Section (1)

Pumping Liquids.

(a) "Data relative to reasonable pumping velocity."

Applicable to Fuel Oil.

Contributed by R. F. Kneale.

In ordinary pumping the pipe velocity of a fluid rarely exceeds 15 feet per second. It is wise to keep under 9 feet per second. The power required to pump oil is relative to the volume handled and pressure, irrespective of the viscosity, as the adjustment necessary for this should be made in the size of pipe.

It is important, when deciding on a permissible velocity, due to quantity, rate and size of piping, that the characteristics of flow be observed; these being streamline or viscous (at low velocities), unstable, and turbulent (at the higher velocities).

Viscous flow in conjunction with pipes of small to medium diameters results in the least loss of head.

Turbulent flow renders greater loss of head.

It is important that any velocity decided upon as an average should manifest a characteristic flow, either definitely "streamline" or "turbulent," when definite law will prevail in regard to pumping effort expended. The critical unstable velocity should be avoided, which is indicated by a sudden change in the definite characteristic flows; it is during this period that friction head increases very rapidly. Variation of temperature can create this stage suddenly if general conditions and nature of fluid being pumped are not observed.

In pumping oil long distances it is considered within a good economic range to pump at a velocity of 2.5 to 3.5 feet per
second, after all factors of operating expenses are considered; as applicable to permanent land lines, etc. This low velocity is obtained by a larger pipe. Undoubtedly the larger pipe line effects the greatest economy generally.

For comparison, and considering 8 in. and 10 in. dia. pipes for a given quantity, the most economic velocity should be determined as falling between 2.5 and 3.5 feet per second, and the pipe chosen with its computed necessary velocity nearest to this economic margin.

It is imperative that some investigation must be made and reliable formula applied in order to determine the characteristic flow and the choice of pipe diameter made accordingly to produce a stable flow.

The following is deduced from an article published in the September, 1936, "Oil and Gas Journal" relative to usual pipe diameters, capacities and initial pressures, and will give comparison indicating the extent of deviation from the economic range already mentioned:

<table>
<thead>
<tr>
<th>Maximum Delivery (U.S.A. bbls./day)</th>
<th>Size of Pipe (Imp. Gals. per Hour)</th>
<th>Initial Working Pressure (Tons per Hour (2240 lbs. per hour))</th>
<th>Velocity per Second</th>
<th>Approximate Distance for Pumping</th>
</tr>
</thead>
<tbody>
<tr>
<td>12,000</td>
<td>73.85</td>
<td>1000</td>
<td>4 ft.</td>
<td>33 miles</td>
</tr>
<tr>
<td>25,000</td>
<td>154</td>
<td>900</td>
<td>4.65 ft.</td>
<td>33 miles</td>
</tr>
<tr>
<td>40,000</td>
<td>246</td>
<td>800</td>
<td>4.75 ft.</td>
<td>35 miles</td>
</tr>
</tbody>
</table>

Pertaining to pumping from tankers with distances shorter than in table above, the velocity according to circumstances and conditions prevailing could possibly be further increased slightly.

Furnace Oil—Specific Gravity .945.

From Actual Tests Viscosity = 252 Secs. Red. No. 1 at 60° Fah.
Viscosity = 137 Secs. Red. No. 1 at 80° Fah.
Viscosity = 95 Secs. Red. No. 1 at 100° Fah.

Using 8 in. dia. pipe lines, and based on a temperature of 60° Fah.—

250 tons per hour = Velocity of 7.5 ft. per second.
260 tons per hour = Velocity of approx. 8 ft. per second.
300 tons per hour = Velocity of approx. 9 ft. per second.

Using 10 in. dia. pipe lines (considered the standard at Singapore prior to Japanese invasion, where 300 and 400 tons per hour were pumped at 70° to 72° Fah.)—

Velocity at 300 tons per hour = Approx. 5.75 ft. per second.
Velocity at 400 tons per hour = Approx. 7.75 ft. per second.

Furnace Oil—Specific Gravity .945.

From Actual Tests Viscosity = 252 Secs. Red. No. 1 at 60° Fah.
Viscosity = 137 Secs. Red. No. 1 at 80° Fah.
Viscosity = 95 Secs. Red. No. 1 at 100° Fah.
Turbulent flow is known to occur when the value expressed as—
\[
\frac{D \times V \times W}{N}
\]
exceeds 2500

Where

\(D\) = Diameter of pipe in feet.

\(V\) = Velocity in feet per second.

\(W\) = Density in lbs. per cubic foot.

\(N\) = Absolute viscosity in foot pound seconds units.

Absolute viscosity for times exceeding 200 seconds Redwood No. 1—

In English Units—“Foot pound seconds”

\[= \text{Time in Secs. Red. No. 1 x Specific Gravity}\]

\[
\frac{5720}{252 \text{ Secs. x .945}} = .0416 \text{ absolute Viscosity}
\]

Then at 60° Fah.

\[
\frac{5720}{\text{Approx. 220 Secs. x .945}} = .0360 \text{ absolute Viscosity}
\]

and at 70° to 72° Fah.

For more detailed explanation refer Victorian Institute Proceedings, May, 1939.

The following are resultant velocities of fluid due to possible high speed pumping discharging at various existing oil discharging centres:

- 8 in. dia. pipe 250 tons per hour = 7.5 ft. per second.
- 8 in. dia. pipe 260 tons per hour = 8 ft. per second.
- 8 in. dia. pipe 300 tons per hour = 9 ft. per second.
- 10 in. dia. pipe 300 tons per hour = 6 ft. per second.
- 10 in. dia. pipe 400 tons per hour = 8 ft. per second.
- 8 in. dia. pipe 200 tons per hour = 6 ft. per second.

Considering 260 tons per hour through the 8 in. dia. pipe as a fair comparison with the maximum of 400 tons through the 10 in. dia. pipe, 8 ft. velocity per sec. seems a fair basis for an absolute maximum, under most favourable conditions.

Example.—Using formula for characteristic flow in the 8 in. pipe discharging 260 tons per hour—

\[
DVW = \frac{.66 \times 8 \times 59.06}{.0416} = 7500
\]

Therefore turbulent flow.
The pumping power required at different centres varies considerably owing to the contour of the land and inclusion of many deviations; static head likewise is in some cases very pronounced. The variation of discharge per hour is due not only to length of pipe line, but the accumulated static and friction head, relative to pumping power available. The velocities of pumping at centres are sometimes not within maximum permissible excess of economic limits due to pipe diameter, if exceeding 6 ft. per second.

Example.—Using 8 in. dia. pipe line, 200 tons per hour; i.e., 6 ft. per sec. velocity should be considered the maximum while retaining any reasonable approach to the economic range.

The rate of loading or discharge of an oil tanker varies at different installations, but the usual time required for each purpose seldom exceeds 48 hours for a full cargo of between 9000 and 12,000 tons.

Viz.—The maximum = 250 tons per hour if considered continuous for the 48 hours.

10 inch pipe line most suitable.
(b) PUMPING THROUGH PIPE LINES.
(AIR VESSELS)

As Installed on Average Service Lines.

Data Collected by R. Kneale.

Pertaining to shock pressure, it is recommended that air chambers be installed in locations most suitable. To be effective, the chamber must be located near the source of the pressure wave. It must also contain an adequate volume of air, which is estimated by computations (refer Proceedings Victorian Institute Engineers, page 33, May, 1939), or arbitrarily as a result of experiments and research generally. According to some authorities, it is considered a source of danger if the air volume in the chamber is inadequate, as it can be a source of amplification of the pressure wave.

Experiments indicate that an air chamber of suitable size is effective in reducing shock pressures to almost negligible proportions.

Difficulty is experienced in keeping an air chamber adequately supplied with air, resulting in the above protective method against shock proving somewhat unsatisfactory. The air volume tends to decrease, due to leakage from the chamber and to solution in the liquid.

The introduction of any kind of air supply from a compressed supply has the element of danger of the possibility of the air pressure falling below the liquid pressure, and then, through leakages, the liquid would force its way through the air chamber into the compressed air line.

It is possible to supply the air chamber with air from an individual compressor under automatic control. This would involve an amount of extra expense. Where the pumping system is one that entrains enough air in the liquid to ensure an adequate air volume in the chamber at all times, an air chamber can be very satisfactory.

It is recognised to a certain degree that under most conditions, in the "chemical industry," for instance, an air chamber does not provide a satisfactory solution to the problem of shock.

An alternative solution is sometimes resorted to in the way of surge relief valves. Where fluids of considerable value are being pumped, this method has a disadvantage, as provision must be made for collecting the relieved liquid and returning it to the main system.

Shock pressure can be controlled by proper manipulation of slow-closing valves, etc.
AIR VESSELS.

To determine air vessel capacity, using chart:

1. Determine air vessel operating head.
2. Static head = operating head + friction head.

Friction head is determined by reference to any available table or deduced from formula.

Inertia head allows 15 to 30 ft. per 100 ft. of pipe.

The size of pipe is not taken into consideration in calculating inertia head.

D refer to chart & join by means of a straight line the value of the operating head & the pipeline diameter. Extend the line to intersect the scale denoting the capacity of air vessel required. Where the static head is greater than 100 ft. multiply the capacity of air vessel obtained by the factor 100.

Note: The inertia head necessary to overcome inertia depends only to air vessel calculations & does not enter into calculations of the total head against which the pump must operate.

Chart by A.B. Penson.
Air chambers can function usefully along the following lines:—

(a) Absorb shock in the line, due to kinetic head, relative to velocity.

(b) Take up expansion in the line.

(c) Cushion the impulses of reciprocating pumps, etc.

The location of the air chamber in the line will affect its functioning relative to a, b, or c accordingly; sometimes more than one chamber is an advantage.

Where long suction lines are employed, coupled with a fair lift, a suction air vessel should be fitted, because the hammer action due to the weight of water in such length is very considerable.

An air vessel promotes a steadiness in flow and generally cushions shocks or suddenly applied loads.

In the "Discussion" recorded in "The Proceedings of the Institute of Mechanical Engineers," Vol. 136, June to Nov., 1937," Dr. Charles Jaeger (Switzerland) referred to air chambers connected to a pipe line at a particular point as having been little studied, etc.

Various authors took for granted what was by no means certain, namely, that the air, water vapour, and water remained very distinct, also they took no account of the fact that the volume of air absorbed by the water might vary very rapidly, and they ignored variations in the volume of water vapour. No one had yet studied the case of an air bubble freed in an inclined pressure pipe, nor the case where a too violent negative hammer cause a vacuum in the pipe.

The paper referred to is one given by Professor Robert W. Angus, B.A.Sc., M.E., M.I.Mech.E.

Water Hammer is always more dangerous with pipes functioning under low heads.

From D. A. Low's "Pocket Book for Mechanical Engineers," we read:—

Air Vessels—The delivery of water is more uniform and the shocks at the beginning of a stroke are reduced by placing an air vessel near to the delivery valve. The volume of the air vessel varies greatly in practice. The volume of the air vessel may be from two to six times the capacity of the pump barrel; sometimes it is as much as ten times.

An air vessel on the suction pipe near the suction valve is also desirable when the suction pipe is long. The capacity of this vessel may be from two to four times the capacity of the pump barrel. The air in this vessel has, of course, a pressure less than that of the atmosphere.
Lionel S. Marks says:—

*Air Chamber*—To take up irregularities and induce a uniform flow in the discharge pipe, an air chamber is generally used. This is a necessity in single pumps and also in crank and flywheel pumps of any type, but may be dispensed with in duplex direct-acting low-service pumps (75 lbs. pressure).

In duplex pumps its volume should be four times the displacement of one plunger per stroke; in the other types eight times.

**Arrangement of Pumps.**

1. Single single-acting pumps are seldom used; they cannot run *without large air chambers*. Exception: Deep well pumps.
2. Single double-acting pumps, either of the direct-acting or crank-and-flywheel type, are used extensively for boiler feeding, elevator service and in waterworks. They must be equipped with *large air chambers*, and special attention must be paid to keeping the water free from air.
3. Duplex double-acting pumps, either direct-acting or with crank and flywheel, deliver a very steady flow of water, and are used for all services. *Only small air chambers required.*
4. Triplex single-acting pumps deliver an almost uniform flow of water. This arrangement is used in power pumps and triple-expansion crank-and-flywheel pumps.
5. Triplex double-acting pumps. The triplex double-acting arrangement is preferred for horizontal pumps because the load on the frame and shaft is only one half of that of a single-acting pump.

A general formula known to be used in practice with good results is as follows:—

\[ C = 0.0015D^2H \]

Where \( C \) = total capacity of air chamber in cub. ft.
\( D \) = diameter of pipe line in inches.
\( H \) = operating head = static head plus friction head plus head necessary to overcome inertia.

Where static head is greater than 100 ft. multiply

\[ \frac{C \times \text{static head}}{100} \]
Mr. W. H. Cumming asked what were the main points to consider when contemplating the installation of a pipe line for pumping large quantities of fuel oil or allied products.

Mr. R. F. Kneale replied that the problem involved consideration of:

(a) Capacity of the line and its behaviour over a number of years, and its depreciation. This involved consideration of the choice of material.

(b) Selection of a proper and economic size of line.

(c) Characteristics of the product to be pumped.

(d) Type of pumps and cost of fuel or power for operating them.

(e) Maintenance.

All these items influenced the annual cost. For example, item (e), seasonal temperature variations of flowing oil seriously affected the viscosity, causing high pumping pressures in the winter, or alternatively lowered the capacity.
Library Digitised Collections

Author/s:
Kneale, R. F.

Title:
Pumping liquids (Data for discussions: Section (1))

Date:
1943

Persistent Link:
http://hdl.handle.net/11343/24885

File Description:
Pumping liquids (Data for discussions: Section (1))