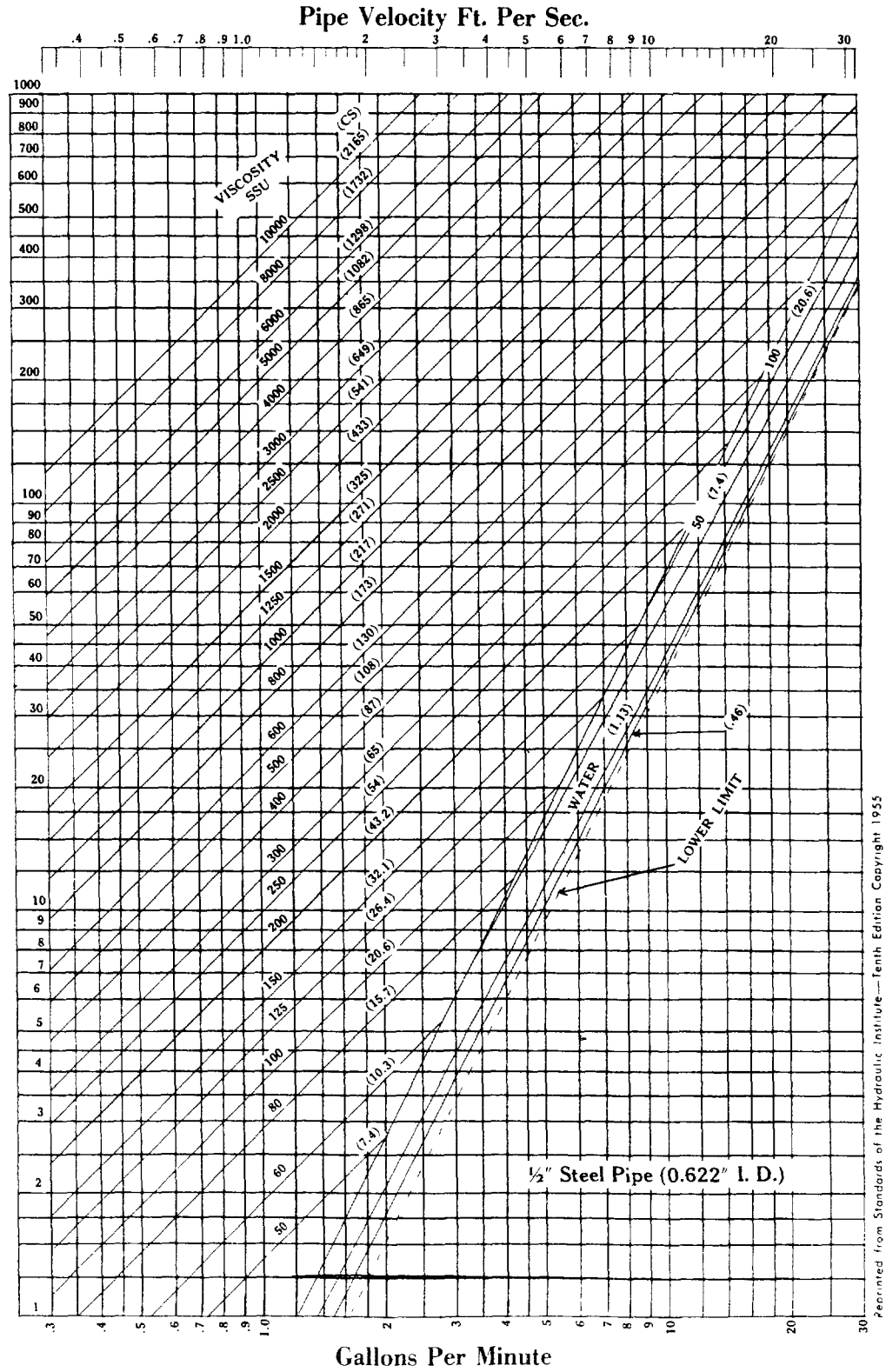
TABLE GIVES EQUIVALENT LENGTHS OF STRAIGHT PIPE IN FEET

TYPE OF FITTING	NOMINAL PIPE DIAMETER												
	½	¾	1	1¼	1½	2	2½	3	4	5	6	8	10
Gate Valve (open)	.35	.50	.60	.80	1.0	1.2	1.4	1.7	2.3	2.8	3.5	4.5	5.7
Globe Valve (open)	17	22	27	38	44	53	68	80	120	140	170	220	280
Angle Valve (open)	8	12	14	18	22	28	33	42	53	70	84	120	140
St'd. Elbow	1.5	2.2	2.7	3.6	4.5	5.2	6.5	8.0	11.0	14	16	21	26
Med. Sweep Elbow	1.3	1.8	2.3	3.0	3.6	4.6	5.5	7.0	9.0	12.0	14.0	18.0	22.0
Long Sweep Elbow	1.0	1.3	1.7	2.3	2.8	3.5	4.3	5.2	7.0	9.0	11.0	14.0	17.0
Tee (straight thru)	1.0	1.3	1.7	2.3	2.8	3.5	4.3	5.2	7.0	9.0	11.0	14.0	17.0
Tee (rt. angle flow)	3.2	4.5	5.7	7.5	9.0	12.0	14.0	16.0	22.0	27.0	33.0	43.0	53.0
Return Bend	3.5	5.0	6.0	8.5	10.0	13.0	15.0	18.0	24.0	30.0	37.0	50.0	63.0

FRICTION LOSS MODULUS FOR 100 FEET OF PIPE

Loss—Lbs. Per Sq. In. = Modulus x Specific Gravity

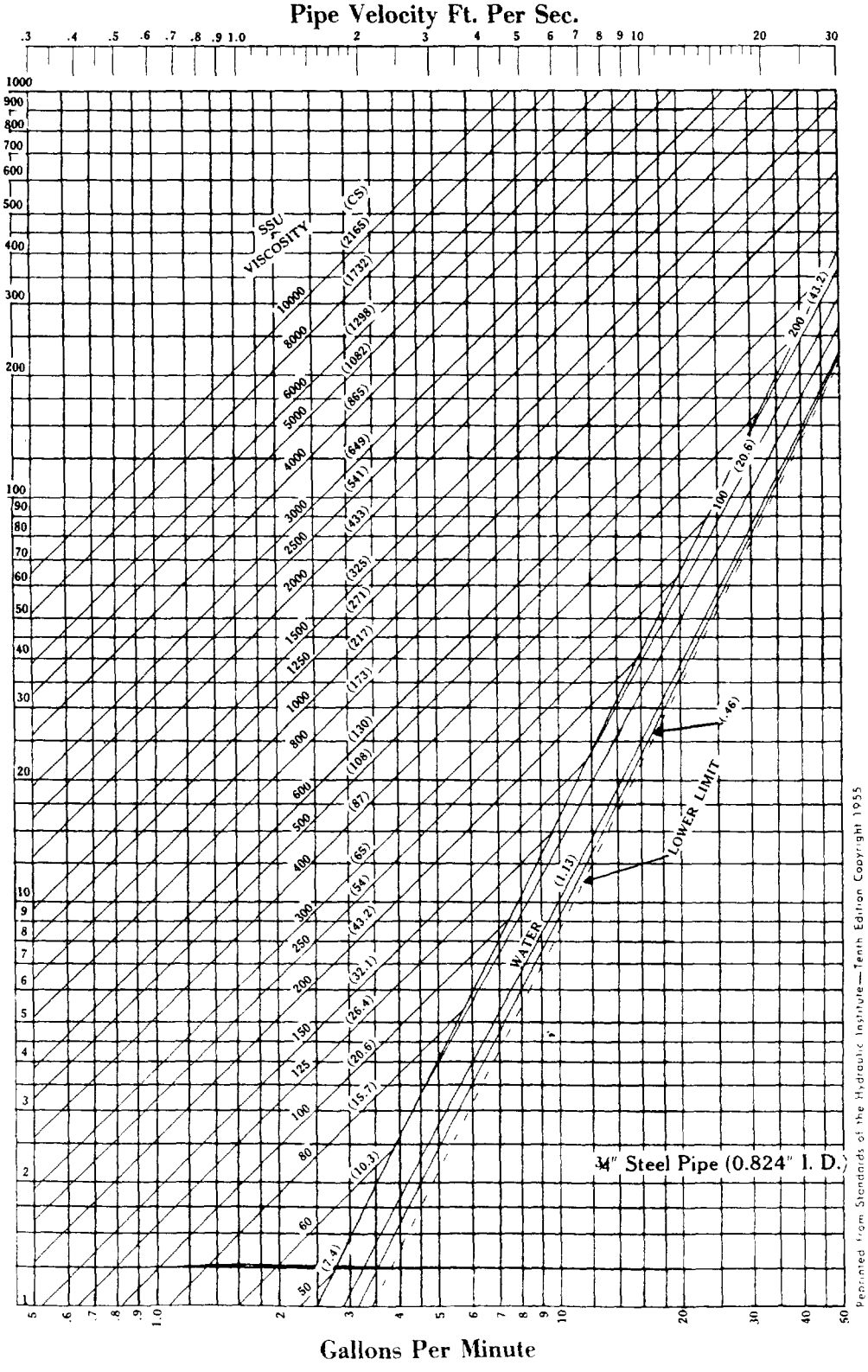
Loss—Feet of Liquid = Modulus x 2.31



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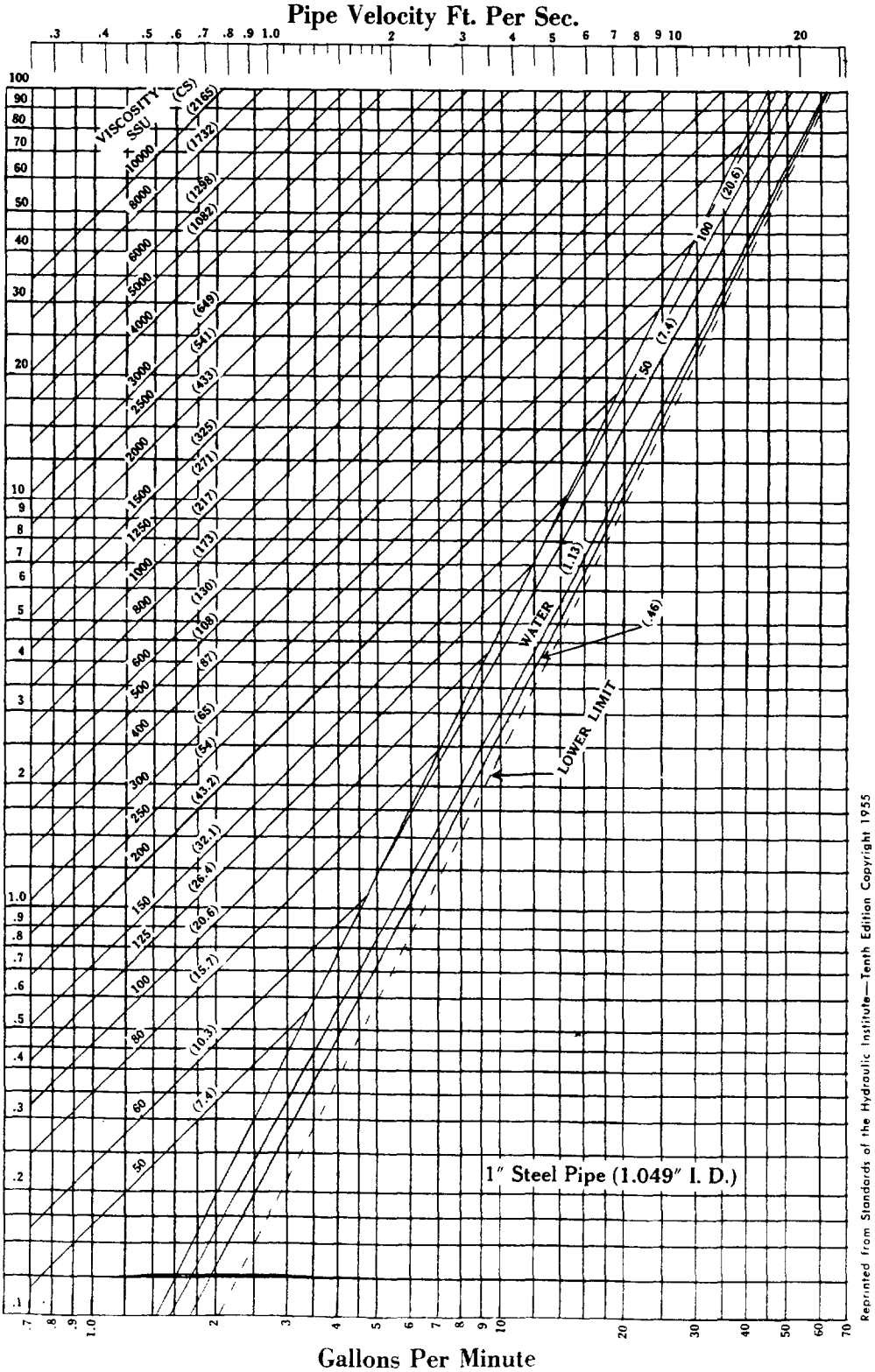
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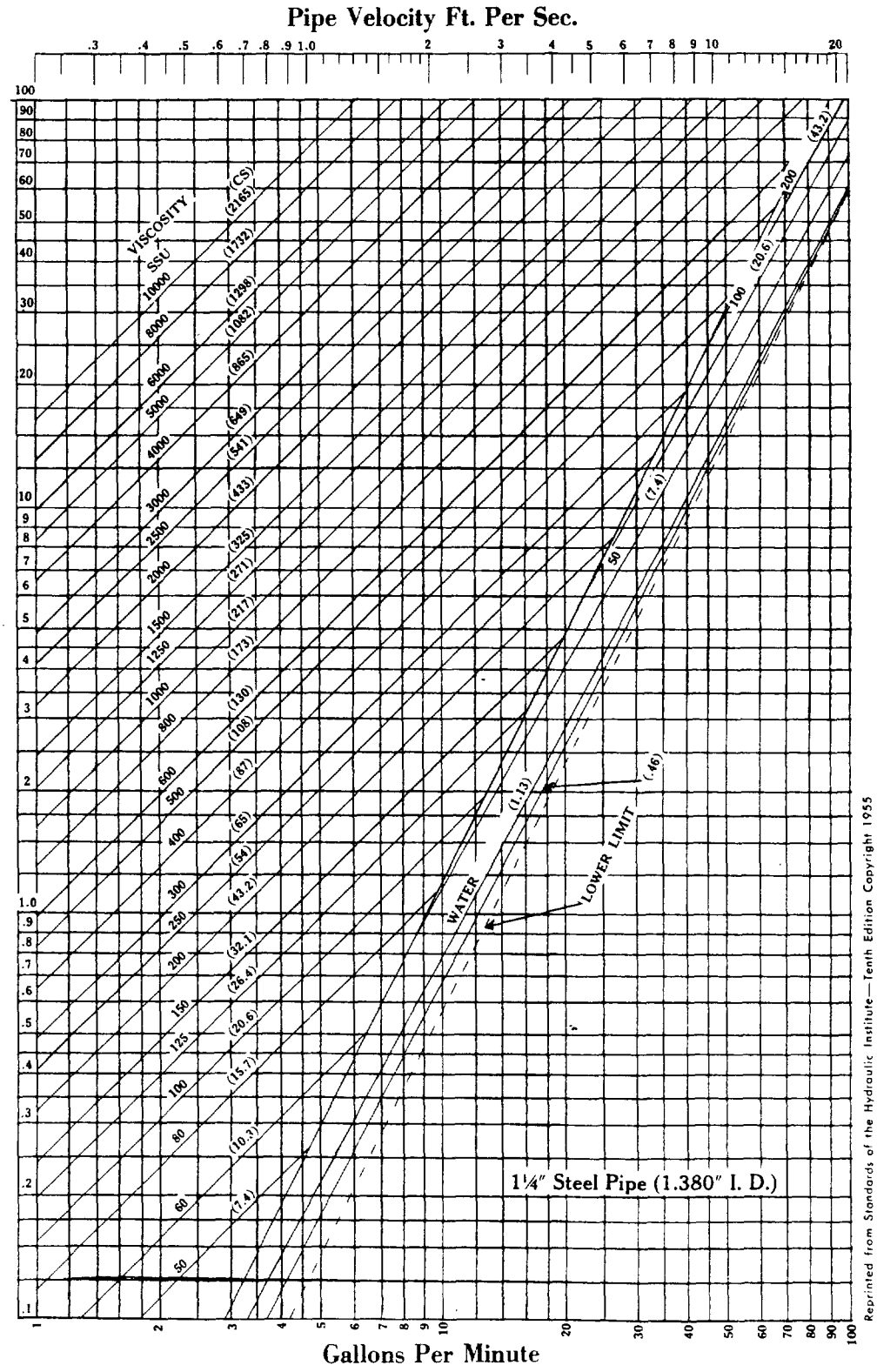
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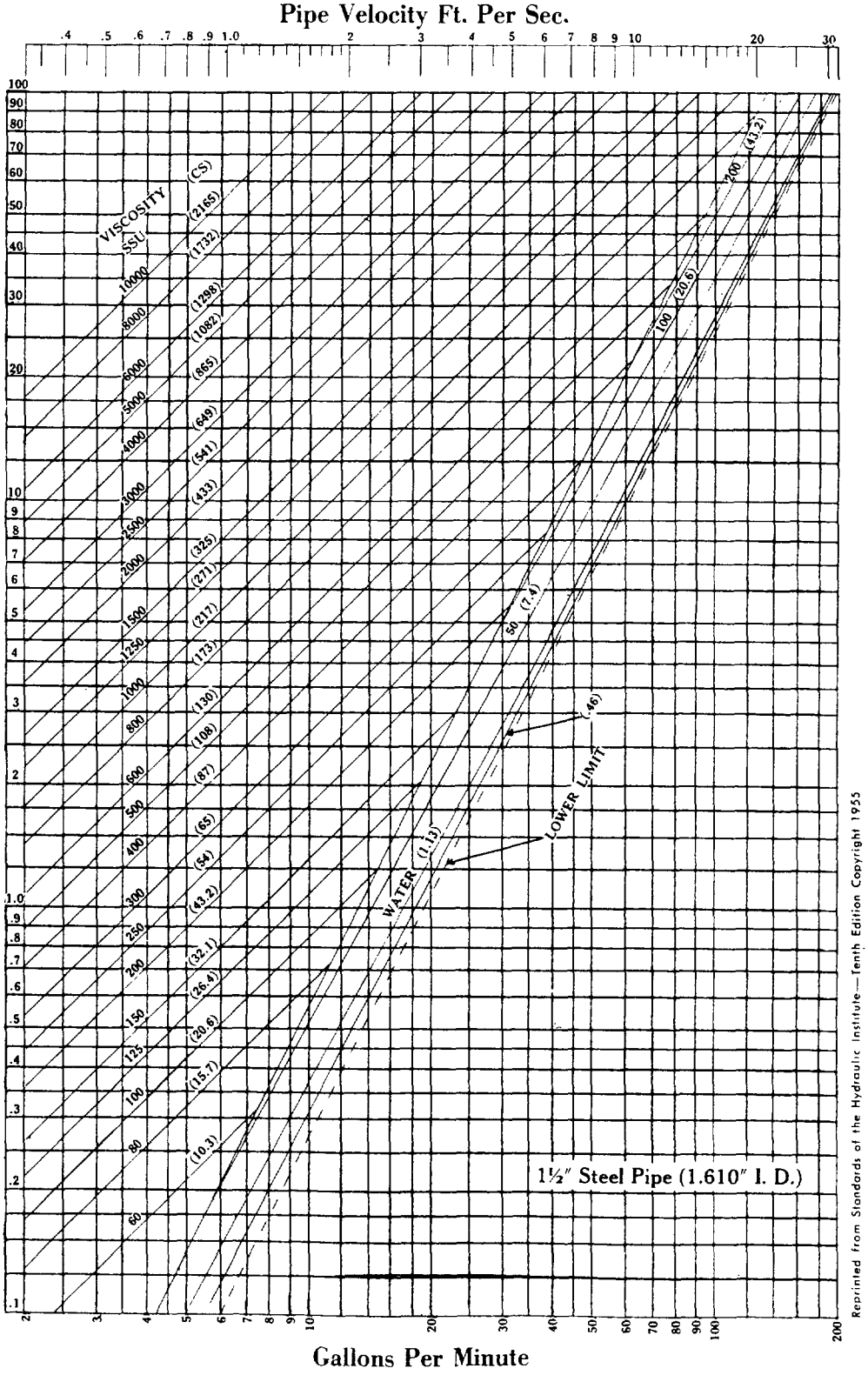
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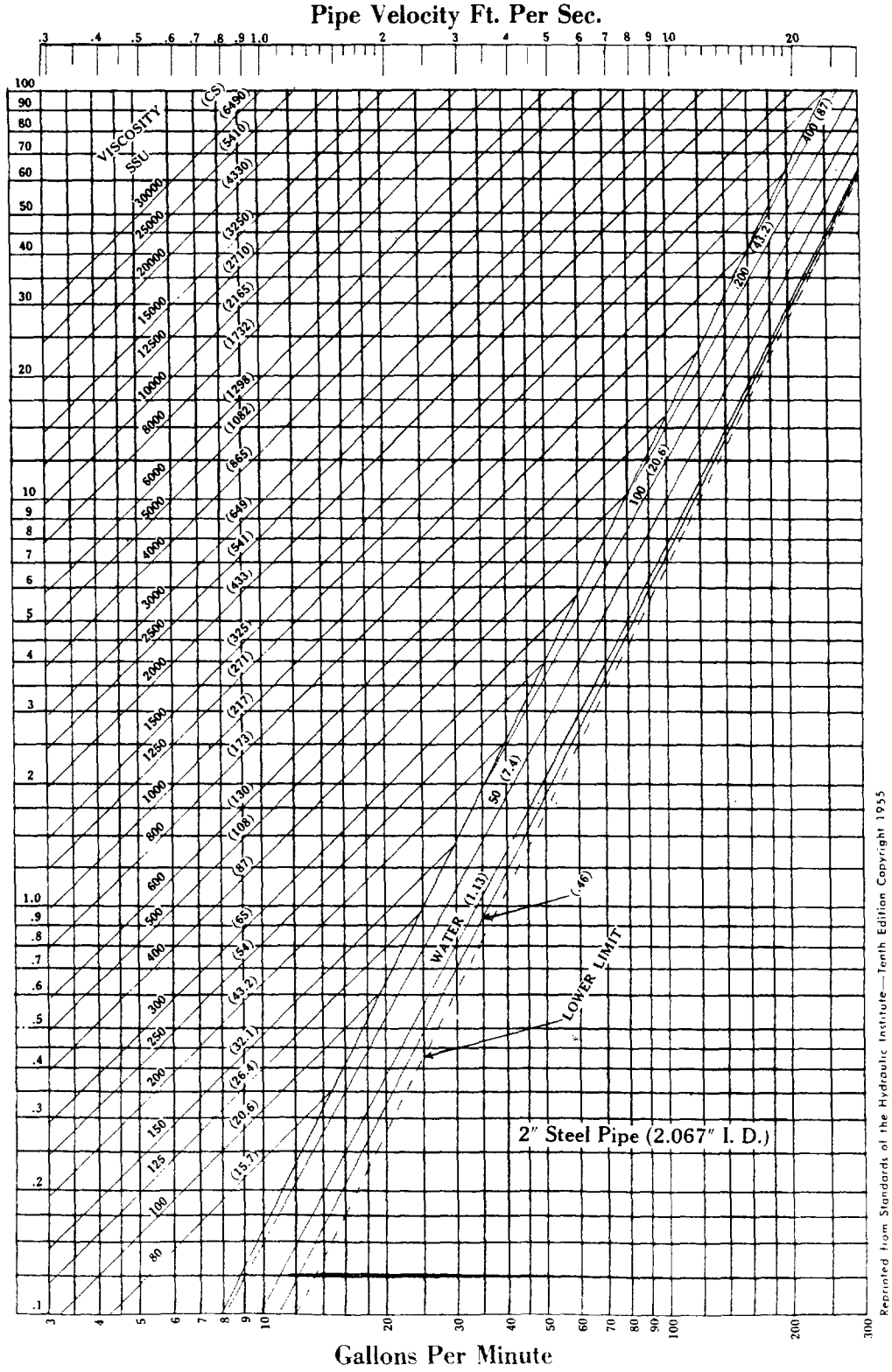


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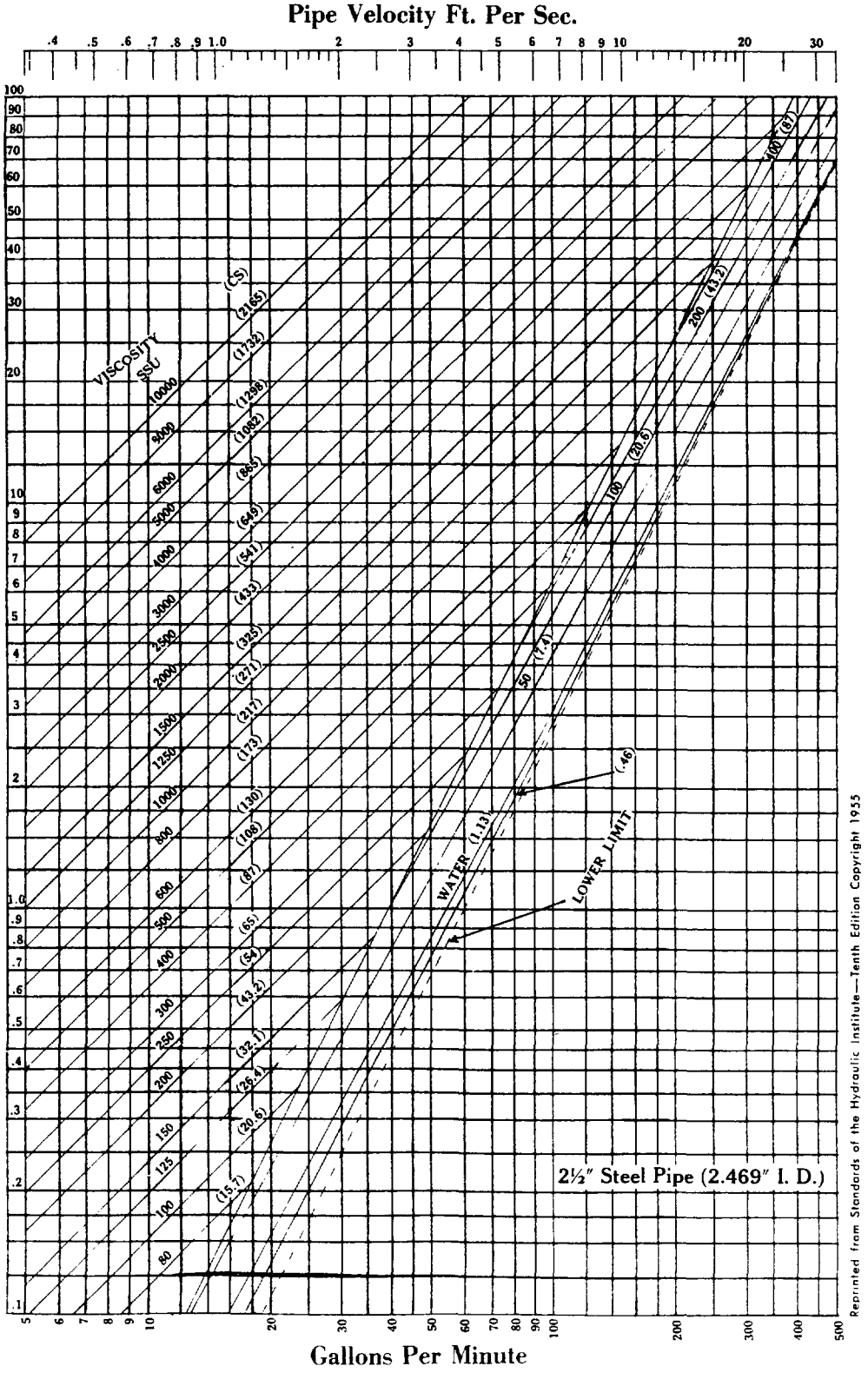


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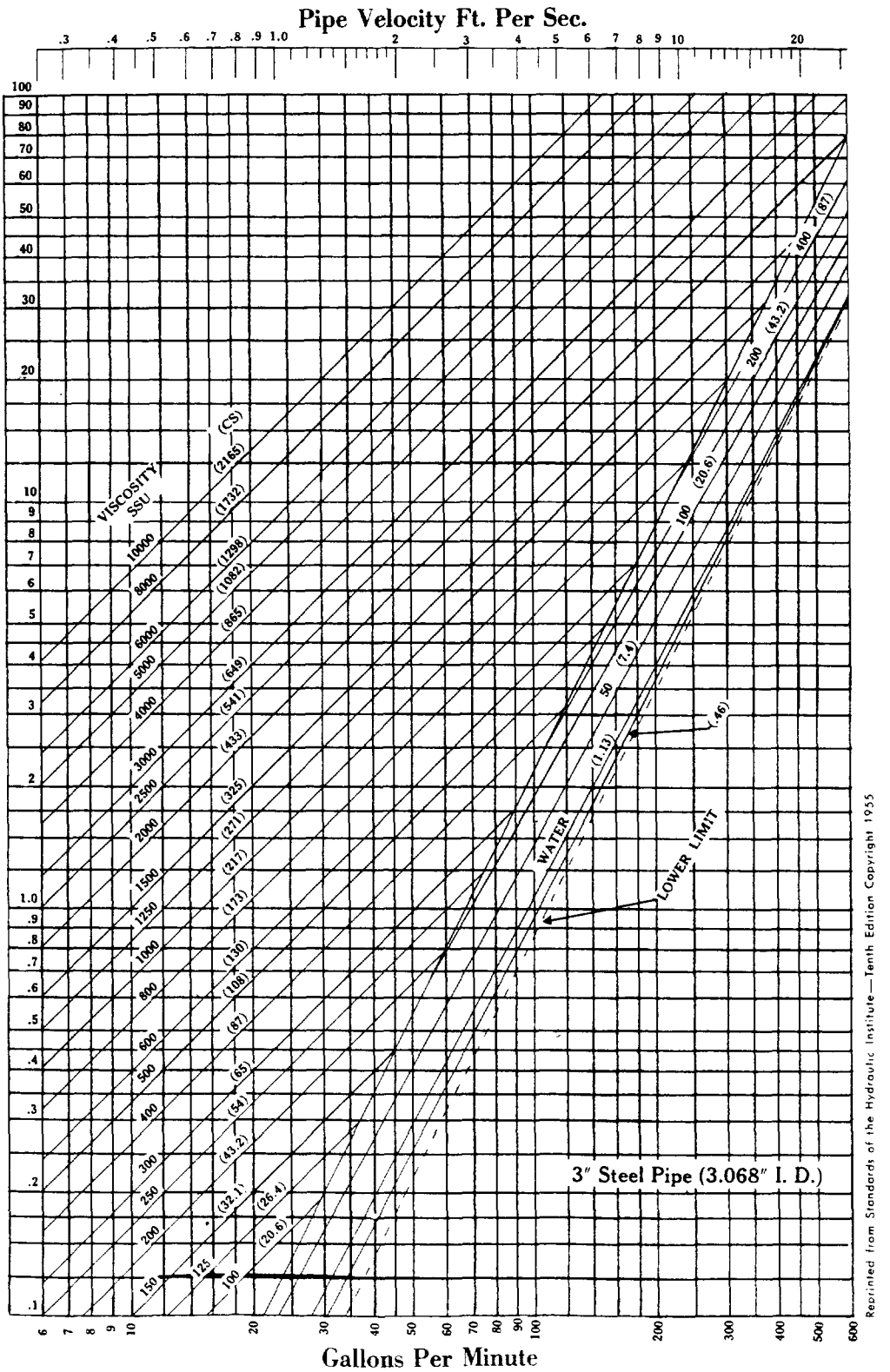


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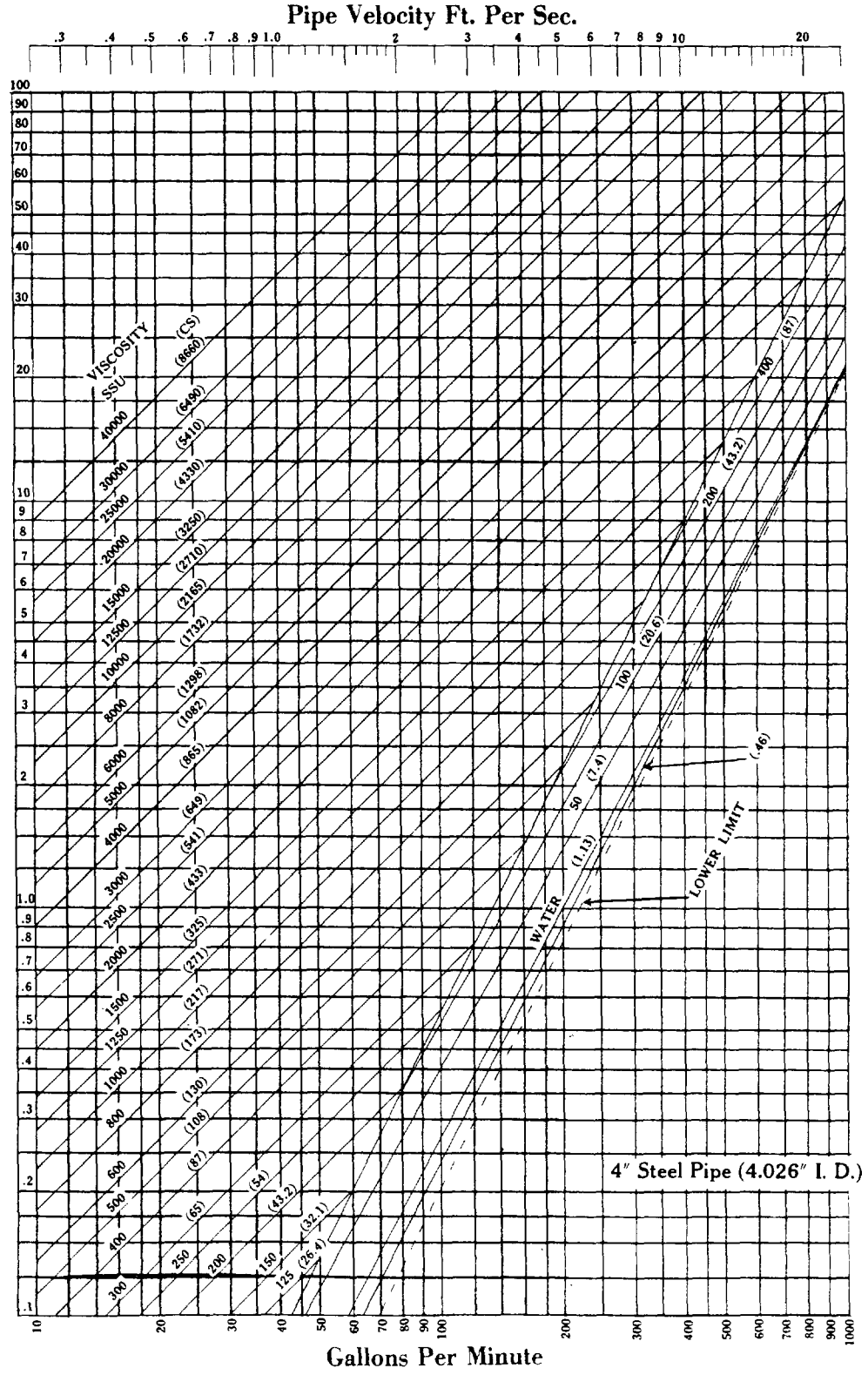


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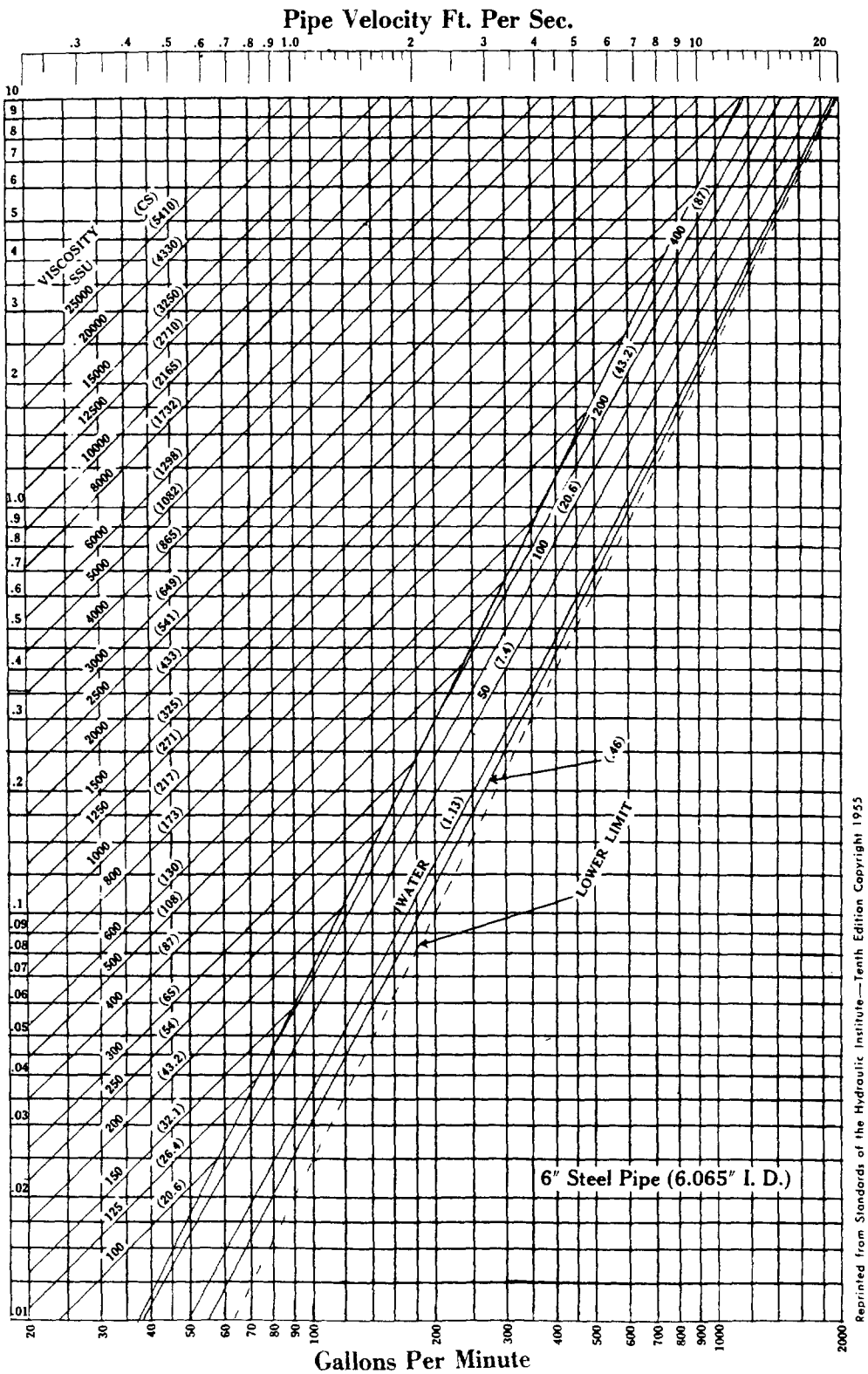


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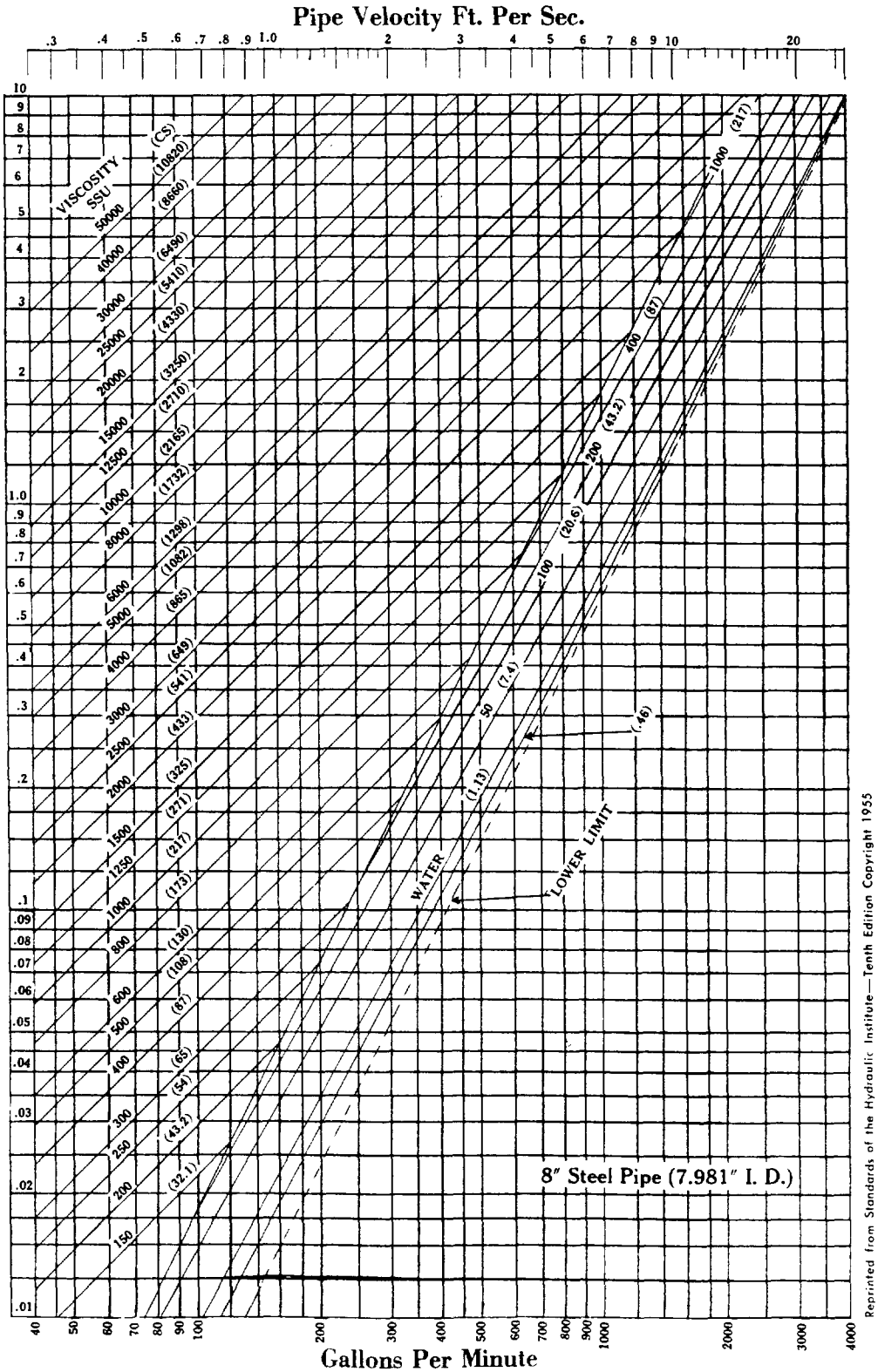


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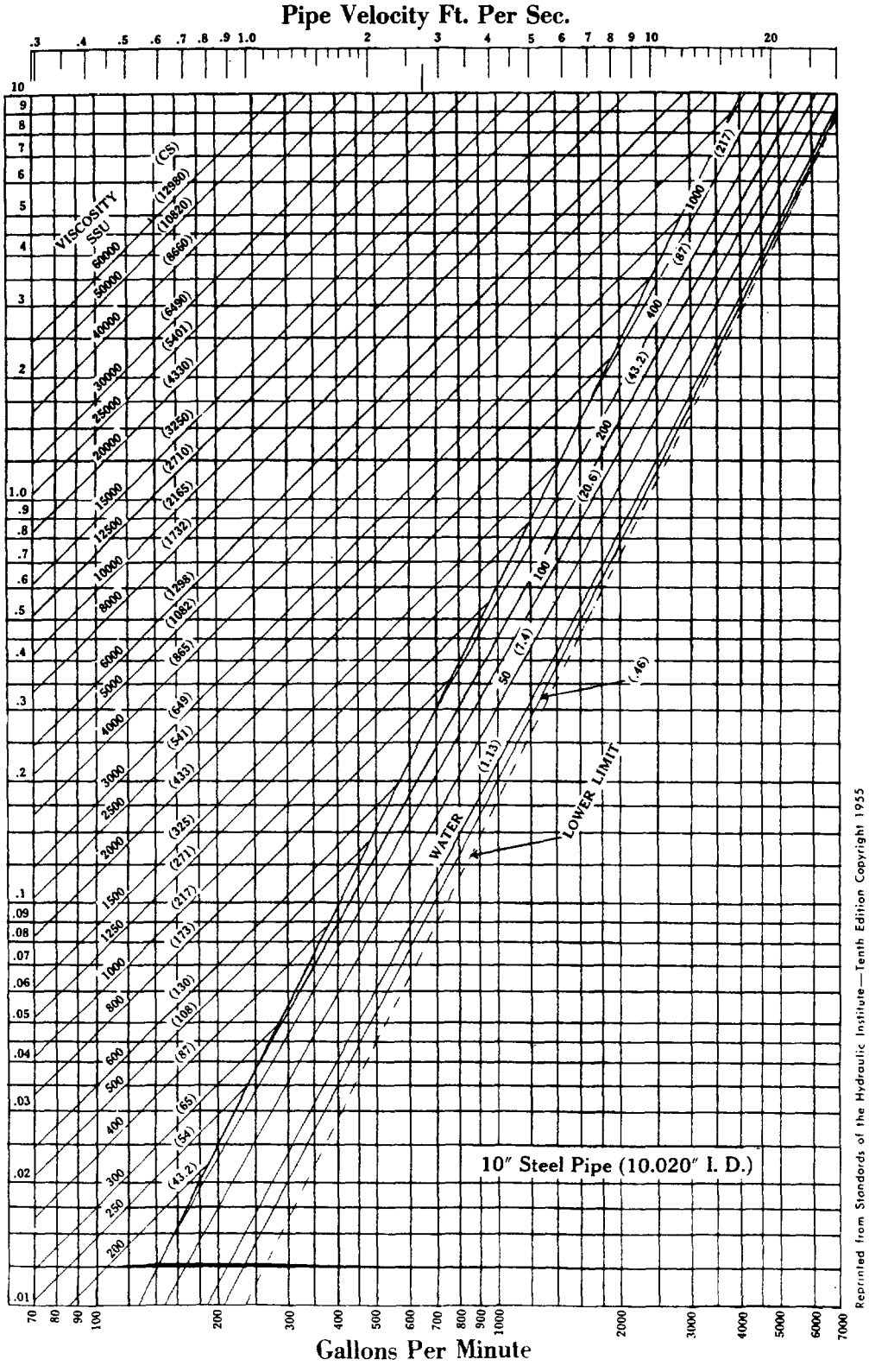


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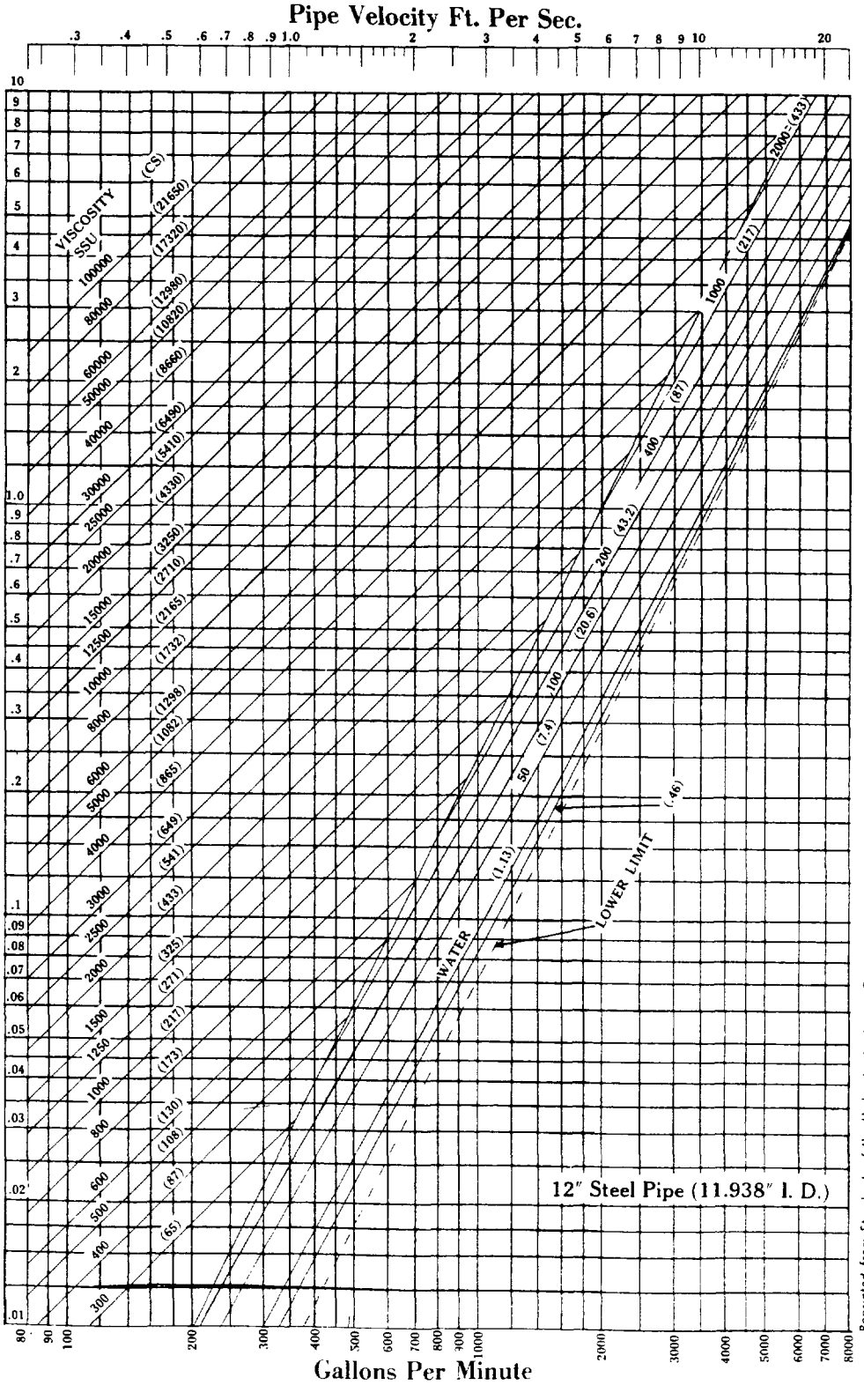


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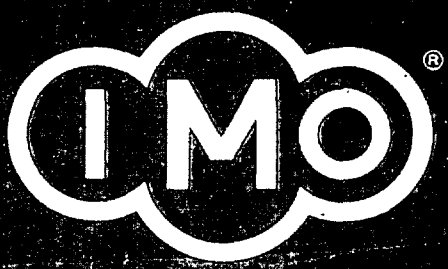
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In the pages of this book are comprehensive data on the selection and application of IMO rotary positive displacement pumps. De Laval offers this information in the hope that it will help you solve problems in handling all types of viscous fluids. We also maintain an experienced engineering staff which you may consult without obligation.



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