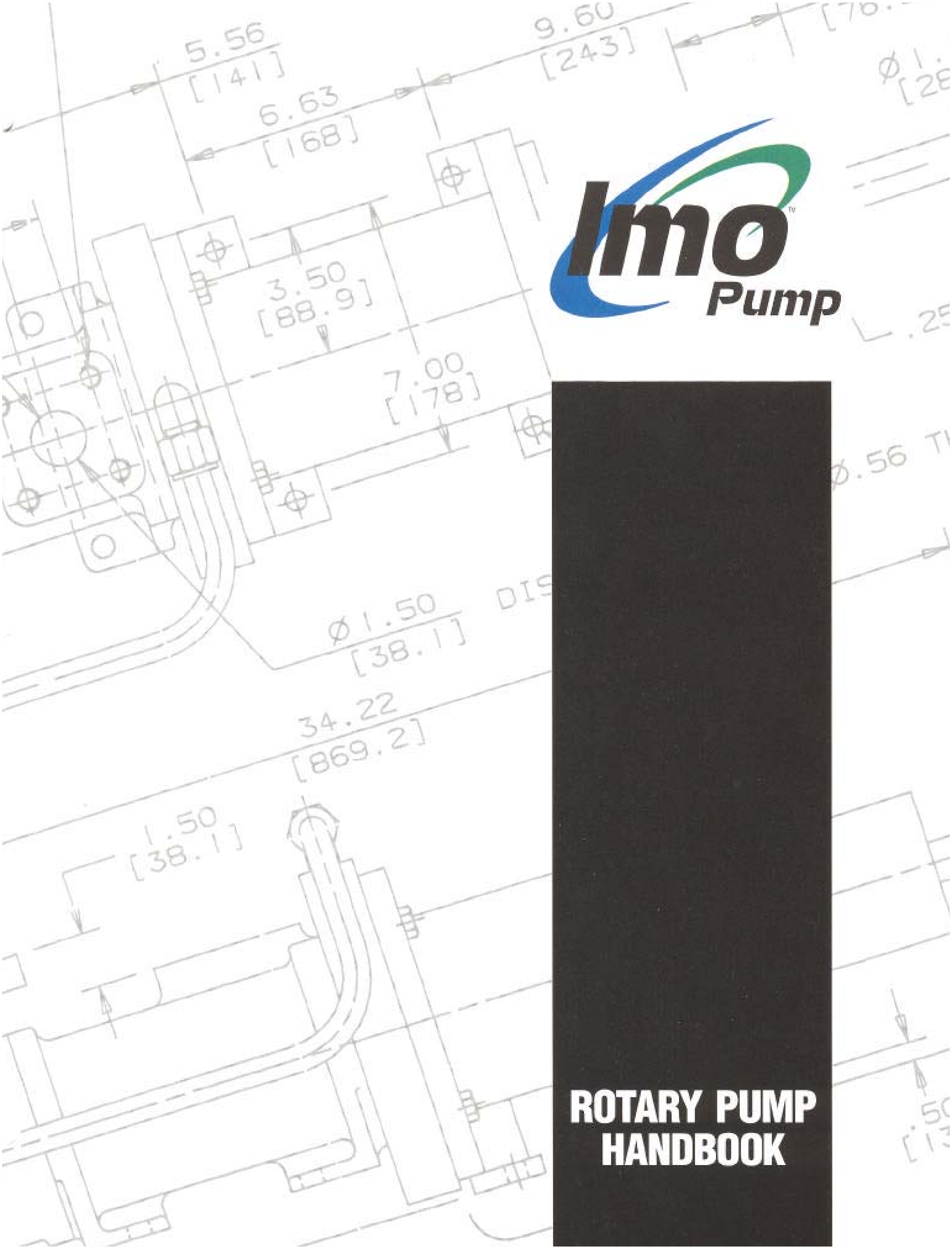

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Pump

ROTARY PUMP HANDBOOK

Contents

Acknowledgements: Some of the material for this handbook has been extracted from the following sources: Hydraulic Institute, Texaco Inc., American National Standards Institute and Engineering Handbook, third edition.

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

The centrifugal pump develops its pressure as a result of centrifugal force 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.

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 requires timing gears whereas the other types do not. All are of the constant displace-

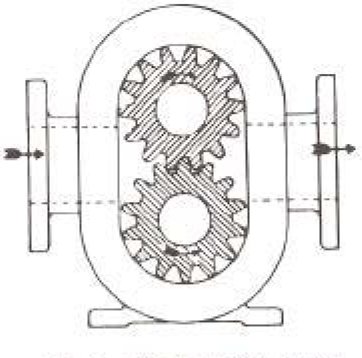


Fig. 1 External Gear Pump

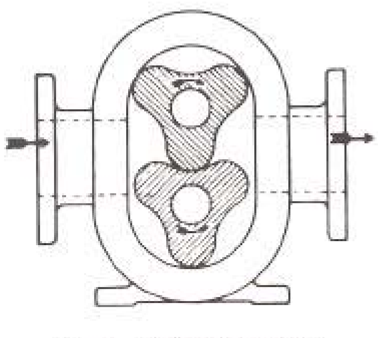


Fig. 2 Three Lobe Pump

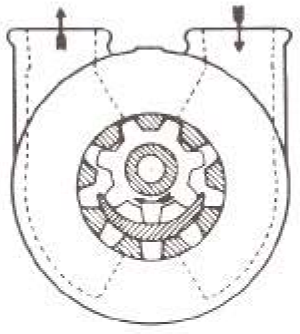


Fig. 3 Internal Gear Pump

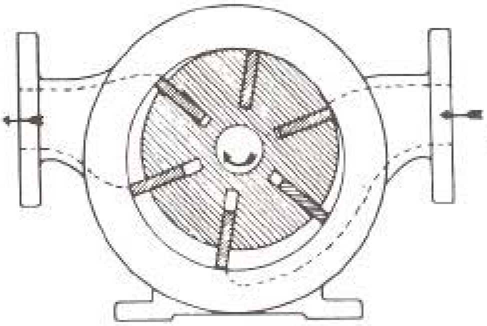


Fig. 4 Sliding Vane Pump

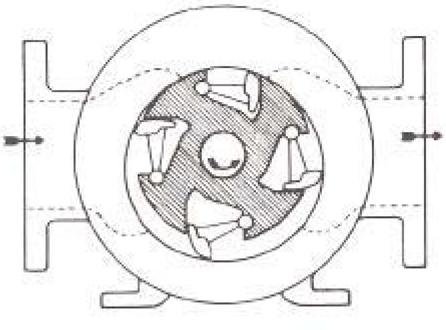


Fig. 5 Swinging Vane Pump

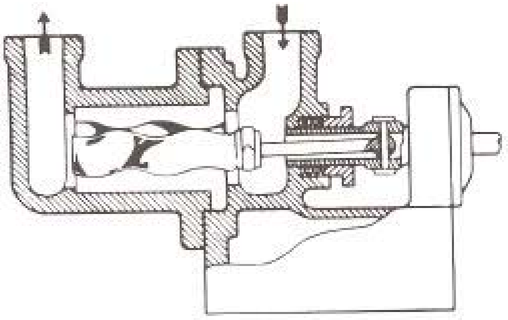


Fig. 6 Single Screw Pump

ment 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 efficiences 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.

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 often quite difficult, however, to obtain accurate information as to these conditions. This is particularly true with reference to inlet conditions and viscosity, since it is a common feeling that in as much as the rotary pump is a positive displacement device these items are unimportant.

In any rotary pump application regardless of the design, suction lift, viscosity, and speed are inseparable. In order to insure quiet, efficient operation, it is necessary to completely fill with the fluid, the compartments in the pumping elements as they open and this becomes more difficult as viscosity or suction lift, or both increase. BASICALLY IF YOU GET THE FLUID INTO A PUMP, OR RATHER THE PUMPING ELEMENTS AS THEY OPEN, YOU CAN GET IT OUT; THE PROBLEM IS GETTING IT IN.

Speed of operation, therefore, is dependent on viscosity and suction lift. If a true picture of these two items can be obtained, the problem of making a proper pump selection becomes infinitely simpler and it is probable that the selection will result in a more efficient unit.

It is not very often that a rotary pump is called upon to handle fluids having a constant viscosity. Normally, due to temperature variations, it is expected that a range of viscosity will be encountered and this can be quite wide, for instance it is not unusual that a pump is required to handle a viscosity range of 150 to 20,000 SSU; the higher viscosity usually being due to cold starting conditions. This is a perfectly satisfactory range in so far as a rotary pump is concerned, but if information can be obtained concerning such things as the amount of time the pump is required to operate at the higher viscosity, whether the motor can be overloaded temporarily, a multi-speed motor used, or the discharge pressure reduced during this period, a better selection can often be made.

Quite often no viscosity is given but only the type of fluid. In such cases assumptions can sometimes be made if sufficient information is available concerning the fluid in question. For instance "Bunker C or No. 6 Fuel Oil" is known to have a wide latitude as to viscosity and usually must be handled over a considerable temperature range. The normal procedure in a case of this type is to assume an operating viscosity range of 20 to 700 SSF. The maximum viscosity, however, might very easily exceed the higher value if extra heavy oil is used or exceptionally low temperatures are encountered. If either should occur the result may be improper filling of the pumping elements, noisy operation, vibration, and overloading of the motor.

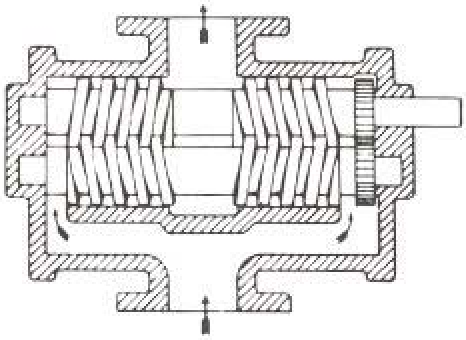


Fig. 7 Two Screw Pump

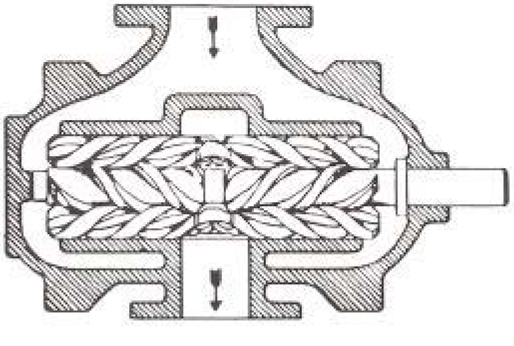


Fig. 8 Three Screw Pump

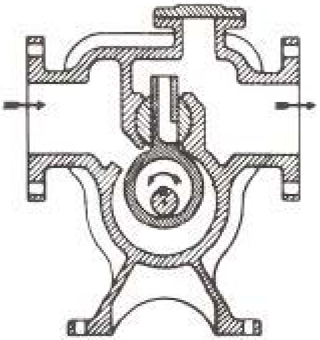


Fig. 9 Cam and Piston Pump

Although it is the maximum viscosity and the expected suction lift that determines the size of the pump and sets the speed, it is the minimum viscosity that determines the capacity. Rotary pumps must always be selected to give the specified capacity when handling the expected minimum viscosity since this is the point at which maximum slip, hence minimum capacity occurs. If this rule is not followed the pump will not meet the requirements of the system unless a considerable margin has been allowed initially in specifying capacity, or there is over-capacity available in the pump. The latter is often the case, since practically all rotary pumps are made in certain stock sizes and it is standard practice to apply the next larger pump when a capacity is specified that falls between sizes.

It should also be noted that the minimum viscosity often sets the model of the pump selected since it is more or less standard policy on the part of most manufacturers to down rate their pumps, in so far as allowable pressure is concerned, when handling liquids having a viscosity of less than 100 SSU. This is done for two reasons. First, to avoid the poorer volumetric efficiency as a result of increased slip under these conditions. Second, due to the fact that a film of the liquid must be maintained between the closely fitted parts which is likely to break down if a combination of low viscosity and high pressure should occur. Although viscosity is not necessarily a definite criterion of film strength, it is generally so used by pump manufacturers.

The viscosity of most liquids, as for example water and mineral oil, are unaffected by any agitation to which they may be subjected as long as the temperature remains constant; they are accordingly known as "true" or "Newtonian," fluids. There is, however, another class of liquids such as cellulose compounds, glues, greases, paints, starches, slurries, candy compounds, etc., which change in viscosity as agitation is varied at constant temperature. Although to be absolutely correct different terms should be used to differentiate between increase or decrease at viscosity with rate of agitation, 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, less expensive, 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.

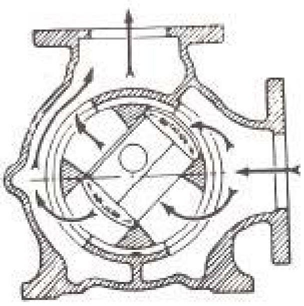


Fig. 10 Shuttle Block Pump

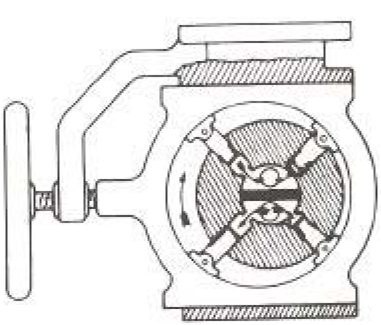


Fig. 11 Radial Plunger Pump

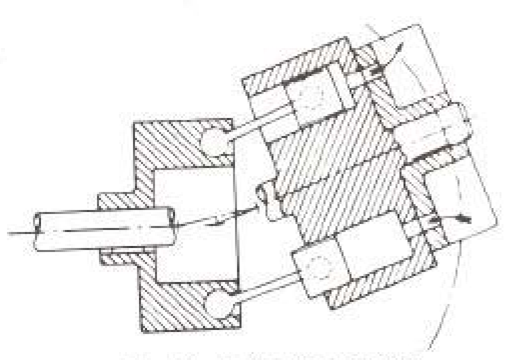


Fig. 12 Axial Plunger Pump

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 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.

There 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 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 the 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 1800 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 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 can cause 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 comparable to that obtained when handling clean fluids. 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.

Positive Displacement, Rotary Pump Characteristics

Theoretical Capacity—Q_T—pump displacement at 0 PSI differential pressure, in GPM.

Differential Pressure—$\triangle P$—Algebraic sum of discharge pressure (gage) and suction pressure (gage)

"Slip"—Q_S—Internal leakage (through working clearances) due to $\triangle P$, in GPM.

Delivered (rated) capacity—Q_C—theoretical capacity minus "slip."

Brake Horsepower—BHP—pump input horsepower.

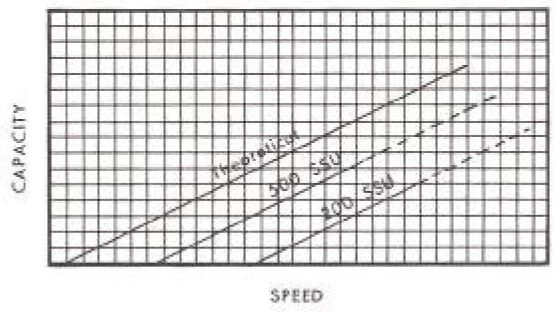
The curves shown below demonstrate in general how capacity and BHP of most positive displacement rotary pumps vary with speed (pressure constant) and with pressure (speed constant). The effect of viscosity is also demonstrated.

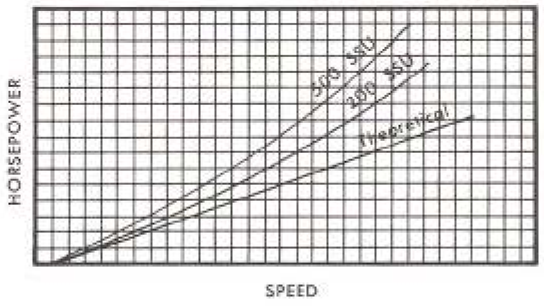
Looking at the curves, one can see that theoretical capacity is directly proportional to speed and is constant regardless of differential pressure. The difference between theoretical capacity and delivered capacity is the "slip" of the pump for a given viscosity. Depending on design, slip can be affected by speed, pressure and viscosity.

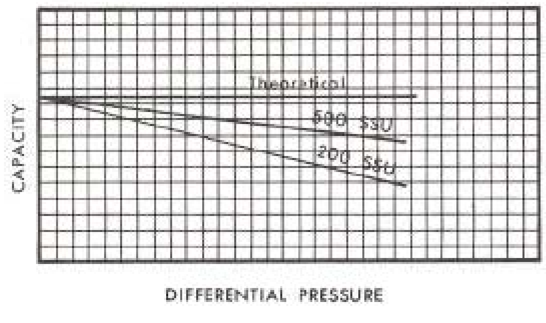
$$\text{Since HP} = \frac{\text{GPM} \times \triangle P}{\text{constant}}$$

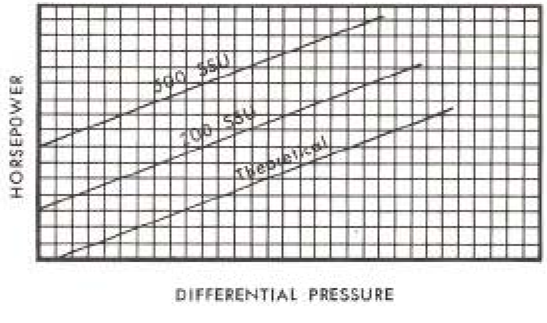
both horsepower curves increase as speed (capacity) or differential pressure increases. It is most important to note the effect of viscosity on horsepower. The difference between actual horsepower and theoretical horsepower for given viscosity is the mechanical loss and slip. Mechanical loss is primarily the loss within the pump due to the resistance of the fluid to shear. Mechanical loss increases with increasing speed and viscosity, and may or may not be dependent on differential pressure.

One significant point demonstrated by these curves is that theoretical capacity is directly proportional to speed while delivered capacity is not, due to the effect of "slip." Thus, half capacity cannot be achieved simply by operating at half speed.









Pump Horsepower and Efficiency

The brake horsepower required to drive a rotary pump is the sum of the theoretical liquid horsepower and the internal power losses. The theoretical liquid horsepower is the actual work done in moving the fluid from its inlet-pressure condition to the outlet at discharge pressure. This work is done on all the fluid of theoretical capacity, not just delivered capacity, as slip does not exist until a pressure differential occurs. Rotary pump power ratings are expressed in terms of horsepower (550 ft-lb/sec) and theoretical liquid horsepower is calculated:

$$\text{tLhp} = \frac{Q_T \triangle P}{1714}$$

Note that theoretical liquid horsepower is independent of viscosity and is concerned only with the physical dimensions of the pumping elements, the rotative speed, and the differential pressure.

The internal power losses are made up of two types: mechanical and viscous. The mechanical losses include all power necessary to overcome the mechanical friction drag of all the moving parts within the pump, including bearings, gears, mechanical seals, etc. The viscous losses include all the power lost from the fluid viscous-drag effects against all the parts within the pump as well as from the shearing action of the fluid itself. It is probable that the mechanical loss is the major component when operating at low viscosities and high speeds while the viscous loss is the larger at high viscosity and slow speed conditions.

$$\text{Volumetric Efficiency, } E_v = \frac{Q_C}{Q_T}$$

$$\text{Overall Pump Efficiency, } E_{op} = \frac{\text{OHP}}{\text{BHP}}$$

$$\text{where OHP (oil horsepower)} = \frac{Q_C \triangle P}{1714}$$

$$\text{Mechanical Efficiency, } E_m = \frac{E_{op}}{E_v}$$

Q_C = delivery capacity, GPM
Q_T = theoretical capacity, GPM
$\triangle P$ = differential pressure, PSI

Suction Pressure

The identification of the pump suction requirement is significant in any pump application. Specifying a higher suction lift than actually exists results in selection of a pump at a lower speed than necessary. Not only does this mean a larger, more expensive pump but also a costlier driver. Should the suction lift requirement be higher than specified, the outcome could be a noisy installation due to pump cavitation.

There is a common misconception that pumps "pull" fluid into the inlet opening unassisted by any outside force. Actually, fluid flows into the pump due to a difference in pressure between pump inlet and the fluid source. A primary step in pump selection is the calculation of system Net Inlet Pressure, sometimes called Net Positive Inlet Pressure. This is the absolute pressure above fluid vapor pressure at pump inlet and is determined as follows:

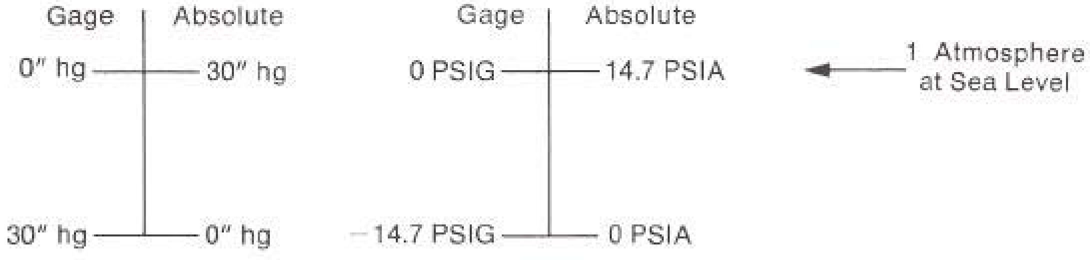
1. Atmospheric pressure (at jobsite altitude—PSIA)
2. Plus static head (minimum level of fluid above pump inlet—PSI)
   or
   Minus static lift (maximum level of fluid below pump inlet—PSI)
3. Minus inlet line friction losses including entrance loss from reservoir to pipe, pressure drops through valves, fittings, strainers, etc. (at maximum viscosity—PSI)
4. Minus fluid vapor pressure (usually at maximum pumping temperature—PSIA)
5. Equals system Net Inlet Pressure in PSIA.

Note that when Net Inlet Pressure is expressed in feet of liquid, it is called Net Positive Suction Pressure.

System Net Inlet Pressure *available* must always equal or exceed pump Net Inlet Pressure *required*.

Suction conditions of rotary pumps are normally rated as suction lift capability in inches of mercury vacuum (with air-free oil of negligible vapor pressure at sea level). This can be expressed as Net Inlet Pressure *required* by subtracting suction lift capability from 30" Hg and converting the remainder to PSIA. Pump Net Inlet Pressure *required* must always be less than or equal to system Net Inlet Pressure available.

Suction condition is the most frequently overlooked parameter in pump selection. Time spent determining it accurately can optimize pump selection and result in a quiet installation.



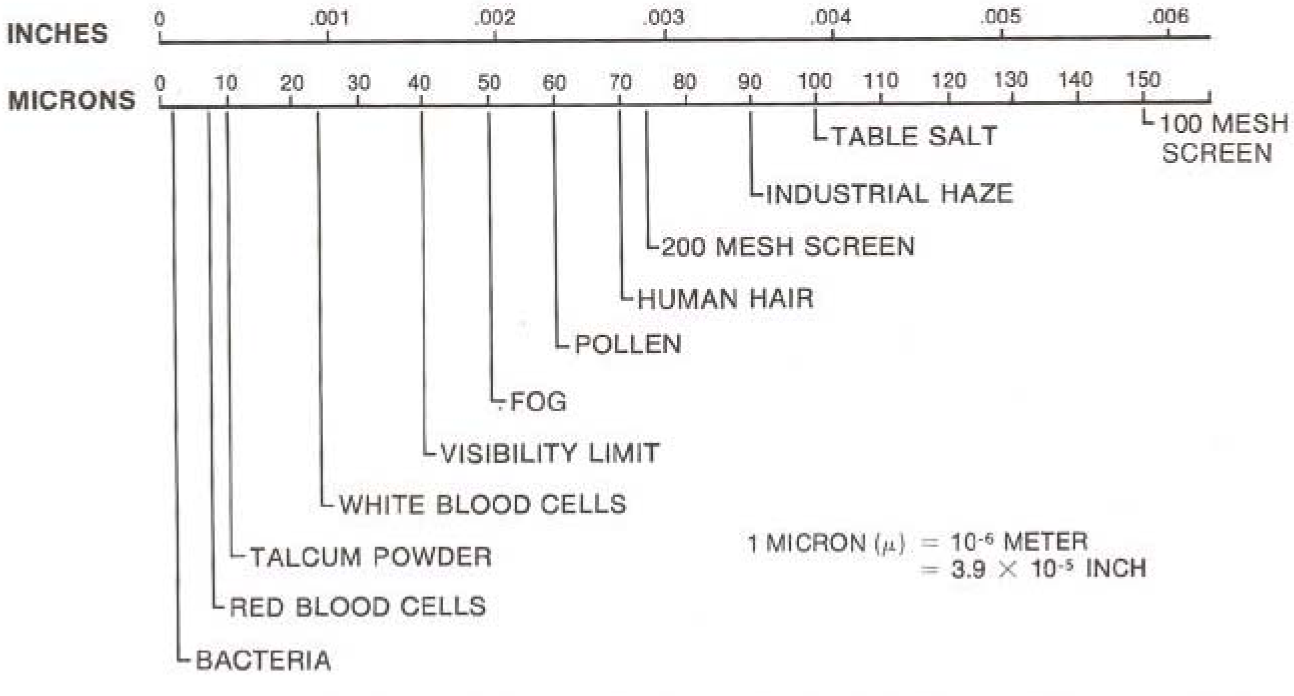
Filtration

As mentioned, positive displacement rotary pumps are not ideally suited to handling abrasives. Relatively close clearances and actual part contact allow dirt, foreign material, etc. to cause scoring, galling, and in extreme cases jamming or seizure. Fine abrasives will cause wear which ultimately reduces pump capacity by increasing slip due to increased clearances.

Firm recommendations for fluid filters are very difficult to make. As a minimum, rotary pumps should have suction strainers to exclude such material as dirt, weld bead, slag, scale, chips, rags, nuts, bolts, etc. Since a suction strainer contributes to suction line loss, the degree of filtration must often be determined by acceptable frequency of cleaning, allowable pressure drop, and cost. When pumping viscous fluids (over 5000 SSU), probably the finest suction filtration practical is 1/16". On light fluids such as distillate fuels, hydraulic oil, and light lube oils, suction strainers of 100 or even 200 mesh are feasible and highly desirable. On recirculating systems, pressure and return line filters of 25 or 10 microns are excellent and quite realistic. In hydraulic servo systems, 2 micron pressure filters are not uncommon.

Strainers and filters require periodic cleaning and, therefore, must be sized and instrumented accordingly. Since clogging of a suction strainer can cavitate a pump, it is particularly important to provide automatic protection by pressure switch or a similar device.

Particle Size Comparison and Conversion Chart



1 MICRON (μ) $= 10^{-6}$ METER

$= 3.9 \times 10^{-5}$ INCH

MESH	INCHES	MICRONS	MESH	INCHES	MICRONS	MESH	INCHES	MICRONS
3250	0.0002	6	130	0.0043	110	24	0.028	718
1600	0.0005	14	120	0.0046	118	20	0.034	872
750	0.0010	25	110	0.0051	131	18	0.039	1000
325	0.0016	40	100	0.0055	149	16	0.045	1154
250	0.0024	62	90	0.0061	156	14	0.051	1308
200	0.0029	74	80	0.0070	179	12	0.060	1538
180	0.0033	85	70	0.0078	200	10	0.075	1923
170	0.0035	90	60	0.0092	238	8	0.097	2488
160	0.0038	97	50	0.0117	300	6	0.132	3385
150	0.0041	100	40	0.015	385	5	0.159	4077
140	0.0042	108	30	0.020	513	4	0.203	5205

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 force.

The two basic viscosity parameters are the DYNAMIC (or ABSOLUTE) VISCOSITY, μ, having the dimension, *force × 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 charts that follow.

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. 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	SS
Air	180.8 x 10^{-6}	0.3369 x 10^{-6}	0.1501	161.6 x 10^{-6}	–
Water	0.010087	21.067 x 10^{-6}	0.010105	10.877 x 10^{-6}	30.

Viscosity Conversion Factors and Formulas

Multiply	by	to obtain
Poises	100	centipoises
pound-seconds/sq. ft.	47,880.1	centipoises
Reyns	6.89473 x 10^{-6}	centipoises
centipoises	2.08855 x 10^{-5}	pound-seconds/sq.
centipoises	1.45038 x 10^{-7}	Reyns
Stokes	100	centistokes
sq. ft./second	92,903.4	centistokes
centistokes	1.07639 x 10^{-5}	sq. ft./second
sq. ft./second	1488.16 x γ 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 x SSU — 195/SSU	for SSU $\leqslant$ 100
μ, in centistokes = 0.220 x SSU — 130/SSU	for SSU $>$ 100
μ, in centistokes = 2.24 x SSF — 184/SSF	for 25 $\leqslant$ SSF $\leqslant$
μ, in centistokes = 2.16 x 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 I (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	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"	Seconds Saybolt Universal ssu
32	1.82	-	30.8	-	1.14	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	32
35	2.71	-	32.1	-	1.16	2420	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	35
40	4.25	-	36.2	5.10	1.31	1440	-	-	-	-	-	-	-	-	-	-	-	-	-	1.3	-	-	-	40
50	7.68	-	44.3	5.83	1.58	838	-	-	-	-	-	-	-	-	-	-	-	-	-	2.3	-	2.6	-	50
60	10.3	-	52.3	6.77	1.88	618	-	-	-	-	-	-	-	-	-	-	-	-	-	3.2	-	3.6	-	60
70	13.1	12.95	60.9	7.60	2.17	483	-	-	-	-	-	-	-	-	-	-	-	-	-	4.1	-	4.6	-	70
80	15.7	13.70	69.2	8.44	2.45	404	-	-	-	-	-	-	-	-	-	-	-	-	-	4.9	-	5.5	-	80
90	18.1	14.44	77.6	9.30	2.73	348	-	-	-	-	-	-	-	-	-	-	-	-	-	5.7	-	6.4	-	90
100	20.5	15.24	85.6	10.12	3.02	307	-	-	-	-	-	-	125	-	38	18	-	-	-	6.5	-	7.3	-	100
150	31.9	19.30	128	14.48	4.48	195	-	-	-	-	-	-	145	-	47	20	-	-	-	10.0	1.0	11.3	-	150
200	43.0	23.5	170	18.90	5.92	144	40	-	-	-	-	-	165	A	54	23	-	-	-	13.5	1.4	15.2	-	200
250	53.8	28.0	212	23.45	7.35	114	46	-	-	-	-	-	198	A	62	26	-	-	-	16.9	1.7	19	-	250
300	64.6	32.5	254	28.0	8.79	95	52.5	15	6.0	3.0	30	20	225	B	73	29	-	-	-	20.4	2.0	23	-	300
400	86.2	41.9	338	37.1	11.70	70.8	66	21	7.2	3.2	42	28	270	C	90	37	-	-	-	27.4	2.7	31	7	400
500	108	51.6	423	46.2	14.60	56.4	79	25	7.8	3.4	50	34	320	D	-	46	-	-	-	34.5	3.5	39	8	500
600	130	61.4	508	55.4	17.50	47.0	92	30	8.5	3.6	58	40	370	F	-	55	-	-	-	41	4.1	46	9	600
700	151	71.1	592	64.6	20.45	40.3	106	35	9.0	3.9	67	45	420	G	-	63	22.5	-	-	48	4.8	54	9.5	700
800	173	81.0	677	73.8	23.35	35.2	120	39	9.8	4.1	74	50	470	-	-	72	24.5	-	-	55	5.5	62	10.8	800
900	194	91.0	762	83.0	26.30	31.3	135	41	10.7	4.3	82	57	515	H	-	80	27	18	-	62	6.2	70	11.9	900
1000	216	100.7	896	92.1	29.20	28.2	149	43	11.5	4.5	92	62	570	I	-	88	29	20	13	69	6.9	77	12.4	1000
1500	324	150	1270	138.2	43.80	18.7	-	65	15.2	6.3	132	90	805	M	-	-	40	28	18	103	10.3	116	16.8	1500
2000	432	200	1690	184.2	58.40	14.1	-	86	19.5	7.5	172	118	1070	Q	-	-	51	34	24	137	13.7	154	22	2000
2500	539	250	2120	230	73.0	11.3	-	108	24	9	218	147	1325	T	-	-	63	41	29	172	17.2	193	27.6	2500
3000	648	300	2540	276	87.60	9.4	-	129	28.5	11	258	172	1690	U	-	-	75	48	33	206	20.6	232	33.7	3000
4000	862	400	3380	368	117.0	7.05	-	172	37	14	337	230	2110	V	-	-	-	63	43	275	27.5	308	45	4000
5000	1079	500	4230	461	146	5.64	-	215	47	18	425	290	2635	W	-	-	-	77	50	344	34.4	385	55.8	5000
6000	1295	600	5080	553	175	4.70	-	258	57	22	520	350	3145	X	-	-	-	-	65	413	41.3	462	65.5	6000
7000	1510	700	5920	645	204.5	4.03	-	300	67	25	600	410	3670	-	-	-	-	-	75	481	48	540	77	7000
8000	1726	800	6770	737	233.5	3.52	-	344	76	29	680	465	4170	Y	-	-	-	-	86	550	55	618	89	8000
9000	1942	900	7620	829	263	3.13	-	387	86	32	780	520	4700	-	-	-	-	-	96	620	62	695	102	9000
10000	2160	1000	8460	921	292	2.82	-	430	96	35	850	575	5220	Z	-	-	-	-	-	690	69	770	113	10000
15000	3240	1500	13700	-	438	2.50	-	650	147	53	1280	860	7720	Z2	-	-	-	-	-	1030	103	1160	172	15000
20000	4320	2000	18400	-	584	1.40	-	860	203	70	1715	1150	10500	Z3	-	-	-	-	-	1370	137	1540	234	20000

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

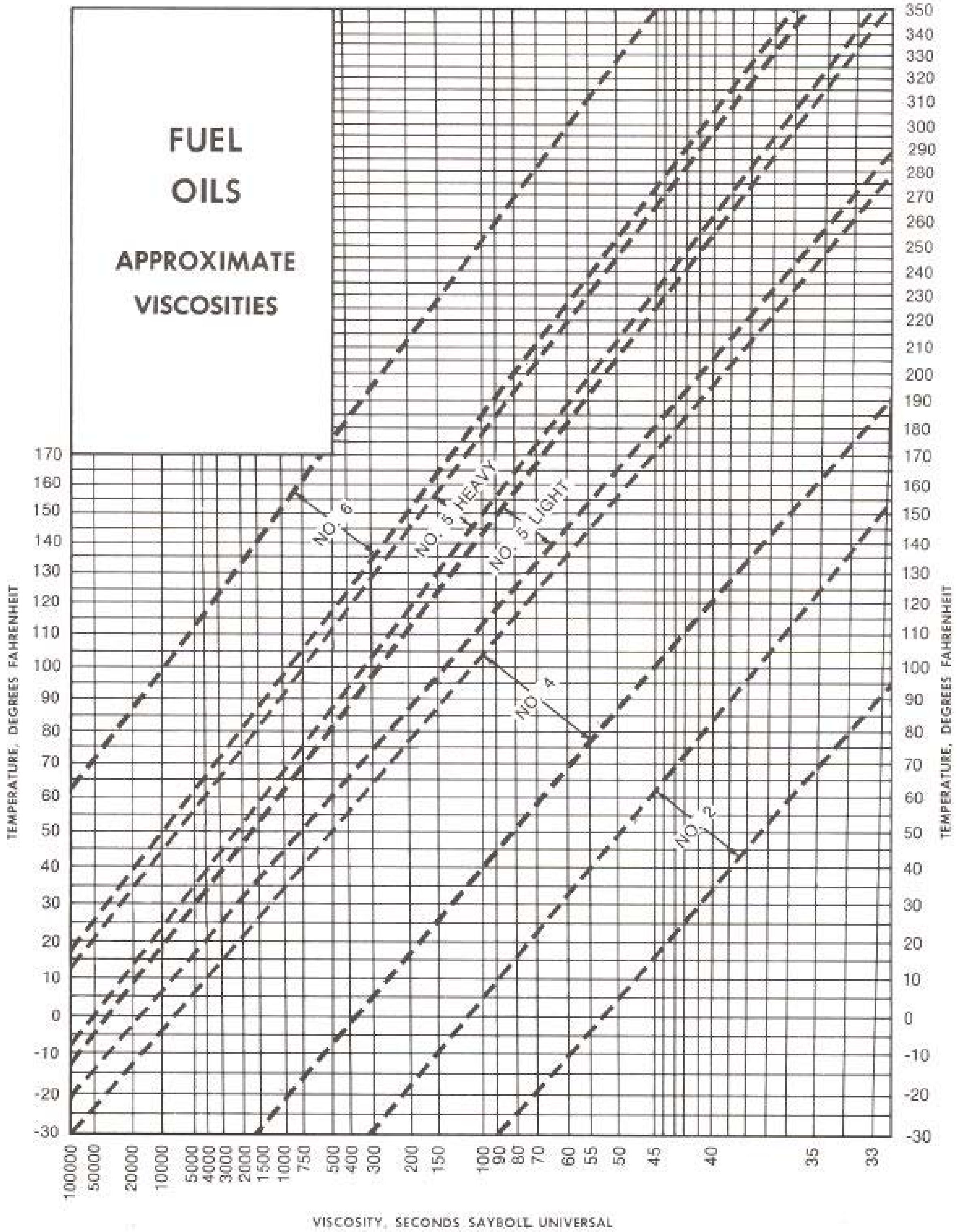
Above 300 SSU, use the following approximate conversion

SSU = Centistokes x 4.635

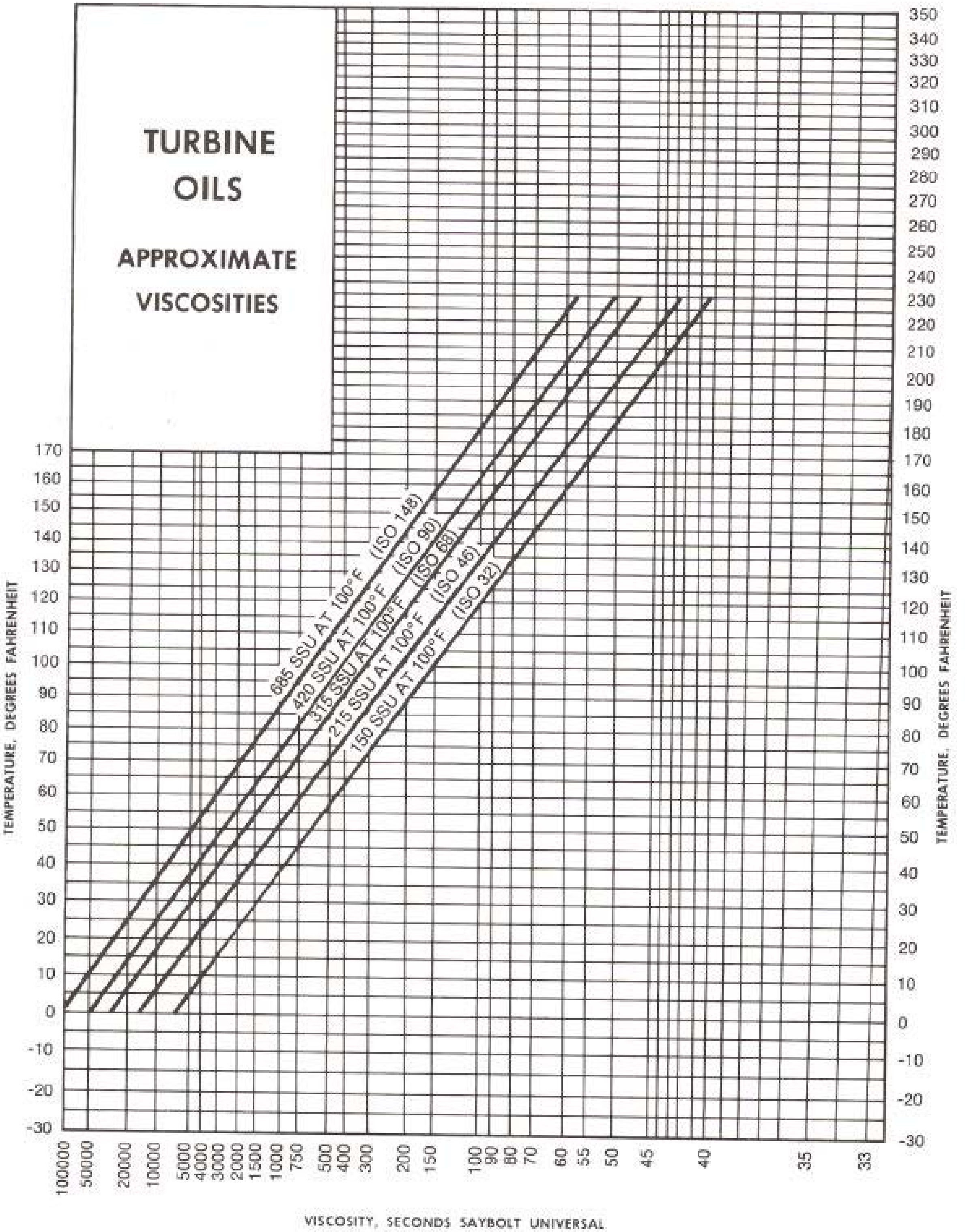
Above the range of this table and within the range of the viscosimeter, multiply their rating by the following factors to convert to SSU.

Viscosimeter	Factor	Viscosimeter	Factor	Viscosimeter	Factor
Saybolt Furol	10.	Parlin cup #15	98.2	Mac Michael	1.92 (approx.)
Redwood Standard	1.095	Parlin cup #20	187.0	Demmler #1	14.6
Redwood Admiralty	10.87	Ford cup # 4	17.4	Demmler #10	146.
Engler Degrees	34.5			Stormer	13. (approx.)

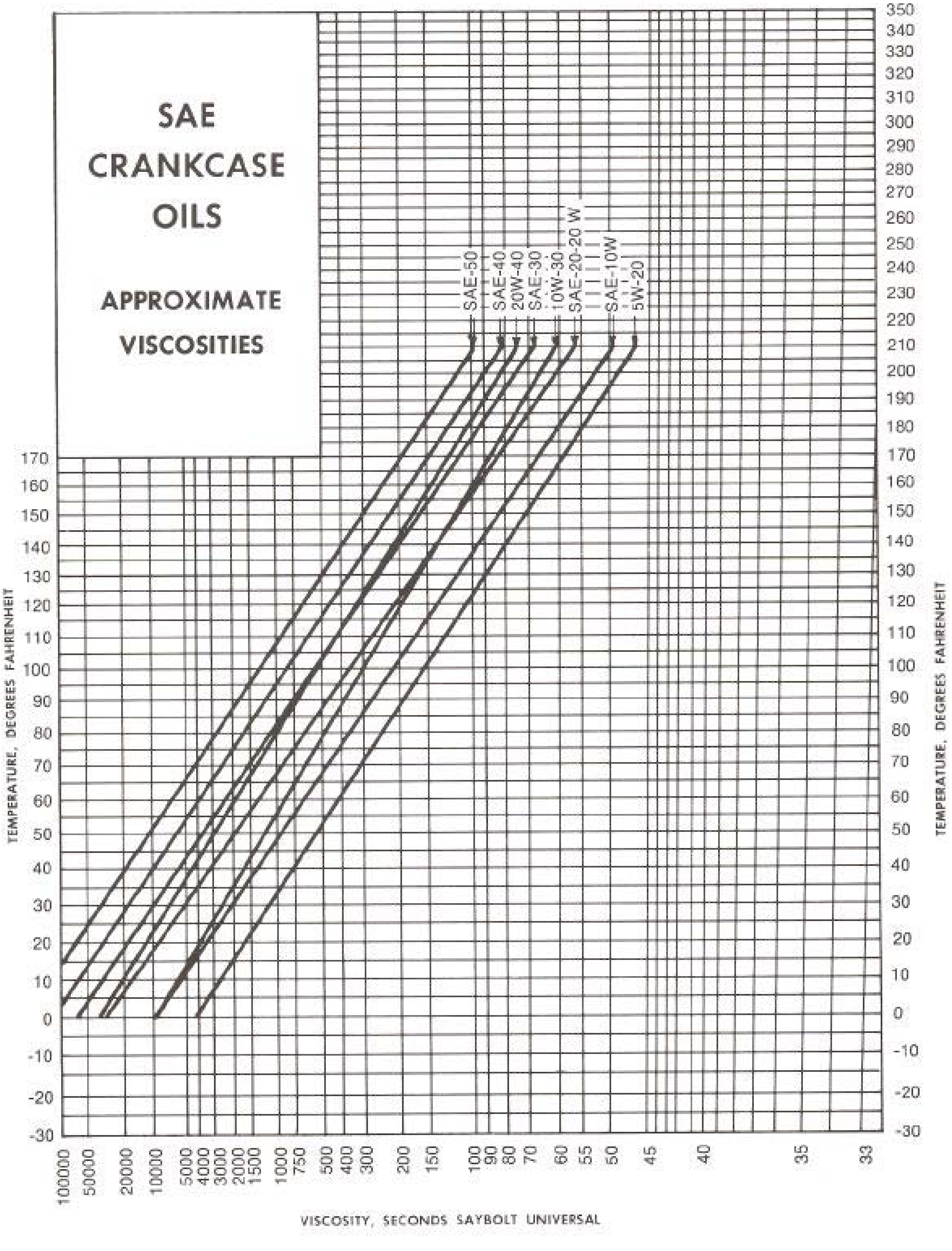
Effect of Temperature on Fuel Oil Viscosity



Effect of Temperature on Turbine Oil Viscosity



Effect of Temperature on SAE Crankcase Oil Viscosity



Viscosity of Common Liquids

Liquid	*Sp Gr at 60°F	VISCOSITY		At°F
		SSU	Centistokes	
ASPHALTS: Unblended or Virgin Asphalts	1.1 to 1.5	2,500 to 12,000 600 to 3,600	539 to 2,810 130 to 776	250 300
Blended Asphalt RS-1, MS-1 or SS-1 emulsified primer or binder	1.0 approx.	155 to 1,000 90 to 350	33 to 216 18.1 to 75.5	77 100
RC-0, MC-0 or SC-0 cutbacks or binders	1.0 approx.	737 to 1,500 280 to 500	159 to 324 60.5 to 108	77 100
RC-1, MC-1 or SC-1 cutbacks or binders	1.0 approx.	2,400 to 5,000 737 to 1,500	518 to 1,079 159 to 324	100 122
RC-2, MC-2 or SC-2 cutbacks or binders	1.0 approx.	2,400 to 5,000 1,000 to 2,000	518 to 1,079 216 to 432	122 140
RC-3, MC-3 or SC-3 cutbacks or binders	1.0 approx.	6,000 to 13,000 2,500 to 5,000	1,295 to 2,810 539 to 1,079	122 140
RC-4, MC-4 or SC-4 cutbacks or binders	1.0 approx.	8,000 to 20,000 1,250 to 2,500	1,726 to 4,320 270 to 539	140 180
RC-5, MC-5 or SC-5 cutbacks or binders	1.0 approx.	28,000 to 85,000 3,000 to 6,000	6,040 to 18,300 648 to 1,295	140 180
CHEMICALS: Glycerine (100%)	1.26 @ 68°F.	2,950 813	648 176	68.6 100
Glycol: Propylene Triethylene Diethylene Ethylene	 1.038 @ 68°F. 1.125 @ 68°F. 1.12 @ 68°F. 1.125 @ 68°F.	 240.6 185.7 149.7 88.4	 52 40 32 17.8	 70 70 70 70
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

*Unless otherwise noted.

Viscosity of Common Liquids

Liquid	*Sp Gr at 60°F.	VISCOSITY		At°F
		SSU	Centistokes	
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
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
AGMA Gear Oils: AGMA No. 1	.880 to .935	180 to 240 45 to 48	38.1 to 51.8 9.9 to 10.6	100 210
AGMA No. 2	.880 to .935	275 to 360 50 to 55	59.3 to 77.5 11.0 to 12.1	100 210
AGMA No. 3	.880 to .935	490 to 650 63 to 71	106 to 140 13.9 to 15.6	100 210
AGMA No. 4	.880 to .935	650 to 1,000 71 to 87	140 to 216 15.6 to 19.2	100 210
AGMA No. 5	.880 to .935	875 to 1,350 83 to 105	189 to 291 18.4 to 22.3	100 210
AGMA No. 6	.880 to .935	1,350 to 1,850 105 to 125	291 to 398 22.3 to 26.5	100 210
AGMA No. 7	.880 to .935	1,850 to 2,500 125 to 150	398 to 539 26.5 to 31.9	100 210
AGMA No. 8	.880 to .935	2,500 to 3,700 150 to 190	539 to 797 31.9 to 40.2	100 210
AGMA No. 8A	.880 to .935	3,700 to 4,700 190 to 215	797 to 1,015 40.2 to 45.6	100 210

*Unless otherwise noted.

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

Viscosity of Common Liquids

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

*Unless otherwise noted.

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

Viscosity of Common Liquids

Liquid	*Sp Gr at 60° F	VISCOSITY		At° F
		SSU	Centistokes	
Insulating Oil: Transformer, switches and circuit breakers		115 max. 65 max.	24.1 max. 11.75 max.	70 100
Kerosene	.78 to .82	35 32.6	2.69 2	68 100
Lubricating Oil: **ISO 32**	.91 Ave.	150 85	32 17	100 130
ISO 46	.91 Ave.	215 115	46 24	100 130
ISO 68	.91 Ave.	315 150	68 32	100 130
ISO 90	.91 Ave.	420 195	90 42	100 130
ISO 148	.91 Ave.	685 280	148 60	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
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	.920	195 112	42 23.4	100 130
Peanut Oil	.924	221 125	47.8 26.4	100 130
Rape Seed Oil	.919	250 145	54.1 31	100 130

*Unless otherwise noted.

Viscosity of Common Liquids

Liquid	*Sp Gr at 60°F	VISCOSITY		At°F
		SSU	Centistokes	
Rosin Oil	.980	1,500 600	324.7 129.9	100 130
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		70 100

*Unless otherwise noted.

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

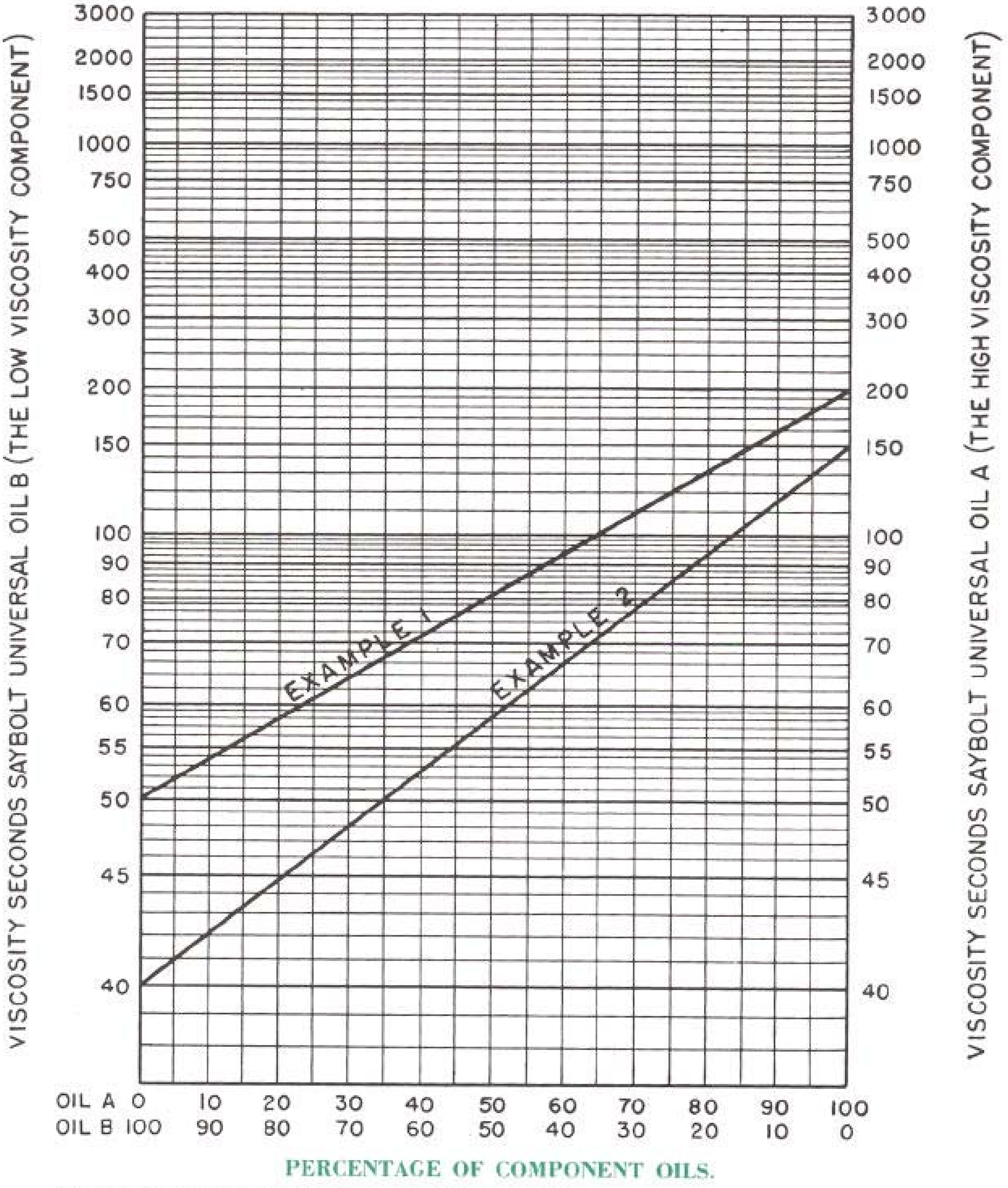
Viscosity of Common Liquids

Liquid	*Sp Gr at 60°F	VISCOSITY		At°F
		SSU	Centistokes	
TARS:				
Tar-Coke Oven	1.12+	3,000 to 8,000 650 to 1,400	600 to 1,760 140.7 to 308	71 100
Tar-Gas House	1.16 to 1.30	15,000 to 300,000 2,000 to 20,000	3,300 to 66,000 440 to 4,400	70 100
Road Tar: Grade RT-2	1.07+	200 to 300 55 to 60	43.2 to 64.9 8.77 to 10.22	122 212
Grade RT-4	1.08+	400 to 700 65 to 75	86.6 to 154 11.63 to 14.28	122 212
Grade RT-6	1.09+	1,000 to 2,000 85 to 125	216.4 to 440 16.83 to 26.2	122 212
Grade RT-8	1.13+	3,000 to 8,000 150 to 225	660 to 1,760 31.8 to 48.3	122 212
Grade RT-10	1.14+	20,000 to 60,000 250 to 400	4,400 to 13,200 53.7 to 86.6	122 212
Grade RT-12	1.15+	114,000 to 456,000 500 to 800	25,000 to 75,000 108.2 to 173.2	122 212
Pine Tar	1.06	2,500 500	559 108.2	100 132
MISCELLANEOUS:				
Corn Starch Solutions: 22 Baumé	1.18	150 130	32.1 27.5	70 100
24 Baumé	1.20	600 440	129.8 95.2	70 100
25 Baumé	1.21	1,400 800	303 173.2	70 100
Ink – Printers	1.00 to 1.38	2,500 to 10,000 1,100 to 3,000	550 to 2,200 238.1 to 660	100 130
Tallow	.918 Avg.	56	9.07	212
Milk	1.02 to 1.05		1.13	68
Varnish – Spar	.9	1,425 650	313 143	68 100
Water – Fresh	1.0		1.13 .55	60 130

*Unless otherwise noted.

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

Viscosity Blending Chart



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At the same temperature, plot viscosity of oil A (high viscosity component) on the right hand scale and oil B (low viscosity component) on the left hand scale. Connect the two points with a straight line. Then read the blend viscosity as a function of percentage of oil A or B.

Newtonian and Non-Newtonian Fluids

Newtonian Materials

Newton deduced that the viscosity of a given liquid should be constant at any particular temperature and pressure and independent of the rate of shear, as illustrated in Fig. 13. In such "Newtonian fluids," shear stress is directly proportional to rate of shear. At temperatures above their cloud points most mineral oils are Newtonian fluids.

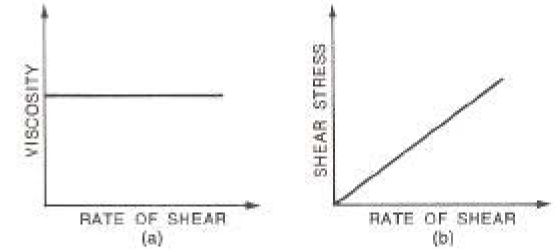

Fig. 13. Characteristics of Newtonian liquids. (a) Viscosity is independent of rate of shear. (b) Shear stress is directly proportional to rate of shear.

Non-Newtonian Materials

The viscosities of some materials, such as greases and polymer-thickened mineral oils, are affected by shearing effects, and these materials are termed *non-Newtonian*. In other words, the viscosity of a non-Newtonian fluid will depend on the rate of shear at which it is measured. Since a non-Newtonian fluid can have an unlimited number of viscosity values (as the shear rate is varied) the term *apparent viscosity* is used to describe its viscous properties. Apparent viscosity is expressed in absolute units and is a measure of the resistance to flow at a given rate of shear. It has meaning only if the rate of shear used in the measurement is also given and is obtained experimentally by measuring and dividing the shear stress by the rate of shear. A "rheogram" or "flow curve" relating shear stress to rate of shear is frequently used to describe completely the viscous properties of a non-Newtonian material.

Non-Newtonian materials may be divided into five types: plastic, pseudo-plastic, dilatant, thixotropic, and rheopectic. Figure 14 presents characteristic rheograms in which shear stress (e.g., pressure in a steady-flow system) is plotted against rate of shear (e.g., flow velocity). The curves at the left in Fig. 15 illustrate how the apparent viscosities of non-Newtonian materials vary with changing rates of shear.

As illustrated in curve 1 of Fig. 14 a *plastic* material, such as a grease, putty, or molding clay, is characterized by a "yield point" or "yield value." This means that a definite minimum stress or force must be applied to the material before any flow takes place. From a rheological standpoint,[1] tomato catsup is a common example of a plastic material. If a bottle is shaken only gently, its contents may not flow out because the "yield point" has not been exceeded. However, if the bottle is struck or shaken more vigorously, the yield point is exceeded, the viscosity is reduced, and the catsup gushes forth.

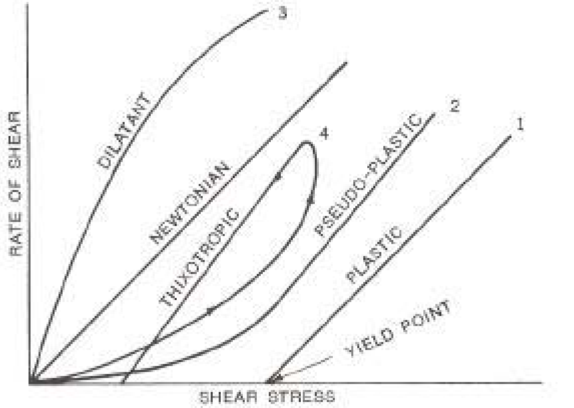

Fig. 14. Flow curves illustrating shear characteristics of various types of materials.

While a pseudo-plastic fluid has no yield point, its apparent viscosity also decreases with increasing shear rates but stabilizes only at very high rates of shear. Many emulsions such as water-base fluids and resinous materials show this type of behavior.

Oppositely, the apparent viscosity of a dilatant fluid increases as the rate of shear increases. Such a fluid often solidifies at high rates of shear. Examples are pigment-vehicle suspensions such as paints and printing inks, and some starches.

The three fluids described above—plastic, pseudo-plastic, and dilatant—are also known as time-independent non-Newtonian fluids, since their rheological or flow properties are independent of time. The rate of shear at any point in the fluid is a simple function of the shear stress at that point.

On the other hand, the flow properties of two other non-Newtonian materials—thixotropic and rheopectic—are dependent on time. The apparent viscosity of these more complex fluids

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[1] Rheology: the science treating of deformation and flow of matter.

depends not only on the magnitude of the shear rate but also on the length of time during which shear has been applied, as illustrated in Fig. 15.

If a thixotropic fluid is subjected to a constant rate of shear for some time, its structure is gradually broken down and its apparent viscosity decreases to some minimum value. When the shear effect is removed and the fluid is at rest, the structure rebuilds gradually and apparent viscosity increases with time to the original value. This is called *reversible thixotrophy*. If, however, upon removing the shear stress, a value less than the original viscosity is obtained with time, the phenomenon is known as *irreversible thixotrophy*. Some oils containing high-molecular-weight polymers and mineral oils at temperatures below their cloud point show this latter effect.

During rotary drilling of deep oil wells, a very special "drilling mud" with thixotropic properties is pumped down the hollow drill stem to force cuttings back to the surface. As long as the mud is agitated by rotation of the drill stem and by pumping, it remains fluid and removes drilling debris. However, whenever drilling is stopped, the drilling mud solidifies to a gel, holds the cuttings in suspension, and thereby prevents them from settling and interfering with subsequent drilling.

Quicksand is also thixotropic, since it becomes more and more fluid when agitated; therefore anyone caught in this water-and-sand mixture improves his chance of survival by remaining as motionless as possible.

If a rheopectic fluid is subjected to a constant rate of shear for a given period of time, its apparent viscosity increases to some maximum value. Upon cessation of shearing and resting for a time, its apparent viscosity decreases again.

Some greases are intentionally manufactured to have partial rheopectic properties, which facilitate pumping from a drum or central grease storage in which the grease is in a relatively fluid condition. Upon shearing in a bearing, however, the grease builds up to a higher apparent viscosity or consistency and stays in place. Such a grease does not have full rheopectic characteristics, however, since after shearing and resting, it still retains a higher consistency.

Since the viscosity of a non-Newtonian lubricant is dependent upon the rate of shear acting on it, the importance of measuring viscosity at various shear rates that will be encountered in the use of such a lubricant can be readily seen. In some machine elements, shear rates up to 3 million reciprocal seconds may be encountered, while in other applications only a few reciprocal seconds or a few tenths are the order of magnitude. In dispensing greases, shear rates as low as 0.1 reciprocal second are sometimes encountered, while leakage from housings during periods of shutdowns involves an even lower range.

As illustrated by Fig. 16, the determination of viscosity of a non-Newtonian liquid at only one shear rate is not usually sufficient. Incorrect conclusions would be drawn and application difficulties would be invited if the viscosities of a Newtonian and a non-Newtonian oil were measured at some specific shear rate R_0, where the two curves happened to cross each other. While both oils have the same apparent viscosity at this one point, the remainders of their viscosity-shear curves are entirely different.

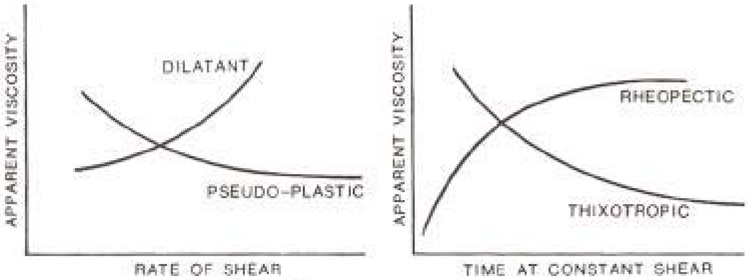


Fig. 15. Different types of non-Newtonian behavior.

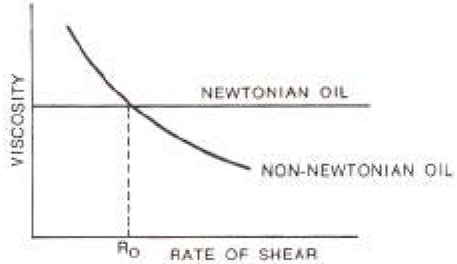
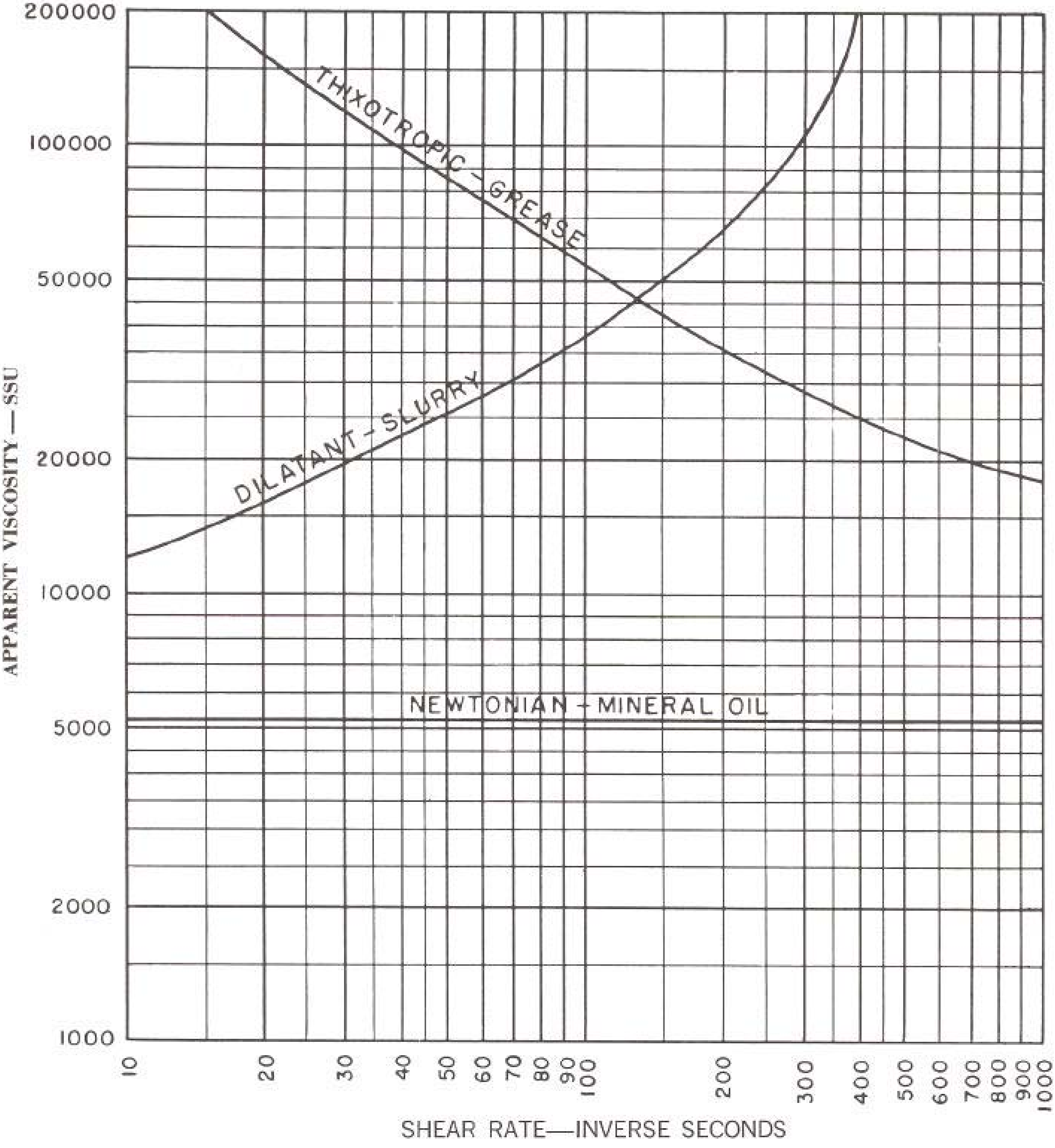


Fig. 16. Viscosity vs. rate of shear for Newtonian and non-Newtonian oils.

Typical Shear Rate Curves



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Conversion Data

ENGLISH SYSTEM

To convert	Multiply by	To obtain
To obtain	Divide by	To convert

UNITS OF LENGTH

yd.	36	in.
yd.	3	ft.
yd.	0.000568	miles
miles	63.360	in.
miles	5,280	ft.
miles	1,760	yd.
miles (naut.)	6,076	ft.

UNITS OF AREA

sq. yd.	9	sq. ft.
sq. yd.	0.0002066	acres
acres	43,560	sq. ft.
acres	4,840	sq. yd.
sq. yd.	1,296	sq. in.

UNITS OF VOLUME

cu. in.	0.00433	gal.
cu. in.	0.000579	cu. ft.
cu. in.	0.0000214	cu. yd.
gal.	231	cu. in.
gal.	0.1337	cu. ft.
gal.	0.00495	cu. yd.
gal.	0.00000307	acre-ft.
gal.	0.0238	bbl (oil)
gal.	0.8327	Imperial gal.
cu. ft.	1,728	cu. in.
cu. ft.	7.48	gal.

UNITS OF WEIGHT

grains	0.00229	oz.
grains	0.0001429	lb.
grains	0.0000000714	tons
oz.	438	grains
oz.	0.0625	lb.
oz.	0.00003125	tons
gal.	8.3322	lb.*
lb.	0.12002	gal.*

VOLUME – FLOW RATES

gps	8.022	cfm
gps	481.3	cu. ft./hr.
gps	60	gpm
gps	3,600	gal./hr.
gpm	0.00223	cfs
gpm	0.1337	cfm
gpm	8.022	cu. ft./hr.
gpm	0.01667	gps
gpm	60	gal./hr.
gpm	499.925	lb./hr.*
gal./hr.	0.0000371	cfs
gal./hr.	0.00223	cfm
gal./hr.	0.1337	cu. ft./hr.
gal./hr.	0.0002778	gps
gal./hr.	0.01667	gpm
bbl/min. (oil)	42	gpm
bbl/day (oil)	0.0292	gpm

METRIC SYSTEM

To convert	Multiply by	To obtain
To obtain	Divide by	To convert

UNITS OF LENGTH

cm	0.3937	in.
meters	39.37	in.
meters	3.281	ft.
meters	1.0936	yd.
km	3,281	ft.
km	1,093.6	yd.
km	0.6214	miles

UNITS OF AREA

sq. mm	0.00155	sq. in.
sq. cm	0.155	sq. in.
sq. meters	10.764	sq. ft.
sq. meters	1.196	sq. yd.
sq. km	0.3861	sq. miles

UNITS OF VOLUME

cu. cm	0.06102	cu. in.
cu. cm	0.03381	fl. oz.
cu. meters	35.31	cu. ft.
cu. meters	1.308	cu. yd.
cu. meters	264.2	US gal.
liters	61.02	cu. in.
liters	0.03531	cu. ft.
liters	0.2642	US gal.

UNITS OF WEIGHT

grams	15.43	grains
grams	0.0353	oz.
kg	35.27	oz.
kg	2.2046	lb.
kg	0.001102	US tons
tonnes	2,204.6	lb.
tonnes	1.1023	US tons

VOLUME – FLOW RATES

liters/sec.	15.85	US gpm
liters/min.	0.2642	US gpm
liters/min.	0.03531	cfm
liters/hr.	0.0044	US gpm
cu. m/min.	35.314	cfm
cu. m/min.	264.17	US gpm
cu. m/hr.	0.5883	cfm
cu. m/hr.	4.4028	US gpm

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*Water at 68° F.

Conversion Data

ENGLISH SYSTEM

To convert	Multiply by	To obtain
To obtain	Divide by	To convert

UNITS OF PRESSURE (Water at 68° F)

in. water	0.0833	ft. water
in. water	0.0736	in. mercury
in. water	82.98	oz./sq. ft.
in. water	0.03602	psi
in. water	5.1869	psf
ft. water	12	in. water
ft. water	0.8832	in. mercury
ft. water	995.8	oz./sq. ft.
ft. water	0.4322	psi
ft. water	62.24	psf
ft. (any liq.)	0.4322 x sp gr	psi
in. mercury	13.57	in. water
in. mercury	1.131	ft. water
in. mercury	1128	oz./sq. ft.
in. mercury	0.4894	psi
in. mercury	70.47	psf
in. mercury	0.03342	atm
atm	29.92	in. mercury

UNITS OF WORK, ENERGY & HEAT

Btu	9,340	in.-lb.
Btu	778.3	ft.-lb.
Btu	0.000293	kwhr.
Btu	0.000393	hp-hr.
in.-lb.	0.000107	Btu
in.-lb.	0.0833	ft.-lb.
in.-lb.	0.00000003138	kwhr.
in.-lb.	0.0000000421	hp-hr.
ft.-lb.	0.001285	Btu
ft.-lb.	12	in.-lb.

UNITS OF POWER

kw	1.341	hp
kw	738	ft.-lb./sec.
kw	44,260	ft.-lb./min.
kw	0.948	Btu/sec.
kw	56.9	Btu/min.
kw	3,413	Btu/hr.
hp	0.7455	kw
hp	550	ft.-lb./sec.
hp	33,000	ft.-lb./min.
hp	0.707	Btu/sec.
hp	42.41	Btu/min.
hp	2,545	Btu/hr.
ft.-lb./sec.	0.001356	kw
ft.-lb./sec.	0.001818	hp
ft.-lb./sec.	60	ft.-lb./min.
ft.-lb./sec.	0.001285	Btu/sec.

METRIC SYSTEM

To convert	Multiply by	To obtain
To obtain	Divide by	To convert

UNITS OF PRESSURE (hg. @ 68°F)

gr/sq. cm	0.01422	psi
gr/cu. cm	0.0361	lb./cu. in.
kg/sq. cm	14.22	psi
kg/cu. m	0.0624	lb./cu. ft.
kg/m	0.6720	lb./ft.
meters water	1.42	psi
meters (any liq.)	1.42 x sp gr	psi
mm mercury	0.001316	atm
mm mercury	1000	microns

UNITS OF WORK, ENERGY & HEAT

gr-cal	0.003969	Btu
kg-cal	3.9693	Btu
kg-cal/kg	1.800	Btu/lb.
gr-cal/sq. cm	3.687	Btu/sq. ft.
kg-cal/cu. m	0.1124	Btu/cu. ft.
joule	0.7376	ft.-lb.
meter-kg	7.2330	ft.-lb.
gr-cal	3.087	ft.-lb.
kg-cal	3,087	ft.-lb.
hp-hr.	1,980,000	ft.-lb.
kwhr.	2,655,000	ft.-lb.
Btu	778.3	ft.-lb.

UNITS OF POWER

watts	0.7376	ft.-lb./sec.
watts	0.001341	hp
kw	1.3410	hp
cheval-vap	0.9863	hp

FORMULAS

TEMPERATURE	$^\circ C = 5/9\ (^\circ F - 32)$ $\quad$ $^\circ F = 9/5\ ^\circ C + 32$
A.P.I. GRAVITY	$\text{Sp. Gr.} = \frac{141.5}{131{:}5 + {}^\circ\text{A.P.I.}}$
POWER	$HP = \frac{T \times RPM}{5252}$ (Torque in foot-pounds)

Equivalent Values of Pressure

Inches of Mercury *			
In. Hg.	PSI	Ft. (S.G.=1.0)	Ft. (S.G.=0.9)
1	.489	1.13	1.26
2	.979	2.26	2.51
3	1.47	3.39	3.77
4	1.96	4.52	5.03
5	2.45	5.66	6.28
6	2.94	6.79	7.54
7	3.42	7.92	8.80
8	3.92	9.05	10.0
9	4.40	10.2	11.3
10	4.89	11.3	12.6
11	5.38	12.4	13.8
12	5.87	13.6	15.1
13	6.36	14.7	16.3
14	6.85	15.8	17.6
15	7.34	17.0	18.8
16	7.83	18.1	20.1
17	8.32	19.2	21.4
18	8.81	20.4	22.6
19	9.30	21.5	23.9
20	9.79	22.6	25.1
21	10.3	23.8	26.4
22	10.8	24.9	27.6
23	11.2	26.0	28.9
24	11.7	27.1	30.2
25	12.2	28.3	31.4
26	12.7	29.4	32.7
27	13.2	30.5	33.9
28	13.7	31.7	35.2
29	14.2	32.8	36.4
30	14.7	33.9	37.7

Feet of 1.0 S.G. Liquid			
Ft. (S.G.=1.0)	In. Hg.	PSI	Ft. (S.G.=0.9)
1	0.883	0.432	1.11
2	1.77	0.864	2.22
3	2.65	1.30	3.33
4	3.53	1.73	4.44
5	4.42	2.16	5.56
6	5.30	2.59	6.67
7	6.18	3.02	7.78
8	7.06	3.46	8.89
9	7.95	3.89	10.0
10	8.83	4.32	11.1
15	13.2	6.48	16.7
20	17.7	8.64	22.2
25	22.1	10.8	27.8
30	26.5	13.0	33.3
35	30.9	15.1	37.8
40	35.3	17.3	44.4
45	39.7	19.4	50.0
50	44.2	21.6	55.6
60	53.0	25.9	66.7
70	61.8	30.2	77.8
80	70.6	34.6	88.9
90	79.5	38.9	100
100	88.3	43.2	111
150	132	64.8	167
200	177	86.4	222
250	221	108	278
500	442	216	556
750	662	324	833
1000	883	432	1111

Pounds per square inch			
PSI	In. Hg.	Ft. (S.G.=1.0)	Ft. (S.G.=0.9)
1	2.04	2.31	2.57
2	4.09	4.63	5.14
3	6.13	6.94	7.71
4	8.17	9.25	10.3
5	10.2	11.6	12.8
6	12.2	13.9	15.4
7	14.3	16.2	18.0
8	16.3	18.5	20.6
9	18.4	20.8	23.1
10	20.4	23.1	25.7
15	30.6	34.7	38.6
20	40.9	46.3	51.4
25	51.1	57.8	64.3
30	61.3	69.4	77.1
35	71.5	81.0	90.0
40	81.7	92.5	103
45	91.9	104	116
50	102	116	128
60	122	139	154
70	143	162	180
80	163	185	206
90	184	208	231
100	204	231	257
150	306	347	386
200	409	463	514
250	511	578	643
500	1021	1157	1285
750	1532	1735	1928
1000	2043	2314	2571

Feet of 0.9 S.G. Liquid			
Ft. (S.G.=0.9)	In. Hg.	PSI	Ft. (S.G.=1.0)
1	0.794	0.389	0.900
2	1.59	0.778	1.80
3	2.38	1.17	2.70
4	3.18	1.56	3.60
5	3.97	1.94	4.50
6	4.77	2.33	5.40
7	5.56	2.72	6.30
8	6.36	3.11	7.20
9	7.15	3.50	8.10
10	7.95	3.89	9.00
15	11.9	5.83	13.5
20	15.9	7.78	18.0
25	19.9	9.72	22.5
30	23.8	11.7	27.0
35	27.8	13.6	31.5
40	31.8	15.6	36.0
45	35.8	17.5	40.5
50	39.7	19.4	45.0
60	47.7	23.3	54.0
70	55.6	27.2	63.0
80	63.6	31.1	72.0
90	71.5	35.0	81.0
100	79.5	38.9	90.0
150	119	58.3	135
200	159	77.8	180
250	199	97.2	225
500	397	194	450
750	596	292	675
1000	795	389	900

Properties of the Lower Atmosphere

Altitude, ft.	Pressure, psia	Pressure, in. Hg at 32° F.	Specific wt, lb./cu. ft.	Temp., ° F.
0	14.696	29.921	0.07648	59.0
100	14.64	29.81	0.0763	58.6
200	14.59	29.71	0.0760	58.3
300	14.54	29.60	0.0758	57.9
400	14.48	29.49	0.0756	57.6
500	14.43	29.38	0.0754	57.2
600	14.38	29.28	0.0751	56.9
700	14.33	29.17	0.0749	56.5
800	14.28	29.07	0.0747	56.1
900	14.22	28.96	0.0745	55.8
1,000	14.17	28.86	0.0743	55.4
1,100	14.12	28.75	0.0740	55.1
1,200	14.07	28.65	0.0738	54.7
1,300	14.02	28.54	0.0736	54.4
1,400	13.97	28.44	0.0734	54.0
1,500	13.92	28.33	0.0732	53.7
1,600	13.87	28.23	0.0730	53.3
1,700	13.82	28.13	0.0727	52.9
1,800	13.76	28.02	0.0725	52.6
1,900	13.71	27.92	0.0723	52.2
2,000	13.66	27.82	0.0721	51.9
2,200	13.56	27.62	0.0717	51.2
2,400	13.47	27.42	0.0712	50.4
2,600	13.37	27.21	0.0708	49.7
2,800	13.27	27.02	0.0704	49.0
3,000	13.17	26.82	0.0700	48.3
3,500	12.93	26.33	0.0689	46.5
4,000	12.69	25.84	0.0679	44.7
4,500	12.46	25.37	0.0669	43.0
5,000	12.23	24.90	0.0659	41.2
6,000	11.78	23.98	0.0639	37.6
7,000	11.34	23.09	0.0620	34.0
8,000	10.92	22.22	0.0601	30.5
9,000	10.50	21.39	0.0583	26.9
10,000	10.11	20.58	0.0565	23.3
12,000	9.346	19.03	0.0530	16.2
14,000	8.633	17.58	0.0497	9.1
15,000	8.293	16.89	0.0481	5.5
16,000	7.965	16.21	0.0466	1.9
18,000	7.339	14.94	0.0436	− 5.2
20,000	6.753	13.75	0.0407	−12.3
25,000	5.453	11.10	0.0343	−30.2
30,000	4.364	8.885	0.0286	−48.0
35,000	3.458	7.041	0.0237	−65.8
40,000	2.720	5.538	0.0188	−69.7
45,000	2.139	4.355	0.0148	−69.7
50,000	1.682	3.425	0.0116	−69.7
55,000	1.323	2.693	0.00915	−69.7
60,000	1.040	2.118	0.00720	−69.7
65,000	0.8180	1.665	0.00566	−69.7

Data from NASA Standard Atmosphere (1962).

ANSI Standard Steel Pipe Flanges

Nominal pipe size	Flange OD	Flange thickness	Bolt circle dia.	Bore weld neck socket weld†	No. of bolts	Bolt dia.
150-lb. Standard*						
1/2	3-1/2	7/16	2-3/8	0.62	4	1/2
3/4	3-7/8	1/2	2-3/4	0.82	4	1/2
1	4-1/4	9/16	3-1/8	1.05	4	1/2
1-1/4	4-5/8	5/8	3-1/2	1.38	4	1/2
1-1/2	5	11/16	3-7/8	1.61	4	1/2
2	6	3/4	4-3/4	2.07	4	5/8
2-1/2	7	7/8	5-1/2	2.47	4	5/8
3	7-1/2	15/16	6	3.07	4	5/8
3-1/2	8-1/2	15/16	7	3.55	8	5/8
4	9	15/16	7-1/2	4.03	8	5/8
5	10	15/16	8-1/2	5.05	8	3/4
6	11	1	9-1/2	6.07	8	3/4
8	13-1/2	1-1/8	11-3/4	7.98	8	3/4
10	16	1-3/16	14-1/4	10.02	12	7/8
12	19	1-1/4	17	12.00	12	7/8
300-lb. Standard*						
1/2	3-3/4	9/16	2-5/8	0.62	4	1/2
3/4	4-5/8	5/8	3-1/4	0.82	4	5/8
1	4-7/8	11/16	3-1/2	1.05	4	5/8
1-1/4	5-1/4	3/4	3-7/8	1.38	4	5/8
1-1/2	6-1/8	13/16	4-1/2	1.61	4	3/4
2	6-1/2	7/8	5	2.07	8	5/8
2-1/2	7-1/2	1	5-7/8	2.47	8	3/4
3	8-1/4	1-1/8	6-5/8	3.07	8	3/4
3-1/2	9	1-3/16	7-1/4	3.55	8	3/4
4	10	1-1/4	7-7/8	4.03	8	3/4
5	11	1-3/8	9-1/4	5.05	8	3/4
6	12-1/2	1-7/16	10-5/8	6.07	12	3/4
8	15	1-5/8	13	7.98	12	7/8
10	17-1/2	1-7/8	15-1/4	10.02	16	1
12	20-1/2	2	17-3/4	12.00	16	1-1/8
600-lb. Standard §						
1/2	3-3/4	9/16	2-5/8	†	4	1/2
3/4	4-5/8	5/8	3-1/4	†	4	5/8
1	4-7/8	11/16	3-1/2	†	4	5/8
1-1/4	5-1/4	13/16	3-7/8	†	4	5/8
1-1/2	6-1/8	7/8	4-1/2	†	4	3/4
2	6-1/2	1	5	†	8	5/8
2-1/2	7-1/2	1-1/8	5-7/8	†	8	3/4
3	8-1/4	1-1/4	6-5/8	†	8	3/4
3-1/2	9	1-3/8	7-1/4	†	8	7/8
4	10-3/4	1-1/2	8-1/2	†	8	7/8
5	13	1-3/4	10-1/2	†	8	1
6	14	1-7/8	11-1/2	†	12	1
8	16-1/2	2-3/16	13-3/4	†	12	1-1/8
10	20	2-1/2	17	†	16	1-1/4
12	22	2-5/8	19-1/4	†	20	1-1/4

Nominal pipe size	Flange OD	Flange thickness	Bolt circle dia.	Bore weld neck socket weld	No. of bolts	Bolt dia.
900-lb. Standard §						
1/2	4-3/4	7/8	3-1/4	†	4	3/4
3/4	5-1/8	1	3-1/2	†	4	3/4
1	5-7/8	1-1/8	4	†	4	7/8
1-1/4	6-1/4	1-1/8	4-3/8	†	4	7/8
1-1/2	7	1-1/4	4-7/8	†	4	1
2	8-1/2	1-1/2	6-1/2	†	8	7/8
2-1/2	9-5/8	1-5/8	7-1/2	†	8	1
3	10-1/2	1-1/2	7-1/2	†	8	7/8
4	11-1/2	1-3/4	9-1/4	†	8	1-1/8
5	13-3/4	2	11	†	8	1-1/4
6	15	2-3/16	12-1/2	†	12	1-1/8
8	18-1/2	2-1/2	15-1/2	†	12	1-3/8
10	21-1/2	2-3/4	18-1/2	†	16	1-3/8
12	24	3-1/8	21	†	20	1-3/8
1,500 lb. Standard §						
1/2	4-3/4	7/8	3-1/4	†	4	3/4
3/4	5-1/8	1	3-1/2	†	4	3/4
1	5-7/8	1-1/8	4	†	4	7/8
1-1/4	6-1/4	1-1/8	4-3/8	†	4	7/8
1-1/2	7	1-1/4	4-7/8	†	4	1
2	8-1/2	1-1/2	6-1/2	†	8	7/8
2-1/2	9-5/8	1-5/8	7-1/2	†	8	1
3	10-1/2	1-7/8	8	†	8	1-1/8
4	12-1/4	2-1/8	9-1/2	†	8	1-1/4
5	14-3/4	2-7/8	11-1/2	†	8	1-1/2
6	15-1/2	3-1/4	12-1/2	†	12	1-3/8
8	19	3-5/8	15-1/2	†	12	1-5/8
10	23	4-1/4	19	†	12	1-7/8
12	26-1/2	4-7/8	22-1/2	†	16	2
2,500-lb. Standard §						
1/2	5-1/4	1-3/16	3-1/2	†	4	3/4
3/4	5-1/2	1-1/4	3-3/4	†	4	3/4
1	6-1/4	1-3/8	4-1/4	†	4	7/8
1-1/4	7-1/4	1-1/2	5-1/8	†	4	1
1-1/2	8	1-3/4	5-3/4	†	4	1-1/8
2	9-1/4	2	6-3/4	†	8	1
2-1/2	10-1/2	2-1/4	7-3/4	†	8	1-1/8
3	12	2-5/8	9	†	8	1-1/4
4	14	3	10-3/4	†	8	1-1/2
5	16-1/2	3-5/8	12-3/4	†	8	1-3/4
6	19	4-1/4	14-1/2	†	8	2
8	21-3/4	5	17-1/4	†	12	2
10	26-1/2	6-1/2	21-1/4	†	12	2-1/2
12	30	7-1/4	24-3/8	†	12	2-3/4

From ANSI Standard, Steel Pipe Flanges and Flanged Fittings, B16.5—1968.

* Flange thickness includes 1/16-in. raised face.

† To be specified by purchaser.

§ Flange thickness does not include 1/4-in. raised face.

ANSI Standard Cast-Iron Pipe Flanges

Nominal pipe size	Flange OD	Flange thickness	Bolt circle dia	No. 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
3-1/2	8-1/2	13/16	7	8	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
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

From ANSI Standard, Cast-Iron Pipe Flanges and Flange Fittings. B16.1–1967.

*Flange thickness includes 1/16-in. raised face.

ANSI Standard Pipe Threads

Nominal pipe size	Pipe OD	Threads per inch	Length of effective threads	Length of hand-tight engagement	Total thread length
1/16	0.3125	27	0.26	0.16	0.39
1/8	0.405	27	0.26	0.16	0.39
1/4	0.540	18	0.40	0.23	0.60
3/8	0.675	18	0.41	0.24	0.60
1/2	0.840	14	0.53	0.32	0.78
3/4	1.050	14	0.55	0.34	0.79
1	1.315	11-1/2	0.68	0.40	0.99
1-1/4	1.660	11-1/2	0.71	0.42	1.01
1-1/2	1.900	11-1/2	0.72	0.42	1.03
2	2.375	11-1/2	0.76	0.44	1.06
2-1/2	2.875	8	1.14	0.68	1.57
3	3.500	8	1.20	0.77	1.63
3-1/2	4.000	8	1.25	0.82	1.68
4	4.500	8	1.30	0.84	1.73
5	5.563	8	1.41	0.94	1.84
6	6.625	8	1.51	0.96	1.95
8	8.625	8	1.71	1.06	2.15
10	10.750	8	1.93	1.21	2.36
12	12.750	8	2.13	1.36	2.56

Table abridged from ANSI Standard B2.1–1968.

Maximum Non-Shock Hydraulic Working Pressure

Temp. °F	Working Pressure PSI							
	125#	150#	250#	300#	600#	900#	1500#	2500#
−20 to 100	200	285	500	740	1480	2220	3705	6170
200	190	260	460	675	1350	2025	3375	5625
300	165	230	375	655	1315	1970	3280	5470
400	140	200	290	635	1270	1900	3170	5280
500	—	170	—	600	1200	1795	2995	4990
600	—	140	—	550	1095	1640	2735	4560

NOTES: 125# and 250# from A.N.S.I. B16.1-1989 for ASTM A126, Class B, 1-12" size.
Others from A.N.S.I. B16.5-1988 for ASTM A105.

Properties of Welded and Seamless Steel Pipe

Size, nominal and outside dia, in.	Identification		Wall thickness, in.	ID.in.	Inside area, sq.in.	Wt/ft, lb	Wt. of water, lb/ft.	External surface, sq.ft/ft
	Schedule No.	Standard, X-strong, XX-strong						
⅛ (0.405)	40	STD	0.068	0.269	0.0568	0.244	0.025	0.106
	80	XS	0.095	0.215	0.0364	0.314	0.016	
¼ (0.540)	40	STD	0.088	0.364	0.1041	0.424	0.045	0.141
	80	XS	0.119	0.302	0.0716	0.535	0.031	
⅜ (0.675)	40	STD	0.091	0.493	0.1910	0.567	0.063	0.177
	80	XS	0.126	0.423	0.1405	0.738	0.061	
½ (0.840)	40	STD	0.109	0.622	0.3040	0.850	0.132	0.220
	80	XS	0.147	0.546	0.2340	1.087	0.101	
	160	......	0.188	0.464	0.1691	1.311	0.073	
		XXS	0.294	0.252	0.0499	1.714	0.022	
¾ (1.050)	40	STD	0.113	0.824	0.5330	1.130	0.230	0.275
	80	XS	0.154	0.742	0.4330	1.473	0.187	
	160	......	0.219	0.612	0.2942	1.944	0.127	
		XXS	0.308	0.434	0.1479	2.440	0.063	
1 (1.315)	40	STD	0.133	1.049	0.8640	1.678	0.374	0.344
	80	XS	0.179	0.957	0.7190	2.171	0.311	
	160	......	0.250	0.815	0.5217	2.840	0.226	
		XXS	0.358	0.599	0.2818	3.659	0.122	
1¼ (1.660)	40	STD	0.140	1.380	1.495	2.272	0.647	0.434
	80	XS	0.191	1.278	1.283	2.996	0.555	
	160	......	0.250	1.160	1.057	3.764	0.457	
		XXS	0.382	0.896	0.630	5.214	0.273	
1½ (1.900)	40	STD	0.145	1.610	2.036	2.717	0.882	0.497
	80	XS	0.200	1.500	1.767	3.631	0.765	
	160	......	0.281	1.338	1.406	4.858	0.610	
		XXS	0.400	1.100	0.950	6.408	0.412	
2 (2.375)	40	STD	0.154	2.067	3.355	3.65	1.45	0.622
	80	XS	0.218	1.939	2.953	5.02	1.28	
	160	......	0.344	1.687	2.235	7.46	0.97	
		XXS	0.436	1.503	1.774	9.03	0.77	
2½ (2.875)	40	STD	0.203	2.469	4.788	5.79	2.07	0.753
	80	XS	0.276	2.323	4.238	7.66	1.83	
	160	......	0.375	2.125	3.547	10.01	1.54	
		XXS	0.552	1.771	2.464	13.70	1.07	
3 (3.500)	40	STD	0.216	3.068	7.393	7.58	3.20	0.916
	80	XS	0.300	2.900	6.605	10.25	2.86	
	160	......	0.438	2.624	5.407	14.31	2.34	
		XXS	0.600	2.300	4.155	18.58	1.80	
3½ (4.000)	40	STD	0.226	3.548	9.886	9.11	4.28	1.047
	80	XS	0.318	3.364	8.888	12.51	3.85	
4 (4.500)	40	STD	0.237	4.026	12.730	10.79	5.51	1.178
	80	XS	0.337	3.826	11.497	14.98	4.98	
	120	......	0.438	3.624	10.315	18.98	4.47	
	160	......	0.531	3.438	9.283	22.52	4.02	
		XXS	0.674	3.152	7.803	27.54	3.38	
5 (5.563)	40	STD	0.258	5.047	20.006	14.62	8.66	1.456
	80	XS	0.375	4.813	18.194	20.78	7.87	
	120	......	0.500	4.563	16.353	27.04	7.08	
	160	......	0.625	4.313	14.610	32.96	6.32	
		XXS	0.750	4.063	12.966	38.55	5.62	
6 (6.625)	40	STD	0.280	6.065	29.90	18.97	12.5	1.734
	80	XS	0.432	5.761	26.07	28.57	11.3	
	120	......	0.562	5.501	23.77	36.42	10.3	
	160	......	0.719	5.187	21.13	45.34	9.2	
		XXS	0.864	4.897	18.83	53.16	8.1	
8 (8.625)	20	......	0.250	8.125	51.8	22.36	22.5	2.258
	30	......	0.277	8.071	51.2	24.70	22.2	
	40	STD	0.322	7.981	50.0	28.55	21.6	
	60	......	0.406	7.813	47.9	35.66	20.8	
	80	XS	0.500	7.625	45.7	43.39	19.8	
	100	......	0.594	7.437	43.4	50.93	18.8	
	120	......	0.719	7.187	40.6	60.69	17.6	
	140	......	0.812	7.001	38.5	67.79	16.7	
		XXS	0.875	6.875	37.1	72.42	16.1	
	160	......	0.906	6.813	36.5	74.71	15.8	
10 (10.750)	20	......	0.250	10.250	82.5	28.04	35.9	2.814
	30	......	0.307	10.136	80.7	34.24	35.0	
	40	STD	0.365	10.020	78.9	40.48	34.1	
	60	XS	0.500	9.750	74.7	54.74	32.3	
	80	......	0.594	9.562	71.8	64.40	31.1	
	100	......	0.719	9.312	68.1	77.00	29.5	
	120	......	0.844	9.062	64.5	89.27	27.9	
	140	XXS	1.000	8.750	60.1	104.13	26.1	
	160	......	1.125	8.500	56.7	115.65	24.6	
12 (12.750)	20	......	0.250	12.250	118.0	33.38	51.3	3.338
	30	......	0.330	12.090	114.8	43.77	49.7	
		STD	0.375	12.000	113.1	49.56	48.9	
	40		0.406	11.938	111.9	53.56	48.5	
		XS	0.500	11.750	108.4	65.42	46.9	
	60	......	0.562	11.626	106.2	73.22	46.0	
	80	......	0.688	11.374	101.6	88.57	44.0	
	100	......	0.844	11.062	96.1	107.29	41.6	
	120	XXS	1.000	10.750	90.8	125.49	39.3	
	140	......	1.125	10.500	86.6	139.68	37.5	
	160	......	1.312	10.126	80.5	160.33	34.9	

From ANSI Standard, Wrought Steel and Wrought Iron Pipe, B36.10–1959, and data of Crane Co. NOTE: Two systems of rating pipe wall thickness are utilized. The newer schedule numbers correspond to definite pressures-stress ratios and are expressed simply as follows:

$$\text{Schedule No.} = 1{,}000 \times \frac{p}{s}$$

where p = internal pressure, psig
s = allowable fiber stress, psi

Properties of the traditional designation for pipe entitled "standard," "extra strong," and "double extra strong" are also shown in the tables.

Pressure Loss Due to Pipe Friction for Standard Steel Pipe

The pressure loss curves on Pages 34 to 46 inclusive are useful to determine the friction losses for various fluids flowing through Schedule 40 new steel pipe. Fluid viscosities are given in Seconds Saybolt Universal (SSU) and centistokes (CS). The horizontal scale at the bottom of each curve shows the rate of flow in gallons per minute and, at the top, the corresponding average velocity in the pipe in feet per second. The vertical scales of friction loss modulus are converted to pressure loss in pounds per square inch or to head loss in feet of liquid as described on the curves.

For example, look at the curve on Page 34. To find the pressure loss for ½" pipe for 5 GPM and a liquid with 200 SSU viscosity, enter the curve at the bottom at 5 GPM and note the intersection of the 5 GPM vertical line with the 200 SSU diagonal reference line. A modulus reading of 40 per 100 feet of pipe length can be found at the left margin. The modulus reading is converted to PSI or feet of liquid loss as described on the curve margin. At the top of the chart directly above 5 GPM, a reading of 5.3 feet per second can be found, which is the velocity for 5 GPM flow in a ½" pipe. All of the other curves for other pipe sizes are in the same form and may be read in the same manner.

In arriving at the pressure loss of 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. Entrance losses from sump to suction pipe are generally less than one foot of head at velocities below 10 ft/sec. Consult a more detailed handbook for losses due to sudden enlargements or contractions.

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

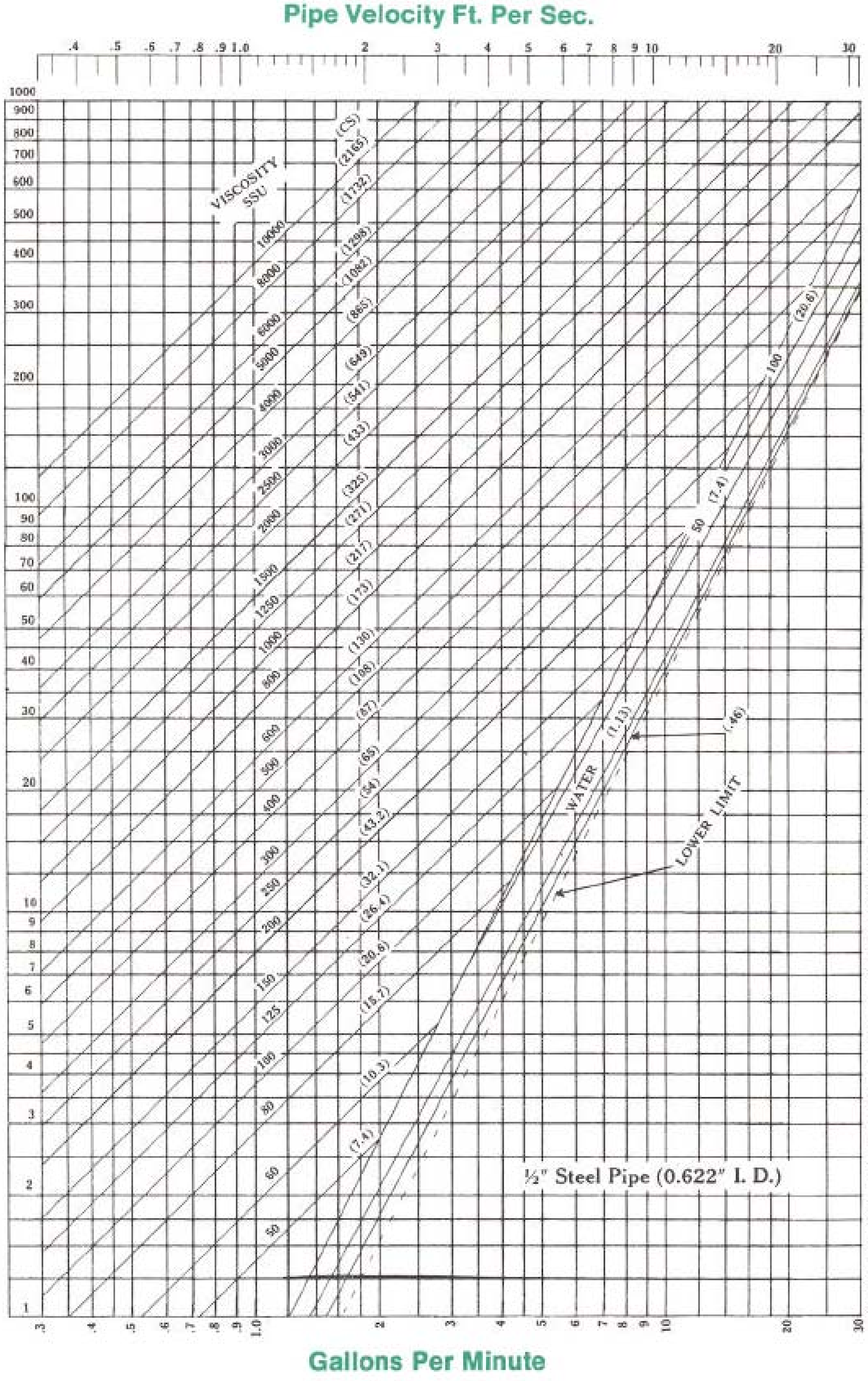
Table Gives Equivalent Lengths of Straight Pipe in Feet (Laminar Flow)

Type of Fitting	Nominal Pipe Diameter													
	½	¾	1	1¼	1½	2	2½	3	4	5	6	8	10	12
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	6.4
Globe Valve (open)	17	22	27	38	44	53	68	80	120	140	170	220	280	336
Angle Valve (open)	8	12	14	18	22	28	33	42	53	70	84	120	140	168
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	30
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	26.0
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	20.2
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	20.2
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	61.0
Return Bend (180°)	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	74.0
45° Elbow	.78	.97	1.23	1.6	1.9	2.4	2.9	3.6	4.7	5.9	7.1	9.4	11.8	14.1

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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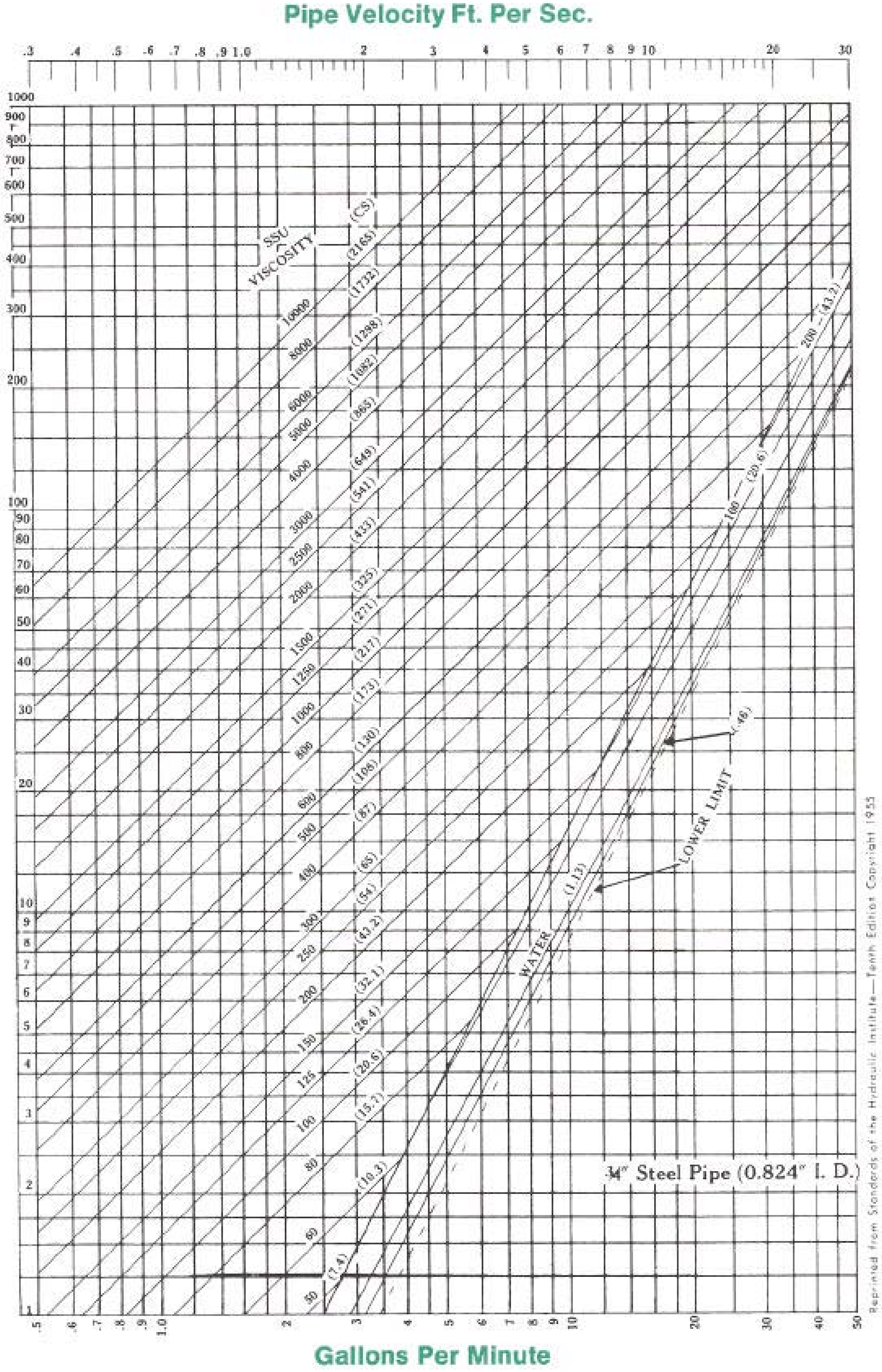

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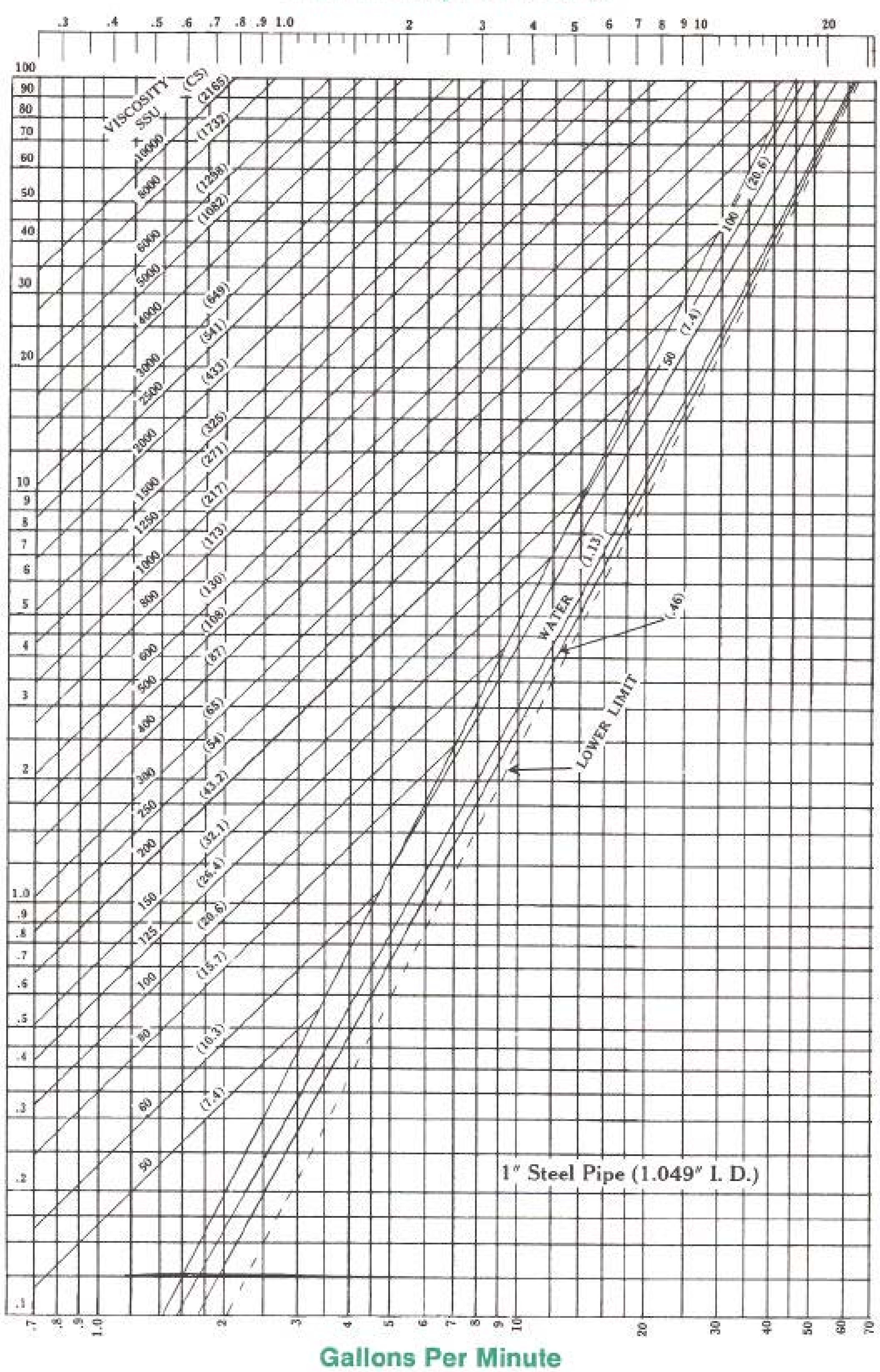

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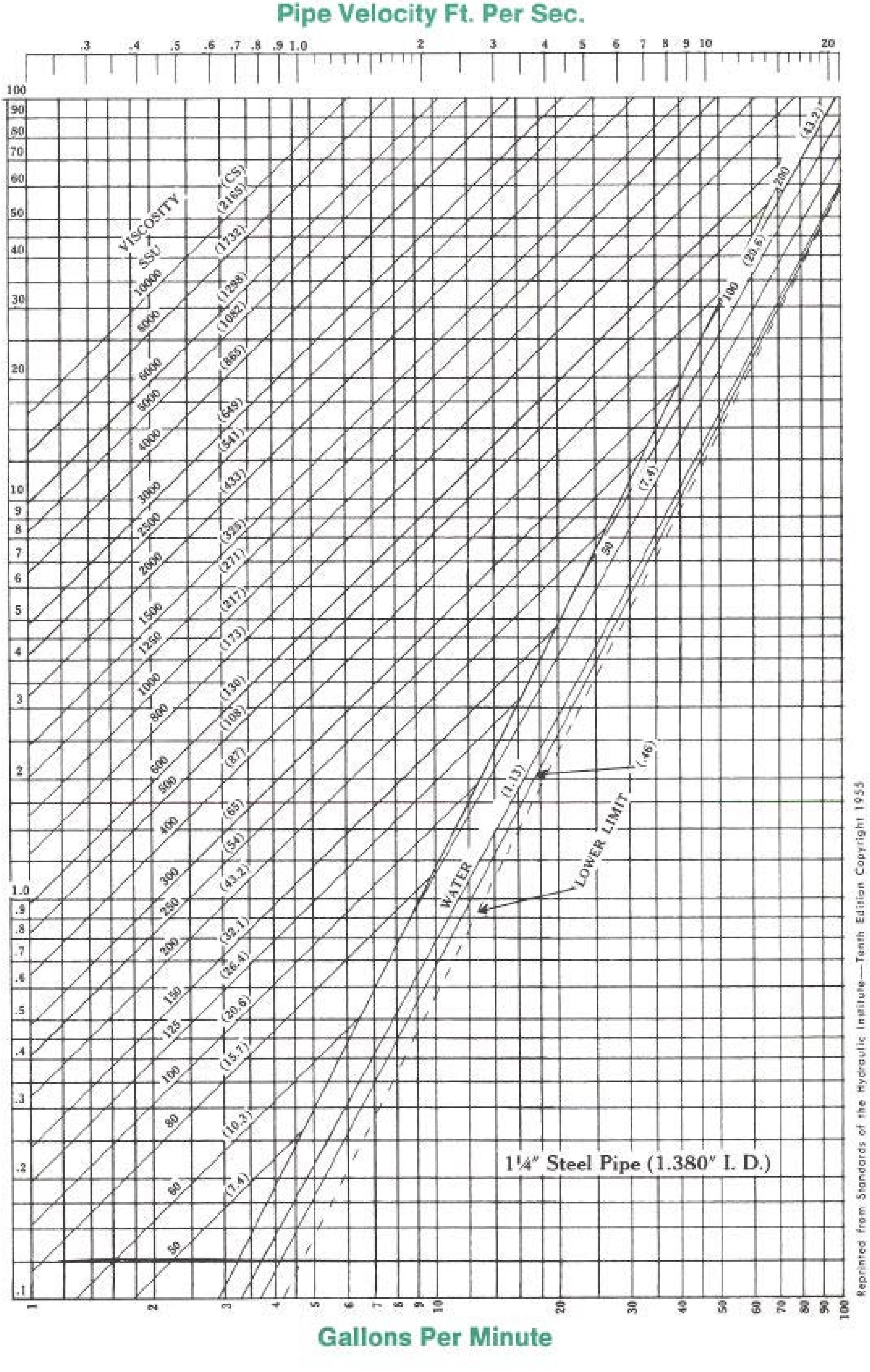

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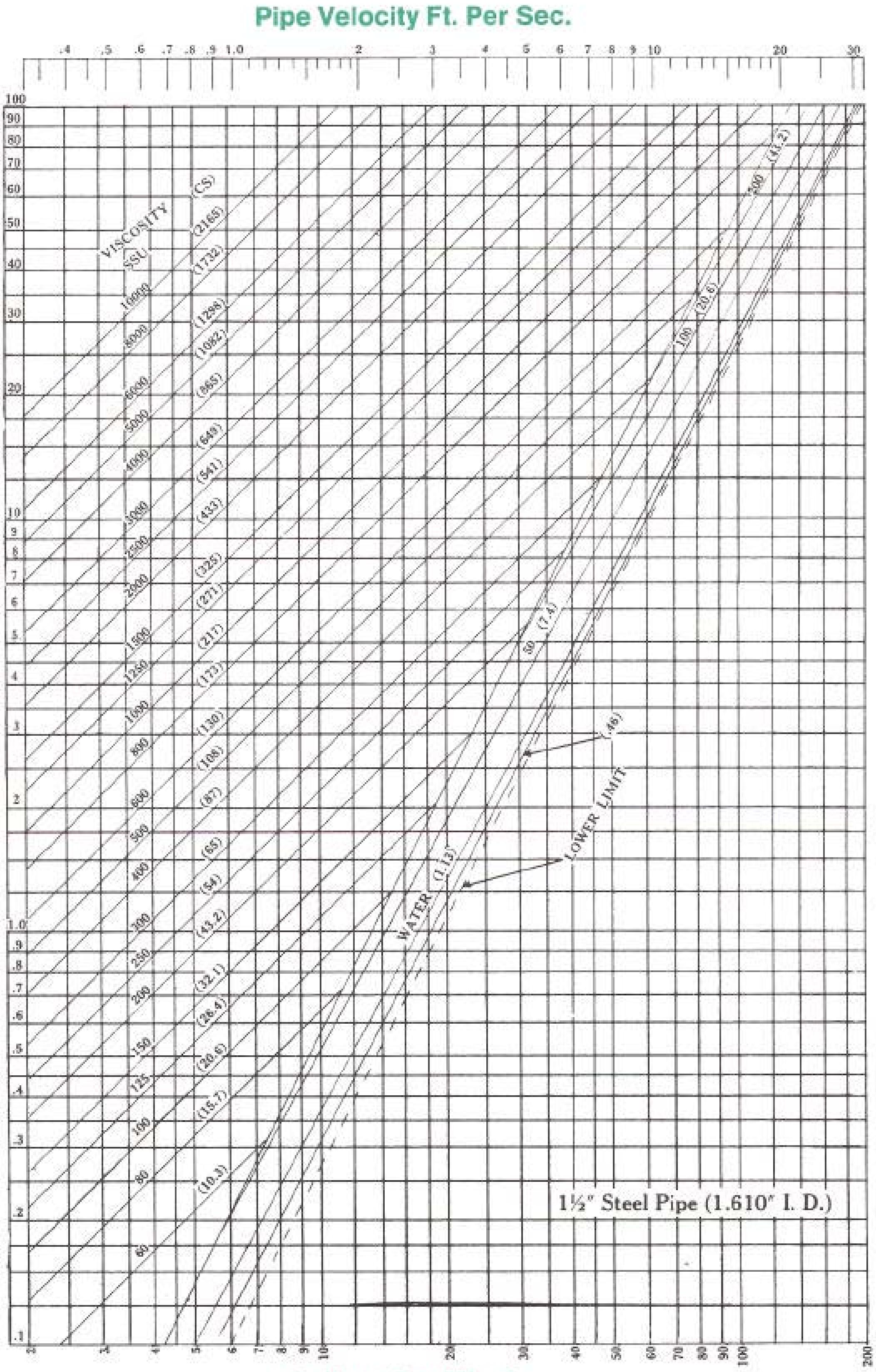

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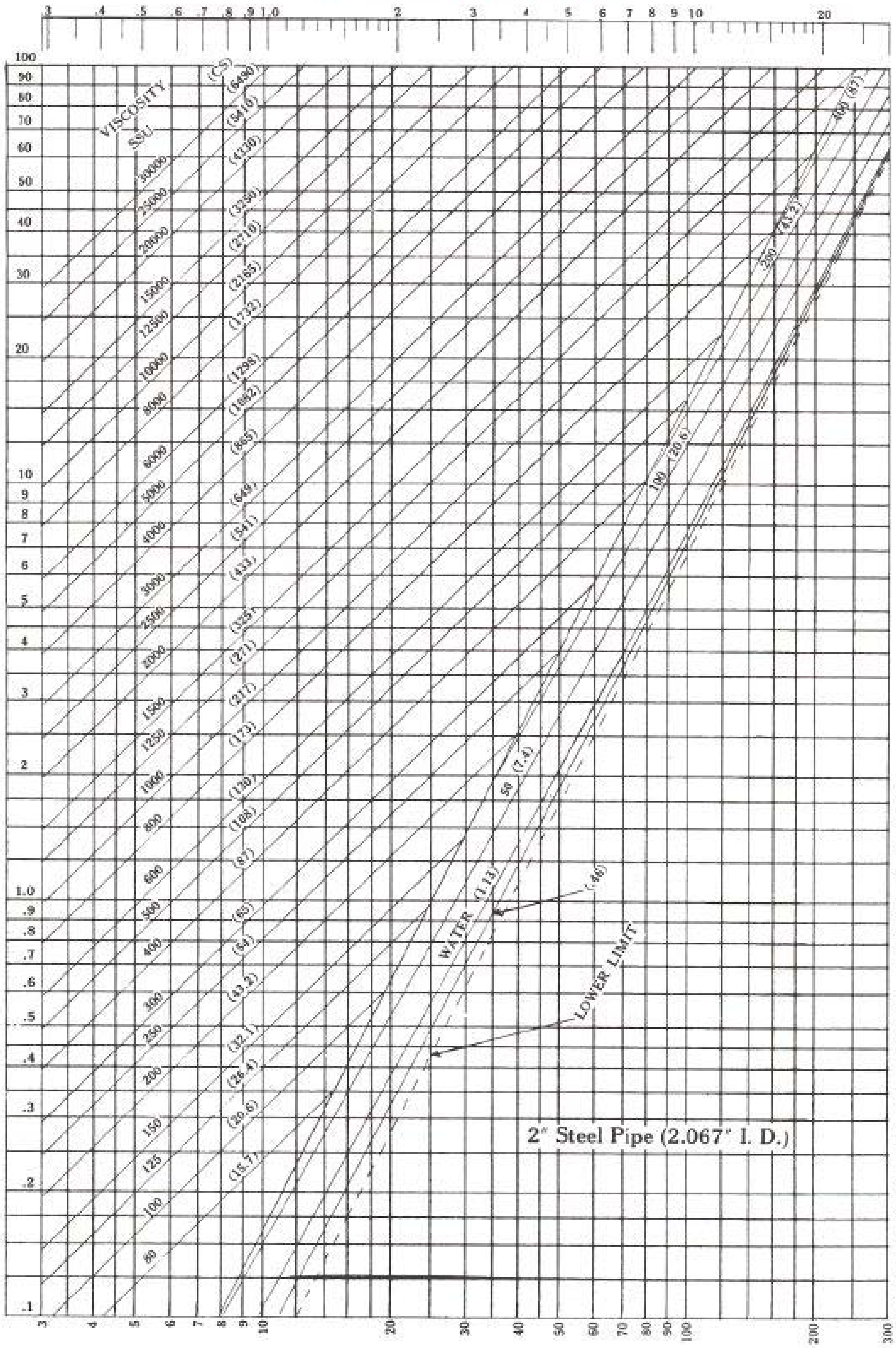
Pipe Velocity Ft. Per Sec.



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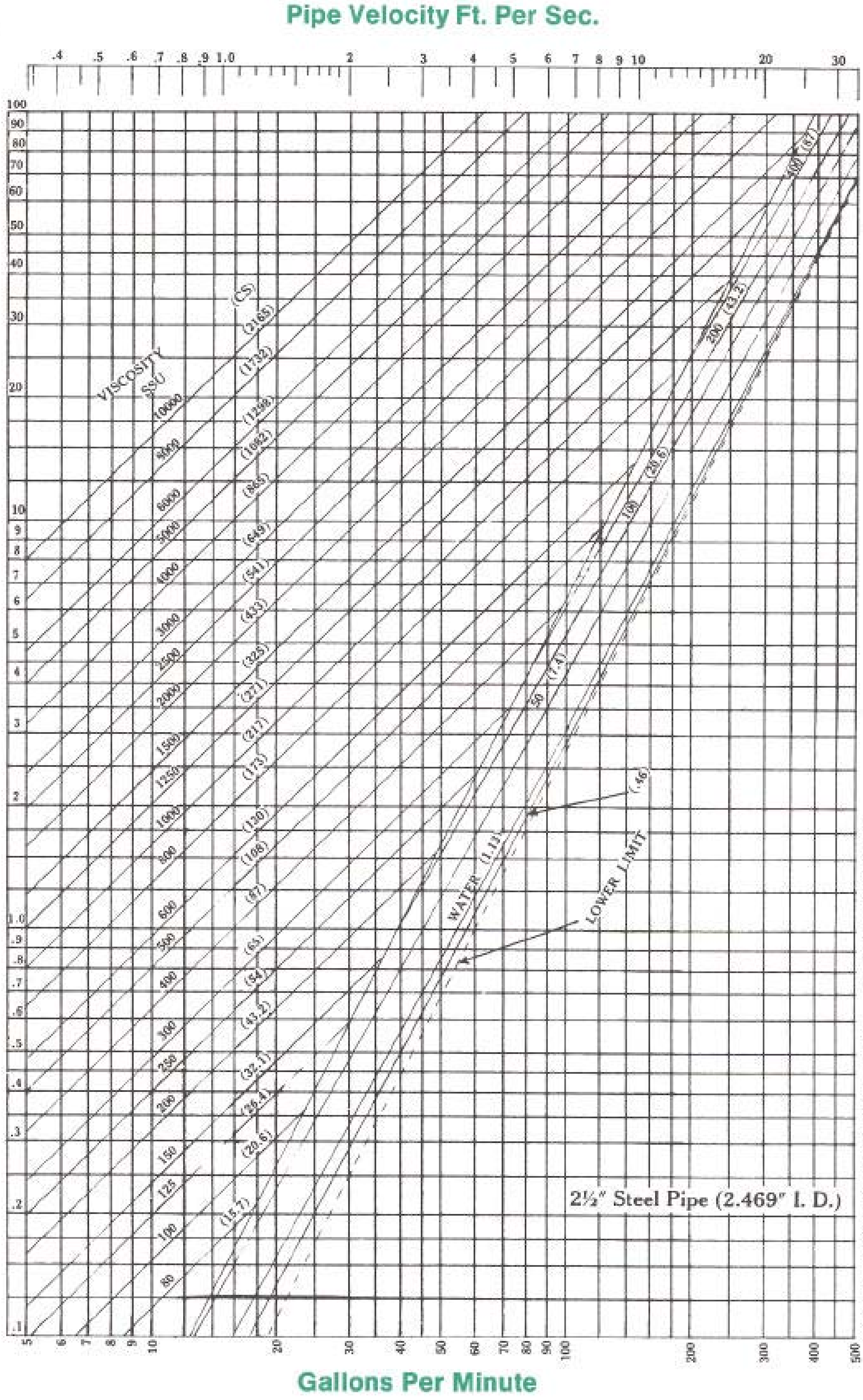
Gallons Per Minute

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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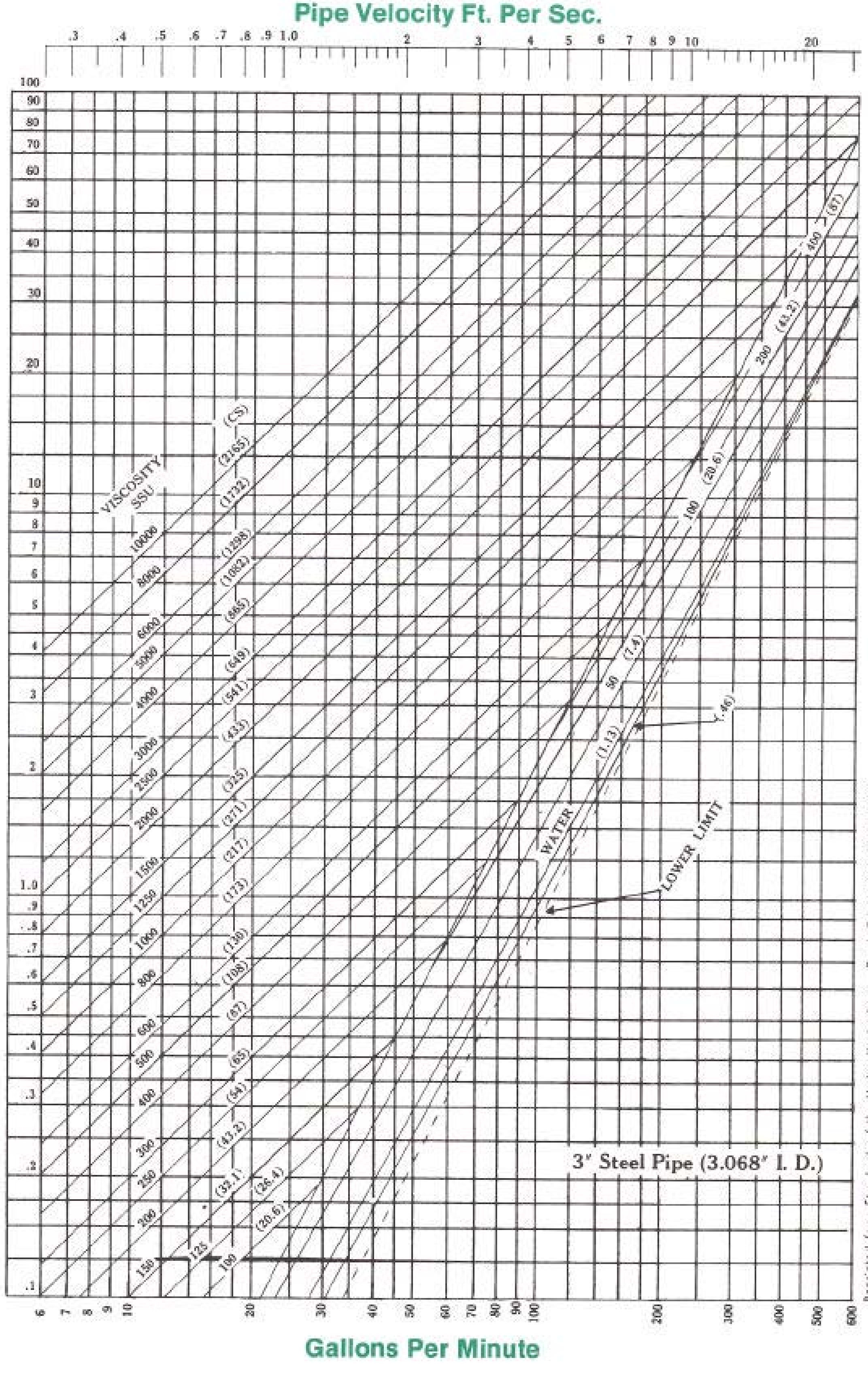

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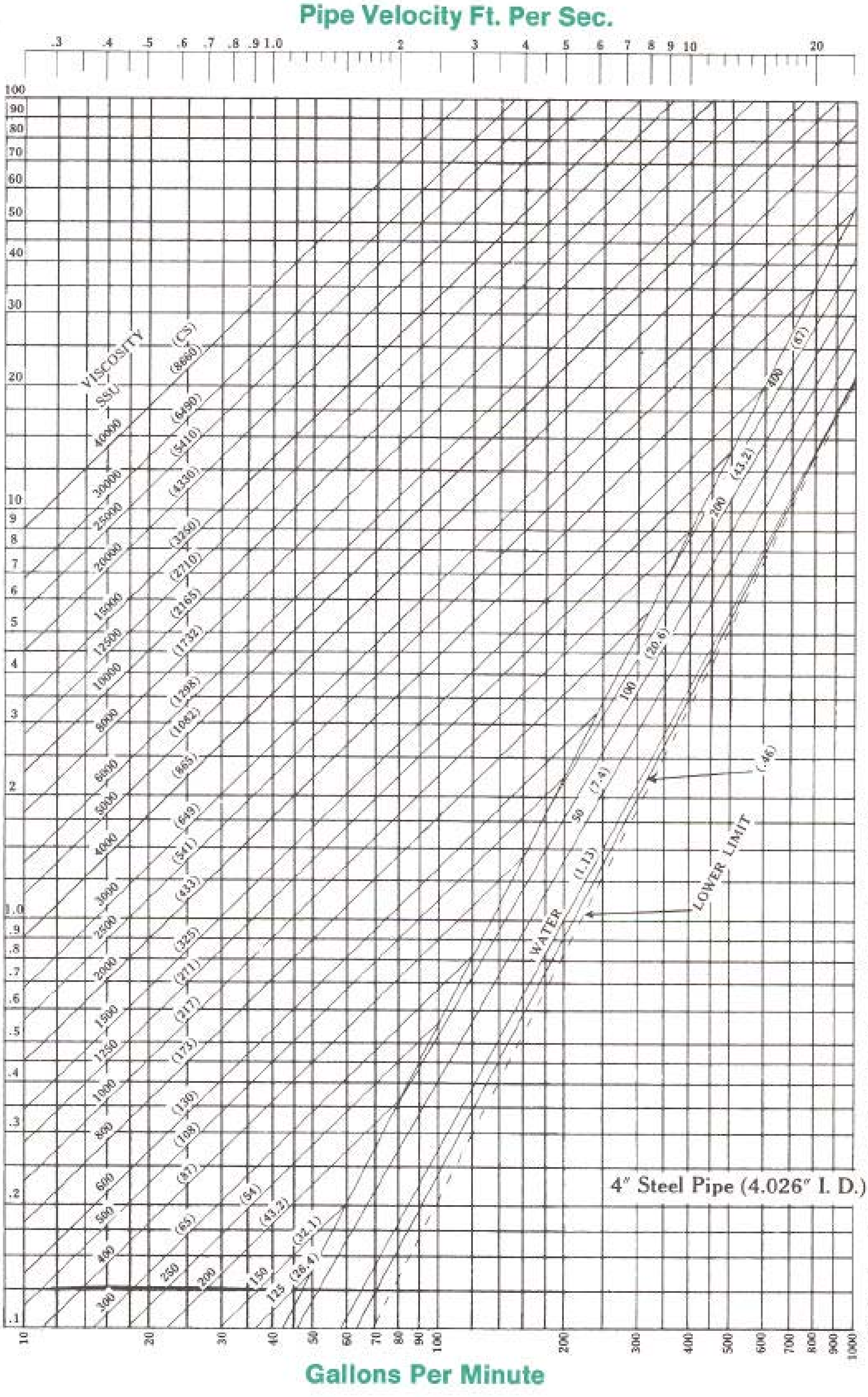

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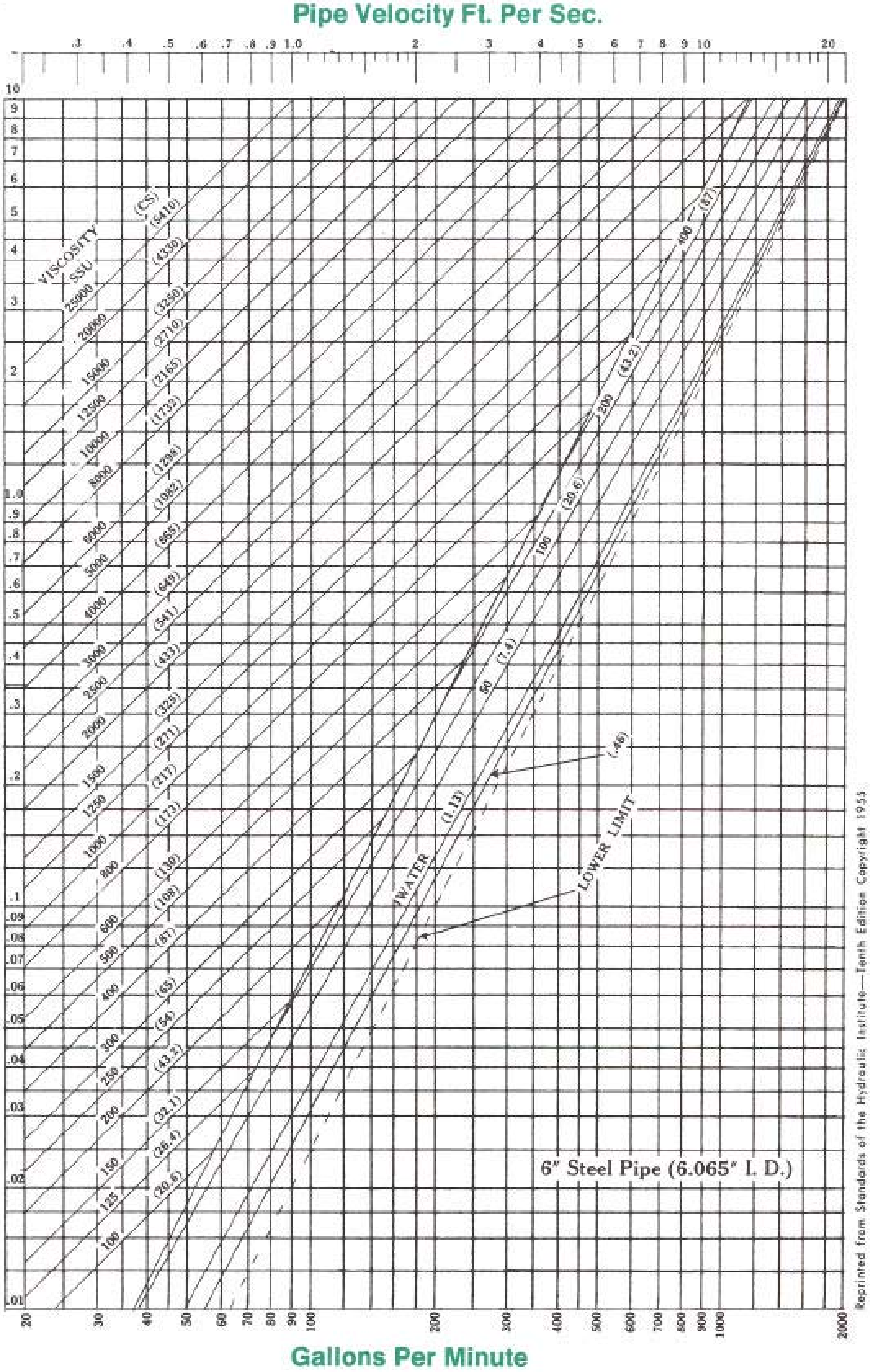

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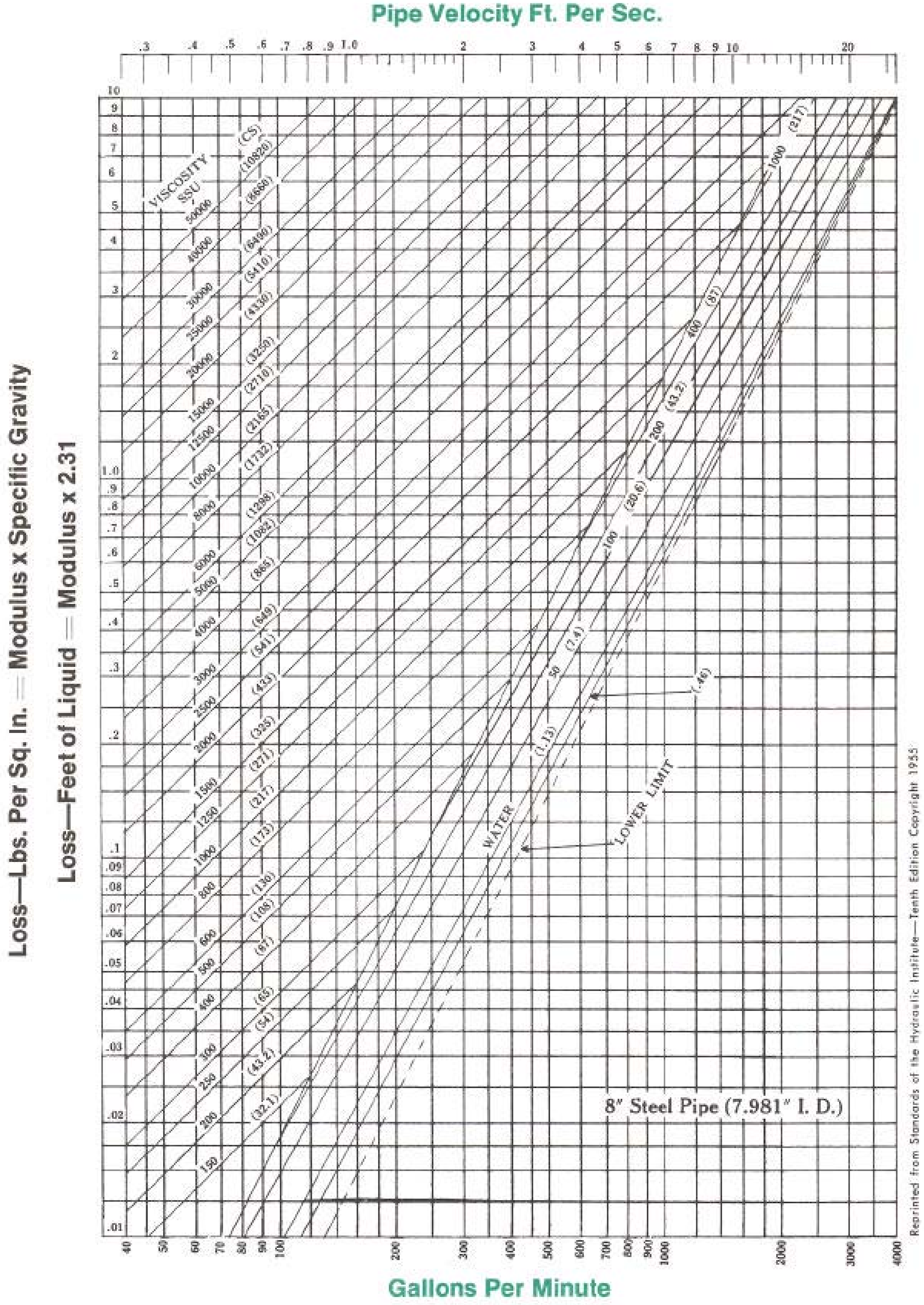
Friction Loss Modulus for 100 Feet of Pipe



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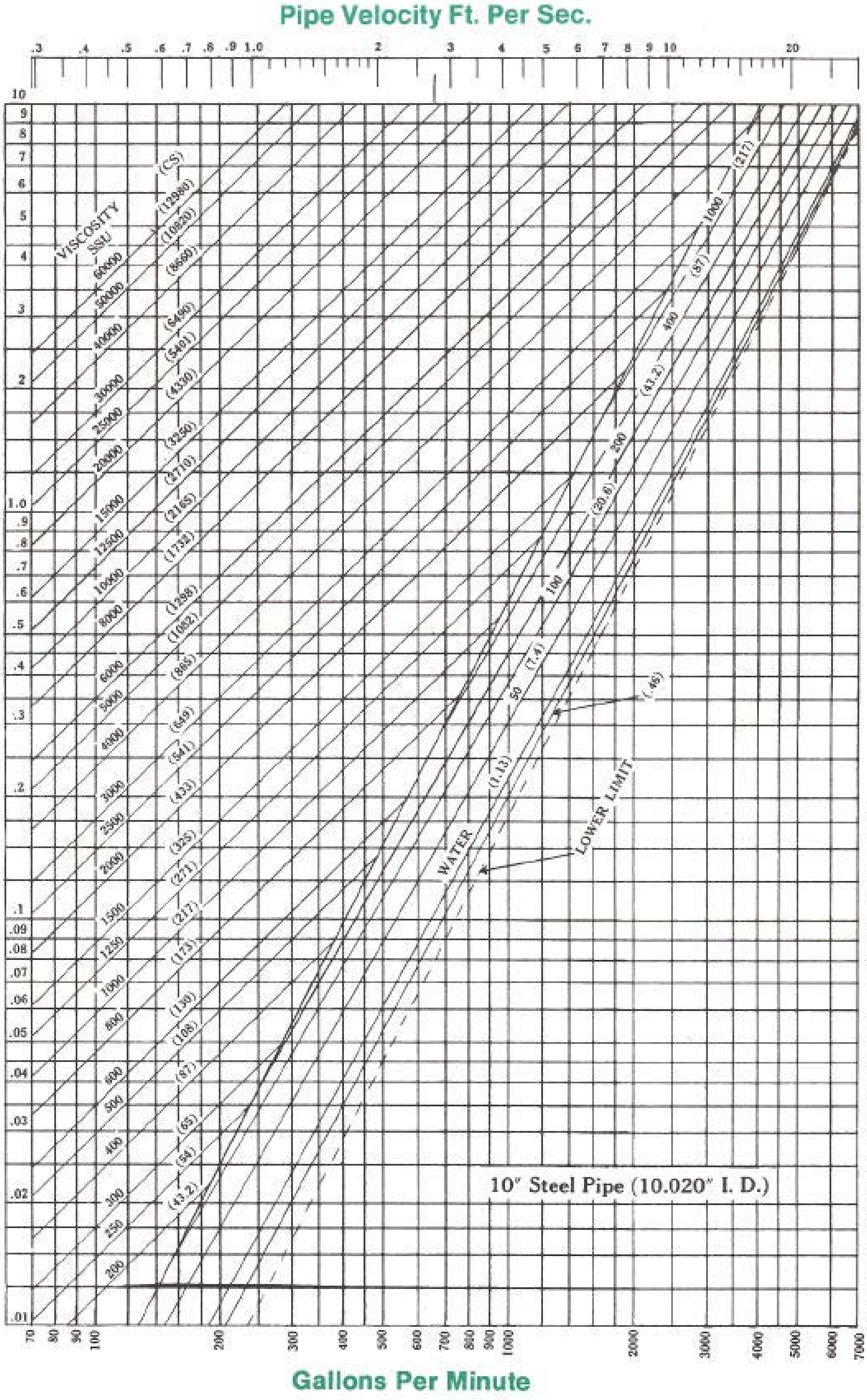

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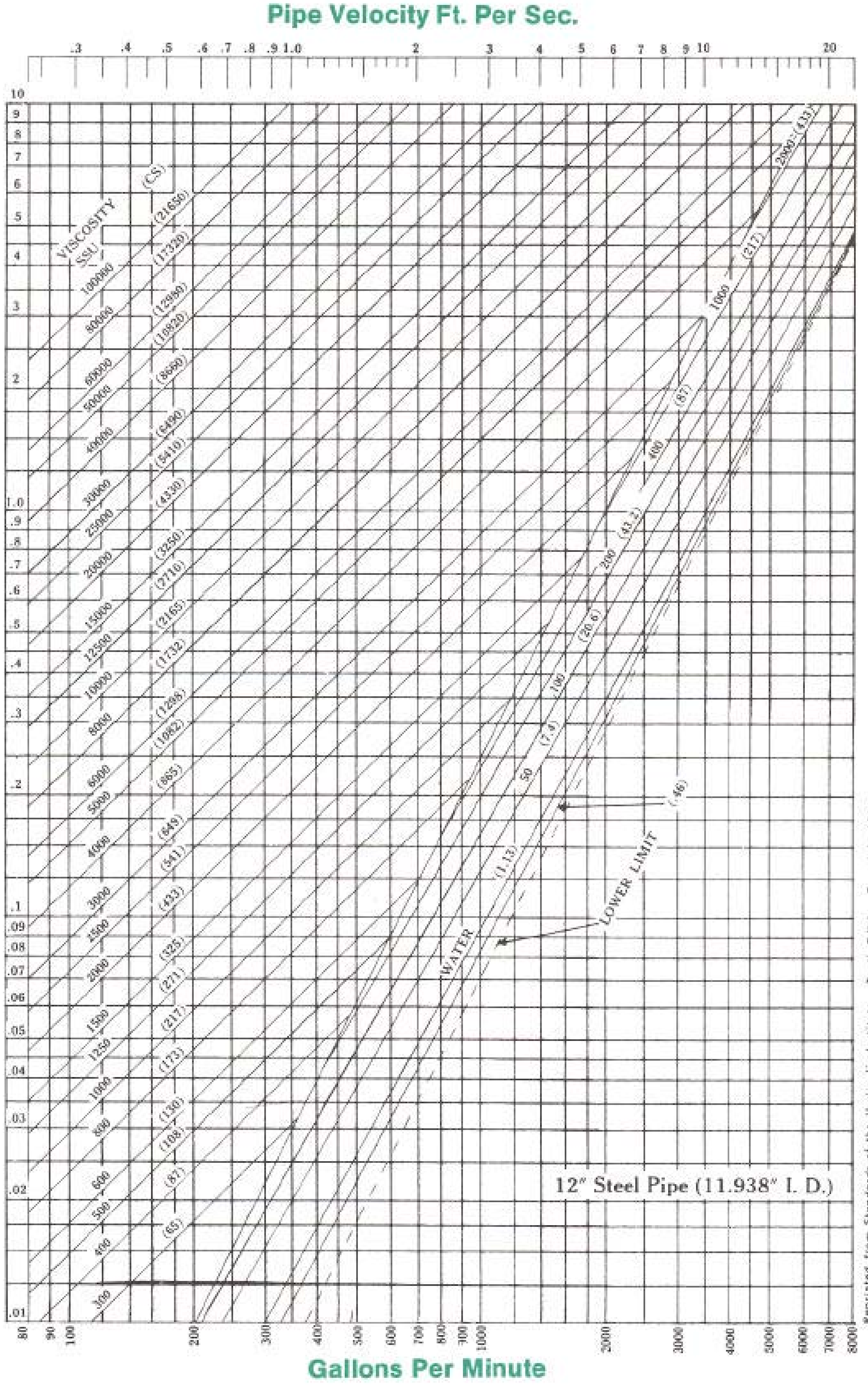

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IMO® Positive Displacement Rotary Pumps

IMO Pump Division, Imo Industries Inc., is the world's largest designer and manufacturer of high performance, positive displacement rotary screw pumps. It produces three positive displacement designs - GTS,™ geared twin screw; CIG®, crescent internal gear; and IMO®, three-screw rotary pumps. Backed with more than 50 years of high performance pumping experience, this precision engineered product line includes pump models to handle flows from one to 8500 gpm, at pressures from 25 to 5000 psi and fluid viscosities from 0.3 to more than a million SSU.

IMO pumps have achieved international recognition for their design simplicity, wide fluid viscosity capabilities, pulsation-free operation, and unusually high reliability over extended periods. Today, IMO has more than one-half million pumps in operation throughout the world. Applications are in a broad range of industries for non-corrosive chemical processing, fuel oil/fluid transfer, lube oil service, hydraulic power, hydraulic sealing and turbine governor use.

GTS™ Geared Twin Screw Pumps

GTS geared twin screw pumps find wide application in petroleum production and refining, power generation, petro-chemical, chemical, and many other process industries. They are used in applications wherever large volumes (up to 8500 gpm) of fuels, feeds, intermediates and end products have to be pumped at pressures up to about 500 psi. These pumps can handle liquids with an extremely wide range of viscosities, vapor pressures and densities at temperatures up to and in excess of 600°F (315°C). Because the profile of the pumping screws results in minimum fluid shear, GTS pumps can also be used with paints, greases and other materials exhibiting non-Newtonian rheological behavior.

Typical Applications

Loading, Off-Loading, Cargo Handling and Transfer of:

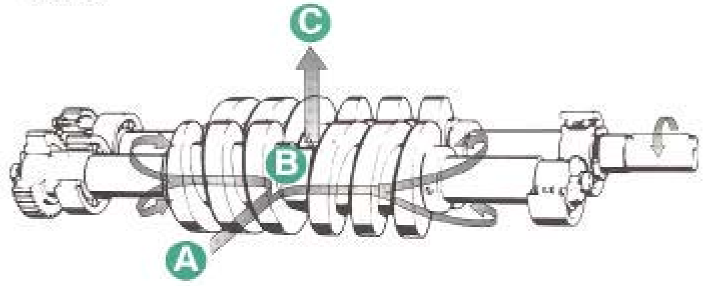
- Crude oil, light, middle and heavy distillates
- Residual oils, asphalts and tars
- Aromatic, naphthenic and aliphatic intermediates and products
- Fats, oils, waxes and soaps
- Molasses, glycerol, glues, paints, resins and polymers

Design and Operation

The GTS is a geared twin screw positive displacement rotary pump. Externally lubricated timing gears and bearings combined with no internal metal-to-metal contact enable the pumping of non-lubricating fluids as well as aggressive media. This arrangement also allows dry running for unloading and stripping tanks, or for evacuating long suction lines.

The basic pump design is double ended (see diagram at right.) The fluid entering the inlet port (A) is split into two equal portions (B) at the inlet end of each set of screws. As the rotating screws intermesh, transfer chambers are formed trapping and conveying the fluid axially to the discharge chamber (C) in the center of the pump. It is also possible to reverse direction of flow by reversing the direction of rotation.

GTS pump flow capacity is determined by the lead angle of the screw set. For a given case size, interchangeable screw sets with varying lead angles are available. This enables the flow capability of any one pump to be extended to cover a wide range of requirements.



Features

- **Dry Running Capability.** External bearings and timing gears plus no internal metal-to-metal contact allow the pumping of non-lubricating fluids as well as dry running.
- **Contamination Tolerant.** Two-screw, contact-free rotor design handles pumpage with a higher percent of contaminants and/or abrasives without the need for costly filtration systems.
- **Wide Viscosity Range.** Two-screw, positive displacement design provides uniform, predictable performance regardless of fluid viscosity or temperature.
- **Variable Capacity.** Shell design permits use of interchangeable screw sets with varying lead angles to provide a wide range of flow, pressure and life capabilities for each case size.
- **Self-Priming.** With low inlet pressure requirements (NPSH) and inlet port located above the screws, GTS pumps are self-priming under marginal or interrupted flow conditions. They cannot become vapor-locked and are capable of evacuating long, empty suction lines.
- **No Cooling Required.** High efficiency timing gears plus an oversized lube oil reservoir eliminate the need for costly external cooling systems.
- **High Speed Operation.** May be direct coupled to more efficient, less costly drivers without the use of gear reduction systems.
- **Low Maintenance.** Opposed-flow design minimizes axial thrust and thrust bearing wear. Timing gears are ground and hardened for extended life. Replaceable rotor housings allow field repair to reduce downtime and expense.

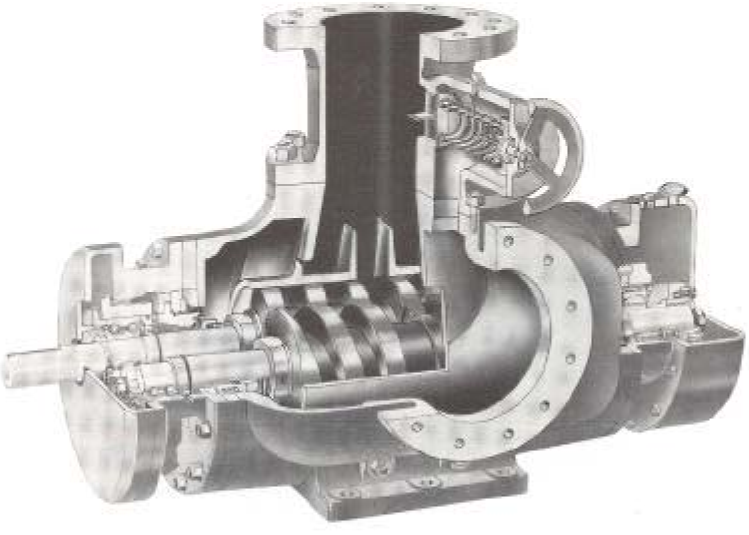
Options

- **Built-in Heating Chamber.** Keeps pump at optimum temperature for start-up when pumping viscous fluids.
- **Integral Relief Valve.** Protects pump from inadvertent overpressure, automatically relieves inlet flow when working pressure is exceeded.
- **High Temperature and High Pressure Models.** Special purpose units are available for operation at temperatures up to 650° and pressures to 650 psi.

GTS Performance Characteristics

GTS™ Model No.	APPLICATION					SPECIFICATIONS								No. of Sizes
	Process/ Transfer	Fuel Oil Service	Lube Oil Service	Hydraulic Sealing	Fluid Power	Flow Range		Maximum Discharge Pressure		Maximum Speed	Maximum Power		
						US GPM	L/MIN	PSIG	BAR-G	RPM	HP	KW	
GTS-074	•	•				5-100	55-375	300	20	3600	50	37	5
GTS-133	•	•				50-750	190-2800	450	31	2400	150	110	6
GTS-208	•	•				450-2700	1700-10,200	450	31	1800	600	450	6
GTS-268	•	•				1100-4800	4100-18,000	300	20	1800	800	600	6
GTS-320	•	•				1400-5400	5300-20,500	300	20	1500	1000	745	5
GTS-400	•	•				2500-8500	9500-32,000	300	20	1500	2000	1500	5

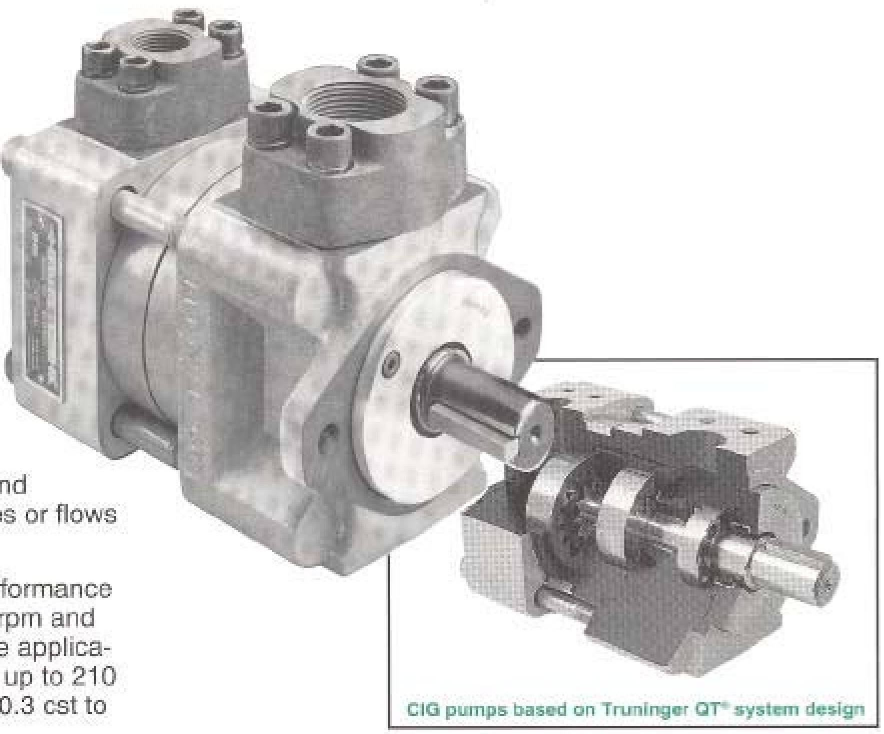
Note: All capabilities are not available in all pumps. Consult your local IMO office or representative for detailed information (see back cover). Special and custom designs are invited.



CIG® Crescent Internal Gear Pumps

Designed for quiet, long lasting, low lubricity operation, the CIG has an operational life of over 30,000 hours - 10 times longer than piston, vane or standard gear pumps. Its noise level is typically 68 db (A) compared to these conventional designs which usually operate at 90 db (A). With ultra-low pressure ripple (0.62% @ 4500 psi), the CIG produces a smooth, constant flow for precision applications. Compact, modular construction permits multi-staging and multi-pump applications to increase pressures or flows in single circuits or to power multiple circuits.

CIG pumps are available in over 1400 performance combinations with speed ranges up to 5700 rpm and can handle a wide range of high performance applications at pressures to 5000 psi and flow rates up to 210 gpm. Fluid viscosity capabilities range from 0.3 cst to over 2157 cst.



CIG pumps based on Truninger QT® system design

Typical Applications

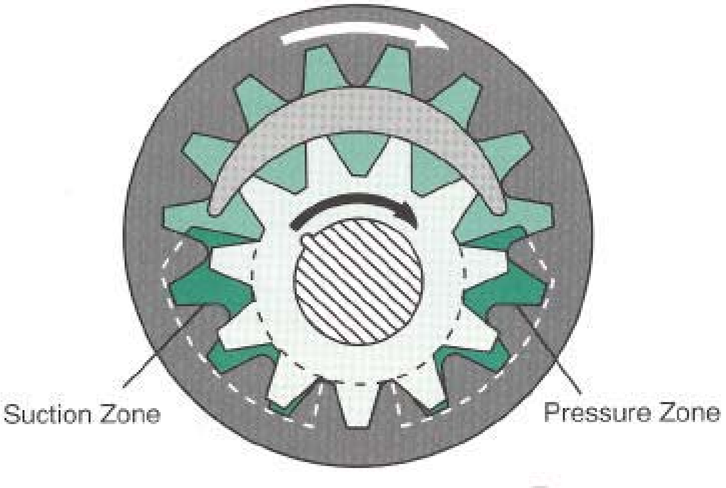
Constant Flow, Fluid Power and Fuel Oil Service Requirements for:

- Manufacturing machinery - machine tools, die casting, injection molding and extrusion machinery
- Turbine generators
- Jet engine test stands
- Processing systems - packaging, material handling, pulp and paper machinery
- High technology applications - computer and medical equipment
- Mobile equipment - mining and aircraft ground support equipment

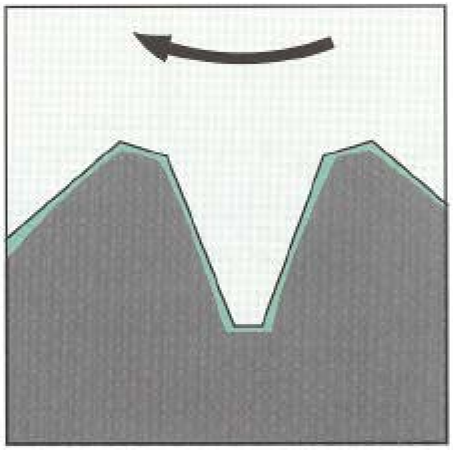
Design and Operation

The CIG pump is a positive displacement internal gear design. It uses a common shaft driving a pair of patented star-shaped gears for each stage. Hydraulic forces are absorbed on hydrodynamic oil films for long life. Gear tooth profile is such that "trapped volume" is nearly zero at the discharge zone, resulting in exceptionally quiet, ultra-low pulsation operation. The modular design allows stacking of gear sets to achieve high pressures without overstressing parts. It also allows the double pump arrangement - a pump with one inlet port, two independent outlet flows and one input shaft.

This is an important design advantage for applications normally requiring two separate pumps. Double CIG pumps can be used to increase flow in a single circuit or to power two circuits. In such applications, installation space and expense can be reduced due to fewer inlet-side piping requirements and the need for only one driver.



Extra Large Suction and Pressure Zones minimize fluid velocities to reduce noise levels and improve suction capabilities.



Patented Gear Tooth Profile, with essentially straight flanks, virtually eliminates "trapped oil" volume to provide ripple-free flow and low-noise operation.

Features

- **Multi-Stage Capability.** Modular design permits "stacking" of stages to increase pressure and flow ratings. Over 1400 performance combinations are available.
- **Quiet Operation.** Extra large suction and discharge areas minimize fluid velocities to maintain low airborn noise levels.
- **Constant Flow.** Excellent suction capabilities and self-priming characteristics allow fluids to fill the pump smoothly, reducing the possibility of cavitation.
- **Ultra-Low Pressure Ripple.** Patented gear tooth profile eliminates "trapped oil" volume and corresponding pressure pulsations to further enhance low noise operation.
- **Simple, Reliable Design.** Each stage consists of only two hydrodynamically supported moving parts which have a typical operating life of over 30,000 hours.
- **High Operating Efficiency.** Precision designed and manufactured, typical pump volumetric efficiencies are in the 90% range to reduce driver size and cooling requirements as well as overall operating costs.
- **Wide Fluid Viscosity Range.** Handle jet fuels, water glycols, phosphate ester, solvents, fuel oils, hydraulic oils and synthetic fluids with viscosities ranging from .3 cst to over 10,000 SSU.

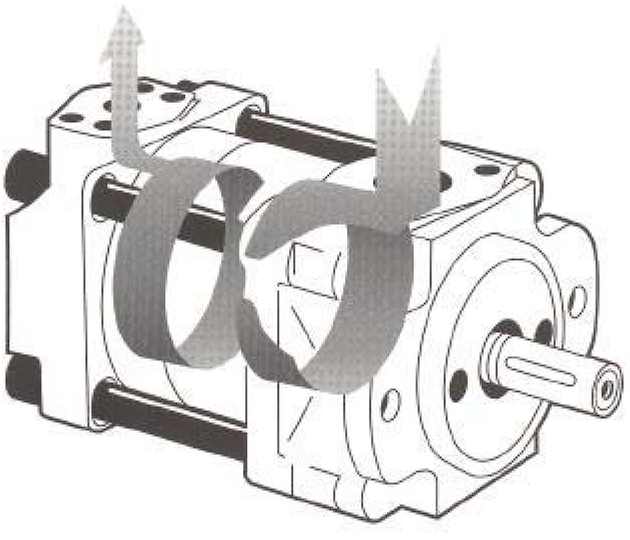
Options

- **Double Pump Combinations.** Two CIG pumps assembled on a common shaft with one suction port can increase flows in single circuits or increase power in two separate circuits.
- **Mechanical Shaft Seals.** Increase reliability for pumping kerosene, diesel fuel, JP-4, Jet-A, #2 fuel oil, Stoddard Solvent and methanol.
- **High Temperature Models.** Designed for use with fuels at temperatures up to 325° F and hydraulic oils at up to 350° F.
- **Ultra-Low Viscosity Models.** Allow pumping of low viscosity industrial turbine fuel from .3 cst to over 2157 cst.

CIG Performance Characteristics

CIG® Model No.	APPLICATION					SPECIFICATIONS							No. of Sizes
	Process/ Transfer	Fuel Oil Service	Lube Oil Service	Hydraulic Sealing	Fluid Power	Flow Range		Maximum Discharge Pressure		Maximum Speed	Maximum Power		
						US GPM	L/MIN	PSIG	BAR-G	RPM	HP	KW	
CIG-2		•			•	1-7	4-26	5000	333	5700	52	—	6
CIG-3		•			•	2-30	8-114	5000	333	4500	82	—	6
CIG-4		•			•	3-48	11-182	5000	333	3600	130	—	6
CIG-5		•			•	6-75	23-284	5000	333	2900	207	—	6
CIG-6		•			•	10-118	38-447	5000	333	2300	329	—	6
CIG-8		•			•	17-122	64-462	5000	333	1800	500	—	6

Note: All capabilities are not available in all pumps. Consult your local IMO office or representative for detailed information (see back cover). Special and custom designs are invited.



Typical Flowpath of a Two-Stage CIG.

IMO® Three-Screw Rotary Pumps

The unique IMO three-screw rotary design features only three moving parts and no wearing contact. This simplicity assures high reliability, maximum pumping efficiency and extended pump life with only minimal maintenance. The IMO pump produces non-pulsating flows against varying pressures over a very wide range of fluid viscosities. Operation is extremely quiet and is at standard motor speeds without the use of speed reduction gears.

A wide range of models are available with flow capacities from 1 to 3500 gpm, discharge pressures to 5000 psi and maximum operating speeds to 8000 rpm. When combined with I-MAG™ magnetic drive couplings, the inherent advantages of the IMO three-screw design permit leak-proof, low-maintenance process pumping of non-corrosive fluids.

Typical Applications

High Performance Pumping in Industrial, Petrochemical, Chemical, Oil Production/Refining, Marine and Electric Power Generation Applications for:

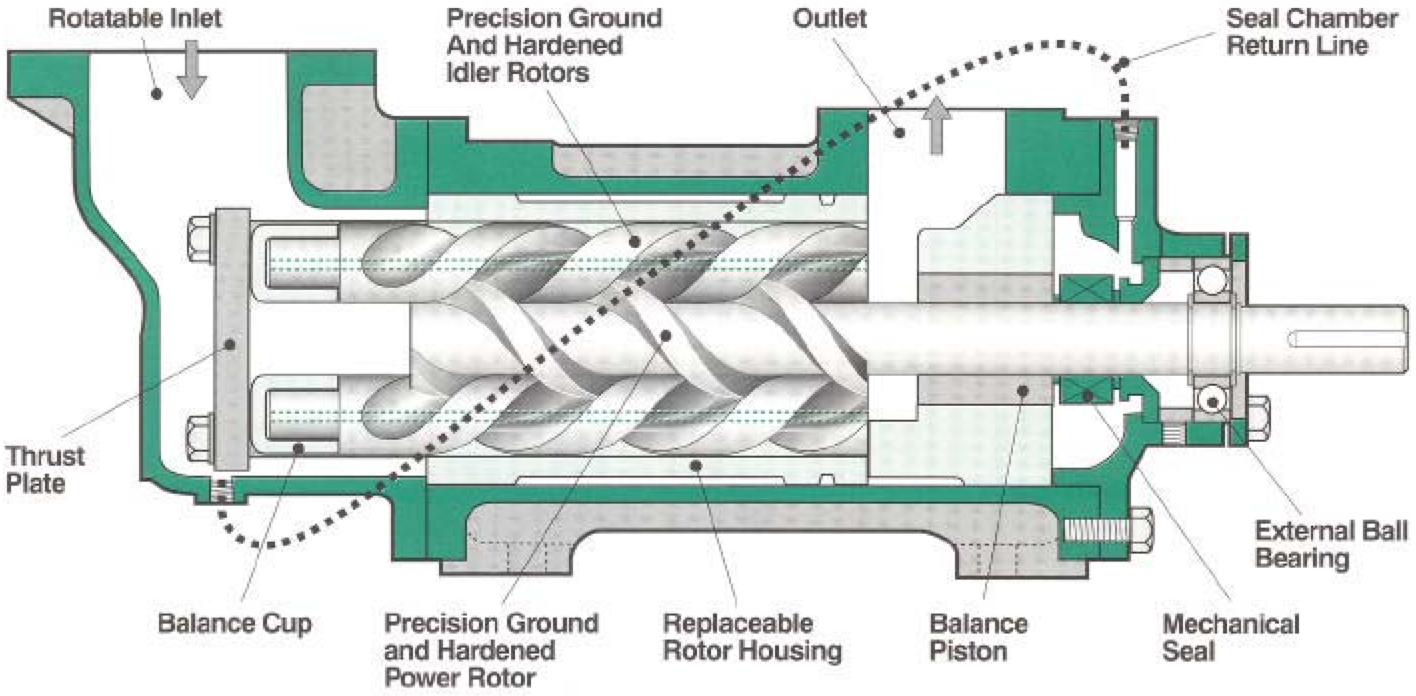
- Non-corrosive chemical processing
- Fuel oil/fluid transfer
- Lube oil service
- Hydraulic power
- Hydraulic sealing
- Turbine governor use
- High pressure coolant service

Design and Operation

The IMO rotary screw pump is the original three-screw pump. It is an axial flow, multi-rotor positive displacement design. There are only three moving parts: the power rotor (main screw) and two symmetrically opposed idler rotors. The three rotors are all precision ground screws, with a double lead design, meshing within a close fitting housing to efficiently and quietly deliver continuous, non-pulsating flows. The idler rotors, turned hydraulically by the fluid being pumped, act as rotating seals confining the pumpage in a succession of closures or stages. There is only a rolling action between the power rotor and the idler rotors. This eliminates noise and vibration. Wearing contact is non-existent because the power rotors are free to "float" in their respective bores on a hydrodynamic film created by the pumped fluid.

The IMO pump is balanced both axially and radially. Radial loads are eliminated because the double lead power rotor produces a symmetrical pressure keeping it centered in the bore. Axial loads, created by discharge pressures, are balanced by means of a balance piston at the pump discharge. An external bearing positions the power rotor and is protected from contaminated or low lubricity fluids.

These features allow the IMO pump to assure dependable performance and long life with minimal downtime and maintenance.



Features

- **Quiet Operation.** Typical airborne noise levels are below 75 db (A) SPL at 3 feet and are well within OSHA requirements.
- **Hardened and Precision Ground Rotors.** Resist wear due to abrasion and distortion to provide extended pump life. The need for matched rotor sets is eliminated.
- **Hydraulically Balanced.** Hydraulically induced axial and radial bearing loads are reduced or completely eliminated for low maintenance operations over long service periods.
- **Replaceable Rotor Housings.** Allow complete field overhaul to the "as new" condition without re-machining or the use of special tools.
- **High Speed Capability.** Low axial flow velocity and the absence of wearing contact permits direct coupling to high speed drivers. This reduces initial cost and improves operating efficiency.
- **External Bearings.** Antifriction bearing is externally located and protected from the pumped fluid. It cannot be damaged by contaminants or low lubricity.

Options*

- **Integral Relief Valves.** Automatically relieve inlet flow when working pressure is exceeded.
- **Component Metallurgy.** Choice of alloys for casings, rotor housings and rotors to meet specific operating conditions.
- **Pump/Drive Packages.** Turnkey packages complete with specified drivers. Includes choice of variable speed and variable frequency motors.
- **Mounting Configurations.** Face or foot mounting in either horizontal or vertical positions without sacrifice in pump performance.
- **I-MAG® Magnetic Drive Couplings.** Permit leak-proof pumping of non-corrosive fluids.

***All options are not available in all pumps. Consult your local IMO office or representative for detailed information (see back cover). Special and custom designs are invited.**

IMO® Pump Performance Characteristics

IMO® Model No.	APPLICATION					SPECIAL FEATURES		SPECIFICATIONS							No. of Sizes
	Process/ Transfer	Fuel Oil Service	Lube Oil Service	Hydraulic Sealing	Fluid Power	Integral Relief Valve	I-MAG® Magnetic Drive Coupling*	Flow Range		Maximum Discharge Pressure		Maximum Speed	Maximum Power		
								US GPM	L/MIN	PSIG	BAR-G	RPM	HP	KW	
3E	•	•	•					1-100	4-375	150	10	8000	20	15	10
3R	•	•	•					150-1100	570-4100	300	20	1800	320	240	10
110H	•				•			2-7	7-25	1500	100	5000	10	7.5	2
210H	•	•			•			7-16	25-60	1500	500	5000	30	22	4
3D	•	•	•	•	•		•	5-400	20-1500	500	35	5000	150	110	11
6D	•	•		•			•	5-400	20-1500	1500	100	5000	300	225	12
12D	•	•		•			•	5-250	20-950	2200	150	5000	350	260	10
ACE			•			•		4-44	16-176	150	10	4000	10	7.5	5
UCG	•		•			•		30-280	110-1060	250	18	4000	75	56	4
UCF	•		•			•		120-800	480-3200	175	12	2000	150	110	7
8L	•	•		•	•			250-900	950-3400	1500	100	2300	1000	745	5
12L	•			•	•			15-100	55-375	5000	345	4400	370	275	5
4U					•			15-175	55-660	1500	140	4400	300	225	7
6U					•			15-175	55-660	2500	170	4400	450	335	8
4T				•				15-175	55-660	1500	100	4400	225	170	7
6T				•				15-175	55-660	2500	170	4400	375	280	7
323F	•	•	•					400-3500	1500-13,250	300	20	1500	775	575	4
324A	•	•	•	•			•	50-900	200-3400	500	35	4200	350	260	10
T324	•	•			•			300-800	1125-3000	700	50	2300	425	315	2

***Optional**

Note: All capabilities are not available in all pumps. Consult your local IMO office or representative for detailed information (see back cover). Special and custom designs are invited.

I-MAG® Magnetic Drive Couplings

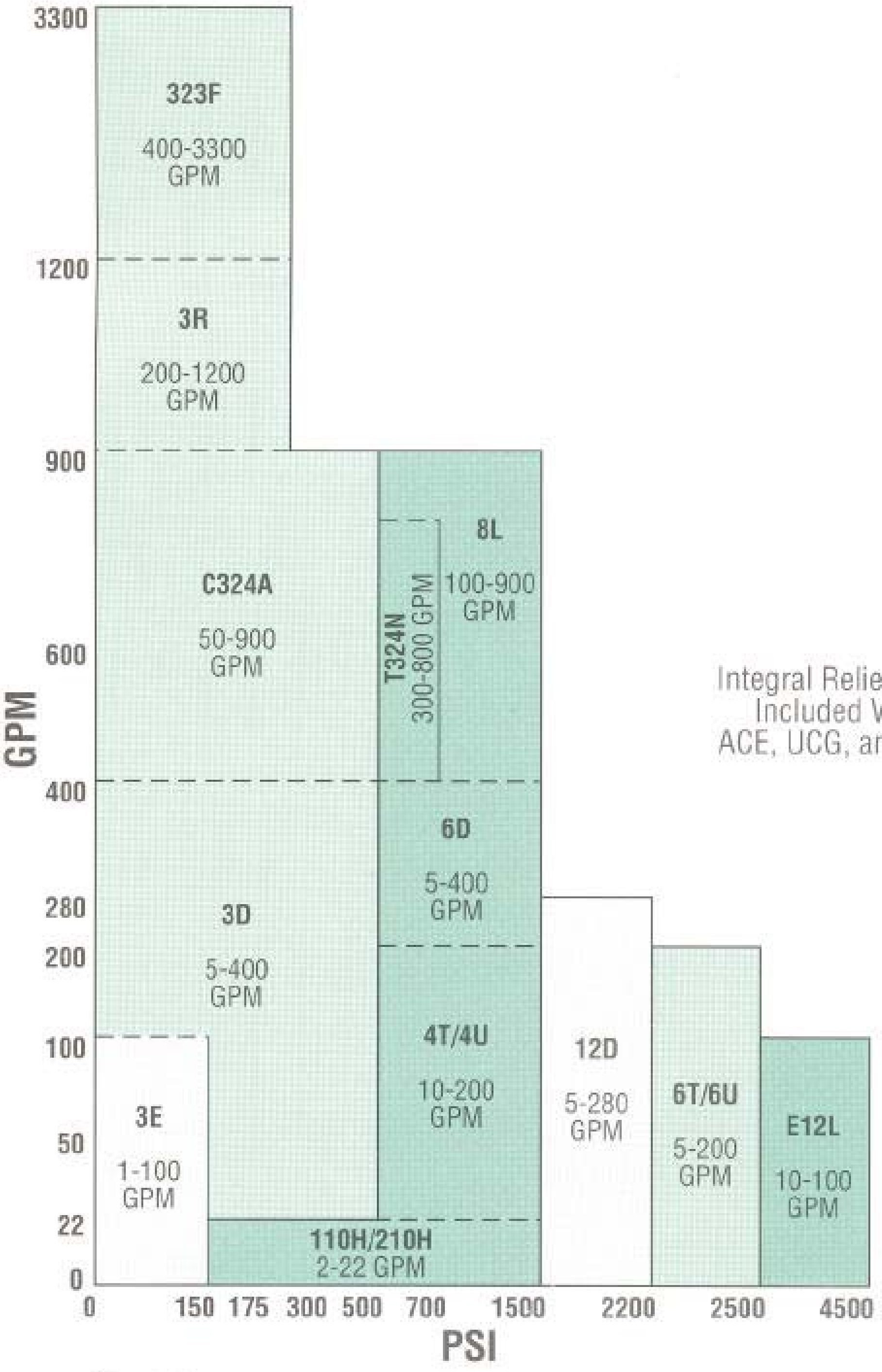
Permit leak-proof, low-maintenance pumping of non-corrosive fluids with all of the advantages of the IMO rotary screw pump design.

- Five coupling sizes for use with IMO 3D, 6D and 12D and 324 Series Pumps.
- Handle flow rates from 5 to 900 gpm at pressures to 2200 psi and speeds to 3600 rpm.
- Elimination of dynamic seals minimizes maintenance and replacement costs associated with shaft seals.
- Hydraulically balanced elements eliminate the need for thrust plates or additional bearings.
- Available in either C-face or frame mounted configurations. Optional bearing carrier permits long-coupling.
- Proven low noise design.

Model MANBC03DVC25OSC with 25 hp magnetic drive and TEFC motor, 100 gpm, 210 psig discharge, 190-2400 SSU, 1800 rpm. Vertically face mounted.

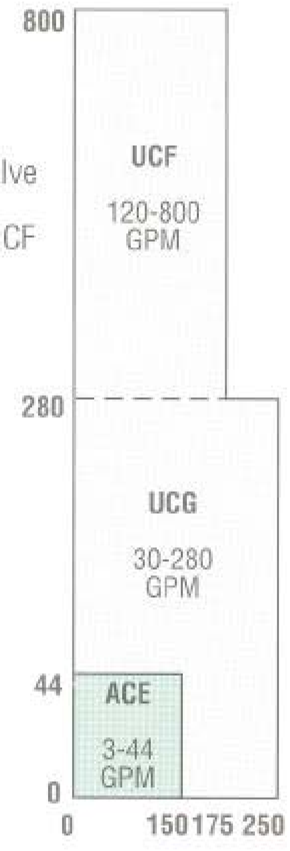
Model A3D with 15 hp magnetic drive and TEFC motor, 15 gpm, 300 psig discharge, 9600 SSU, 1200 rpm. Horizontal foot mounted.

IMO® Three-Screw Rotary Pump Quick Selection Chart



Graph 1

Integral Relief Valve Included With ACE, UCG, and UCF



Graph 2

The reliable pump people

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