the position shown in the right-hand diagram, and the water is imprisoned between the lobe and the case. An instant later the two lobes are forcing this water up the pipe E, while the water coming in at D is flowing to the left. The greatest objection to

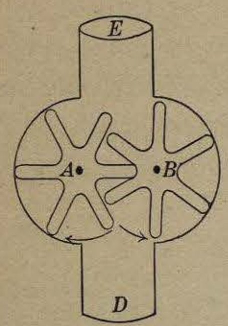

Fig. 199*a*.

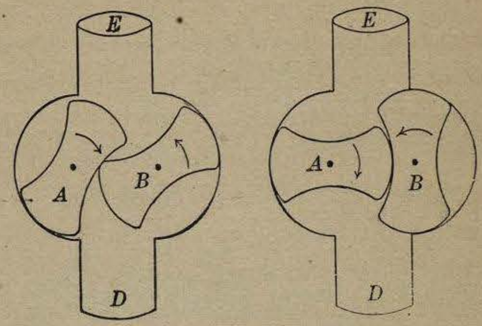

Fig. 199*b*.

these pumps is that it is difficult to maintain close contact between the case and the lobes or wheels, owing to wear, so that after being in use for some time there is much back leakage of water, and the capacity and efficiency of the pump are diminished. The only apparent advantage of the rotary pump is that it has no valves. Five rotary pumps of the type of Fig. 199*b* were installed in 1902 at a pumping station near Chicago, the lobes or impellers being 4 feet long and the distance between their centers 2.7 feet; these pumps run at 100 revolutions per minute, and each has a capacity of 6000 cubic feet per minute under the total lift of about 8 feet.*

The pumps thus far described, with the exception of the hydraulic ram, may be called mechanical pumps, because they act under energy communicated to them from motors. All mechanical pumps are reversible; that is, when the water moves in the opposite direction under a pressure-head, they become hydraulic motors. The reverse of the chain and bucket pump is the overshot or breast wheel, that of the suction and lift pump is the water-pressure engine, and that of the centrifugal pump is the turbine. The hydraulic ram does not operate under the action of a motor, and it does not appear to be reversible.

* Engineering News, 1903, vol. 49, p. 172.

Pumps which have no moving parts and which operate through the action of air suction and dynamic pressure constitute another class which will now be briefly considered. Here belong the jet or ejector pumps which act largely through suction, and the injector pump used on locomotives. The latter produces a vacuum through the flow of steam, and cannot be discussed here, as it involves principles of thermodynamics. The fundamental principle, however, is indicated in Fig. 199*c*, which shows the jet apparatus invented by James Thomson in 1850.* The water to be lifted is at C, and it rises by suction to the chamber B, from which it passes through the discharge pipe to the tank D. The forces of suction and pressure are produced by a jet of water issuing from a nozzle at the mouth of the discharge pipe, the nozzle being at the end of a pipe AB through which water is brought from a reservoir; or the water delivered from the nozzle may come from a hydrant or from a force pump.

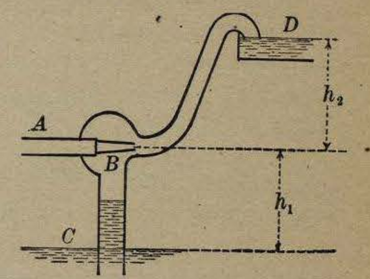

Fig. 199*c*.

Let H be the effective head of the jet as it issues from the nozzle, h_1 the suction lift, and h_2 the lift above the tip of the nozzle; let q be the discharge through the nozzle and q_1 that through the suction pipe. Then, neglecting frictional resistances,

$$qH = qh_2 + q_1(h_1 + h_2)$$
$$e = (qh_2 + q_1h_1 + q_1h_2)/qH$$

It is found by experiments that the efficiency of this jet pump is very low, usually not exceeding 20 percent, the highest efficiencies being for low ratios of $h_1 + h_2$ to H. This form of pump has, however, been found very convenient in keeping coffer dams and sewer trenches free from water, as it requires little or no attention and has no moving parts to get out of order.

Another class of pumps uses the pressure of air or of steam in order to elevate water. The idea of these pumps is old, yet it was not until 1875 that the steam pulsometer was perfected by Hall, while

* Report of British Association, 1852, p. 130.

the air-lift pump of Frizell dates from 1880. The air-lift pump is now extensively used for raising water from deep wells, the compressed air being forced down a vertical pipe in the well tube and issuing from its lower end. As it issues, bubbles are formed in the entire column of water in the well tube, and being lighter than a column of common water, it rises to a greater height under the atmospheric pressure, assisted by the upward impulse of the bubbles to a slight extent. In this manner water having a natural level 50 feet or more below the surface of the ground may be caused to rise above that surface. It has been found in practice that for lifts of 15 to 50 feet from 2 to 3 cubic feet of air are necessary for each cubic foot of water that is elevated. The efficiency of this form of pump is low, rarely reaching 30 percent, although a maximum of 50 percent has been claimed.*

Among the many forms of pumps operating under the pressure of compressed air only the ejector pump used in the Shone system of sewerage can here be mentioned. The sewage from a number of houses flows to a closed basin, called an injector, in which it continues to accumulate until a valve is opened by a float. The opening of this valve allows compressed air to enter, and this drives out the sewage through a discharge pipe to the place where it is desired to deliver it. In the installation of this system of sewerage at the World's Fair of 1893 in Chicago, there were 26 ejectors which lifted the sewage 67 feet, the total pressure-head being about 108 feet. Vacuum methods of moving sewage have also been used in Europe, but these cannot compete in efficiency with those using compressed air.

Prob. 199. For Fig. 199*c* let the diameter of the nozzle be 1 inch and that of the discharge pipe 4 inches. Let H be 64 feet, h_1 be 18 feet, h_2 be 3 feet, and the discharge from the nozzle be 0.25 cubic feet per second. Compute the greatest quantity of water that can be lifted per second through the suction pipe, and the efficiency of the apparatus when doing this work.

Art. 200. Pumping through Pipes

When water is pumped through a pipe from a lower to a higher level, the power of the pump must be sufficient not only to raise the required amount in a given time, but also to overcome the various resistances to flow. The head due to the resistances is

* Journal of Association of Engineering Societies, 1900, vol. 25, p. 173.

thus a direct source of loss, and it is desirable that the pipe should be so arranged as to render this as small as possible. The length of the pipe is usually much greater than the vertical lift, so that the losses of head in friction are materially higher than those indicated by the discussion of Art. 195, where vertical discharge pipes were alone considered.

Let w be the weight of a cubic foot of water and q the quantity raised per second through the height h, which, for example, may be the difference in level between a canal C and a reservoir R, as in Fig. 200*a*. The useful work done by the pump in each second is wqh. Let h' be the head lost in entering the pipe at the canal, h'' that lost in friction in the pipe, and h''' all other losses of head, such as those caused by curves, valves, and by resistances in passing through the pump cylinders. Then the total work performed by the pump per second is

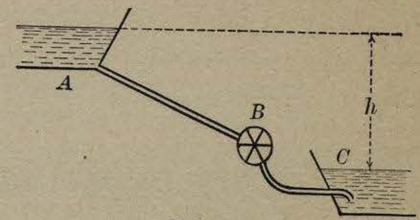

Fig. 200*a*.

$$k = wqh + wq(h' + h'' + h''') \qquad (200)_1$$

Inserting the values of the lost heads from Arts. 89-92, this expression takes the form

$$k = wqh + wq\left(m + f\frac{l}{d} + m_2\right)\frac{v^2}{2g} \qquad (200)_2$$

in which v is the velocity in the pipe, l its length, and d its diameter. In order, therefore, that the losses of work may be as small as possible, the velocity of flow through the pipe should be low; and this is to be effected by making the diameter of the pipe large. The size of the pipe is here regarded as uniform from the canal to the reservoir; in practice the suction pipe is usually larger in diameter than the discharge pipe, in order that the suction valves may receive an ample supply of water.

For example, let it be required to determine the horse-power of a pump to raise 1 200 000 gallons per day through a height of

230 feet when the diameter of the pipe is 6 inches and its length 1400 feet. The discharge per second is

$$q = \frac{1\,200\,000}{7.481 \times 24 \times 3600} = 1.86 \text{ cubic feet,}$$

and the velocity in the pipe is

$$v = \frac{1.86}{0.7854 \times 0.5^2} = 9.47 \text{ feet per second.}$$

The probable head lost in entering the pipe is, by Art. 89,

$$h' = 0.5\frac{v^2}{2g} = 0.5 \times 1.39 = 0.7 \text{ feet.}$$

When the pipe is new and clean, the friction factor f is about 0.020, as shown by Table 90*a*; then the loss of head in friction in the pipe is, by Art. 90,

$$h'' = 0.020 \times \frac{1400}{0.5} \times 1.39 = 77.8 \text{ feet.}$$

The other losses of head depend upon the details of the pump cylinder and the valves; if these be such that $m_2 = 4$, then

$$h''' = 4 \times 1.39 = 5.6 \text{ feet.}$$

The total losses of head hence are

$$h' + h'' + h''' = 84.1 \text{ feet.}$$

The work to be performed per second by the pump now is

$$k = 62.5 \times 1.86(230 + 84.1) = 36\,510 \text{ foot-pounds,}$$

and the horse-power to be expended is 36 510/550 = 66.4. If there were no losses in friction and other resistances, the work to be done would be simply

$$k = 62.5 \times 1.86 \times 230 = 26\,740 \text{ foot-pounds,}$$

and the corresponding horse-power would be 26 740/550 = 48.6. Hence 17.8 horse-power is wasted in injurious resistances, or the efficiency of the plant is only 73 percent.

For the same data let the 6-inch pipe be replaced by one 14 inches in diameter. Then, proceeding as before, the velocity of flow is found to be 1.74 feet per second, the head lost at entrance

0.03 feet, the head lost in friction 1.13 feet, and that lost in other ways 0.19 feet. The total losses of head are thus only 1.35 feet, as against 84.1 feet for the smaller pipe, and the horse-power required is 48.9, which is but little greater than the theoretic power. The great advantage of the larger pipe is thus apparent, and by increasing its size to 18 inches the losses of head may be reduced so low as to be scarcely appreciable in comparison with the useful head of 230 feet.

A pump is often used to force water directly through the mains of a water-supply system under a designated pressure. The work of the pump in this case consists of that required to maintain the pressure and that required to overcome the frictional resistances. Let h_1 be the pressure-head to be maintained at the end of the main, and z the height of the main above the level of the river from which the water is pumped; then $h_1 + z$ is the head H, which corresponds to the useful work of the pump, and, as before,

$$k = wqH + wq(h' + h'' + h''')$$

To reduce the injurious heads to the smallest limits the mains should be large in order that the velocity of flow may be small. In Fig. 200*b* is shown a symbolic representation of the case of pumping into a main, P being the pump, C the source of supply, and DM the pressure-head which is maintained upon the end of the pipe during the flow. At the pump the pressure-head is AP; so that AD represents the hydraulic gradient for the pipe from P to M. The total work of the pump may then be regarded as expended in lifting the water from C to A, and this consists of three parts corresponding to the heads CM or z, MD or h_1, and AB or $h' + h'' + h'''$, the first overcoming the force of gravity, the second maintaining the discharge under the required pressure, while the last is transformed into heat in overcoming friction and other resistances. In this direct method of water supply a standpipe, AP, is often erected near the pump, in which the water rises to a height corresponding to the required pressure, and which furnishes a supply when a temporary stoppage of the pumping engine

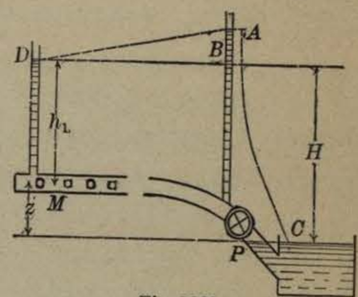

Fig. 200*b*.

occurs. This standpipe also relieves the pump to some extent from the shock of water hammer (Art. 157).

Prob. 200. Compute the horse-power of a pump for the following data, neglecting all resistances except those due to pipe friction: $q=1.5$ cubic feet per second, which is distributed uniformly over a length $l_1=3000$ feet (Art. 104), the remaining length of the pipe being 4290 feet; $d=10$ inches, $h_1=75.8$ feet, and $z=10.6$ feet.

Art. 201. Pumping through Hose

In Art. 109 the flow of water through fire hose was briefly treated and the friction factors given for different kinds of hose linings. It was shown that the loss of head in a long hose line becomes so great, even under moderate velocities, as to consume a large proportion of the pressure exerted by the hydrant or steamer. As another example, let the pressure in the pump of the fire engine be 122 pounds per square inch, corresponding to a head of 281 feet, and let it be required to find the pressure-head in $2\frac{1}{2}$-inch rough rubber-lined cotton hose at 1000 feet distance, when a nozzle is used which discharges 153 gallons per minute, the hose being laid horizontal. The discharge is 0.341 cubic feet per second, which gives a velocity of 10.0 feet per second in the hose. Hence by (90) the loss of head in friction is 231 feet, so that the pressure-head at the nozzle entrance is only 50 feet, which corresponds to about 22 pounds per square inch. The remedy for this great reduction of pressure is to employ a smaller nozzle, thus decreasing the discharge and the velocity in the hose; but if both head and discharge are desired, they may be obtained either by an increase of pressure at the steamer or by the use of a larger hose.

Another method of securing both high velocity-head and quantity of water is by the use of siamesed hose lines, and this is generally used when large fires occur. This method consists in having several lines of hose, generally four, lead from the steamer to a so-called siamese connection, from which a short single line of hose leads to the nozzle. In Fig. 201 the pump or fire steamer is represented by A, the siamese joint by B, the nozzle entrance by C, and the nozzle tip by D. From A let n

lines of hose, each having the length l_1 and the diameter d_1, lead to B; and from B let there be a single line of length l_2 and diameter d_2 leading to the nozzle which has the diameter D. The hydraulic gradient (Art. 99) is shown by $abcD$, the pressure-heads

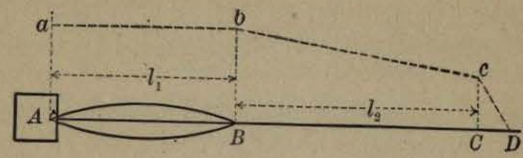

Fig. 201.

at A, B, C being represented by Aa, Bb, Cc. Let h be the pressure-head on the nozzle tip or the difference of the elevations of the points a and D. It is required to deduce a formula for the velocity at the nozzle tip and to determine the pressure-heads at B and C.

This case is one of diversions, already treated in Art. 105, and the same principles may be applied to its solution. Neglecting losses in entrance, in curvature, and in the siamese joint, the total head h is expended in friction in the hose lines and in the nozzle, or

$$h=f_1\frac{l_1}{d_1}\frac{v_1^2}{2g}+f_2\frac{l_2}{d_2}\frac{v_2^2}{2g}+\frac{1}{c_1^2}\frac{V^2}{2g}$$

in which v_1 and v_2 are the velocities in the lines l_1 and l_2, and V is that from the nozzle, while c_1 is the coefficient of velocity of the nozzle (Art. 83). The first term of the second member is the head lost between A and B, and the algebraic expression for this is independent of the number of hose lines between those points; the velocity v_1 in these hose lines depends, indeed, upon their number, but the hydraulic gradient ab is the same for each and all of them. The law of continuity of flow (Art. 31) gives, however,

$$nd_1^2v_1=d_2^2v_2=D^2V$$

and, taking from these the values of v_1 and v_2 in terms of V and inserting them in the expression for h, there results

$$V^2=\frac{2gh}{\frac{f_1l_1}{n^2d_1}\left(\frac{D}{d_1}\right)^4+\frac{f_2l_2}{d_2}\left(\frac{D}{d_2}\right)^4+\frac{1}{c_1^2}} \qquad (201)$$

from which the velocity V and the velocity-head $V^2/2g$ can be computed, while the discharge is given by $q = \frac{1}{4}\pi D^2 V$. The pressure-head h_2 at the nozzle entrance and the pressure-head h_1 at the siamese joint may then be found from

$$h_2 = \frac{1}{c_1^2}\frac{V^2}{2g} \qquad h_1 = \left[\frac{fl_2}{d_2}\left(\frac{D}{d_2}\right)^4 + \frac{1}{c_1^2}\right]\frac{V^2}{2g}$$

and, as a check, the latter should equal h minus the drop of the hydraulic gradient between a and b.

This discussion shows that, by increasing the number n, the loss of head between A and B may be made very small, the effect being practically the same as that of moving the steamer to B and using but a single hose line l_2. As a numerical example, let $h = 230.4$ feet, $l_1 = 500$ feet, $l_2 = 60$ feet, $d_1 = d_2 = 2.5$ inches, $D = 1$ inch, and $c_1 = 0.975$. Then, taking f as 0.03, the computed results for different values of n are as follows, V being in feet per second, $V^2/2g$ in feet, and q in gallons per minute. It is seen that

$n =$	1	2	3	4	6	∞
$V =$	68.9	92.2	99.8	10.3	105	107
$V^2/2g =$	73.7	132	155	165	173	180
$q =$	169	226	244	252	258	263

for four lines the velocity-head is more than double that for a single line and that the discharge is 50 percent greater. With more than four lines the velocity-head and discharge increase slowly, and for $n = \infty$ they are practically the same as for $n = 10$. The number of hose lines generally used is four, since the slight advantage of more lines is not sufficient to warrant their use.

Many other interesting problems relating to hose lines may be solved by using the same principles. If there are four lines of hose between the pump and the siamese joint, three having the diameter d_1 and one having the diameter d, it can be shown that the formula (201) applies, provided n be replaced by $3 + (d/d_1)^{\frac{5}{2}}$. For instance, if d be 3 inches and d_1 be $2\frac{1}{2}$ inches, this makes n^2 about 19. In deducing this expression for n it is assumed that the friction factors are the same for both sizes of hose, although in strictness the smaller hose has the higher value of f.

Another case is where two of the hose lines between A and B have the diameter d_1 and the length l_1, while the two other lines are of the length $l + l_3$, the length l having the diameter d and the length l_3 the diameter d_3. Here the principles regarding compound pipes (Art. 100) are also to be regarded, and formula (201) applies likewise to this case, if n be computed from

$$n = 2 + 2\left(\frac{d}{d_1}\right)^2\sqrt{\frac{e_1}{e + e_3(d/d_3)^4}}$$

in which e represents $f(l/d)$, while e_1 and e_3 represent $f_1(l_1/d_1)$ and $f_3(l_3/d_3)$ respectively. For instance, if $l_1 = 100$, $l_3 = 100$, and $l = 50$ feet, while $d_1 = d_3 = 2\frac{1}{2}$ inches and $d = 3$ inches, then the value of n^2 is about 21, so that this arrangement is more effective than that of the preceding paragraph.

In the deduction of the above formulas losses of head at entrance and in the siamese joint have not been regarded, and it is unnecessary to consider these when the hose lines are long. For lines less than 100 feet in length the losses of head at entrance may be taken into account by adding the term $0.5(D/d_1)^2/n^2$ to the denominator of (201). The loss of head due to the siamese joint may, in the absence of experimental data, be approximately accounted for by adding about 0.02 to that denominator, thus considering its influence about one-half that of the nozzle. In a case like that of the last paragraph, where the length l in two of the hose lines is nearest the pumps, the values of e and e_1 may be increased by 0.5 in order to introduce the influence of the entrance heads. Errors of 5 percent or more are liable to occur in computations on pumping through short hose lines.

Prob. 201*a*. Three hose lines run from a pump to a siamese connection, each being 500 feet long and $2\frac{1}{2}$ inches in diameter, and from the siamese one line 50 feet long and $2\frac{1}{2}$ inches in diameter leads to a $1\frac{1}{2}$-inch nozzle having a velocity coefficient of 0.96. When the pressure at the pump is 100 pounds per square inch, what is the discharge from the nozzle and the velocity-head of the jet? What friction heads are lost in the hose and nozzle?

Prob. 201*b*. In a fire-engine test made in 1903, the lengths l_1 and l_2 were 50 feet, the length l was 12 feet, and l_2 was zero, as the nozzle was attached directly to the siamese joint. The diameter d_1 was 3 inches, while d and d_3 were $2\frac{1}{2}$ inches, and D was 2 inches. The pressure gage on the steamer read 90, while one on the siamese joint read 63 pounds per square inch. Compute the pressure-head at the siamese joint.

Prob. 201*c*. What is the efficiency of a bucket pump which lifts 2000 liters of water per minute through a height of 3.5 meters with an expenditure of 2.5 metric horse-powers?

Prob. 201*d*. When the height of the mercury barometer is 760 millimeters, water at a temperature of 0° centigrade is raised by suction in a perfect vacuum to a height of 10.33 meters (Art. 193). Under the same atmospheric pressure, how high can it be raised when the temperature is 32° centigrade?

Prob. 201*e*. What metric horse-power is required to raise 4 000 000 liters per day through a height of 75 meters when the diameter of the pipe is 20 centimeters and its length 500 meters?

Prob. 201*f*. The calorie is the metric thermal unit, this being the energy required to raise one kilogram of water one degree centigrade when the temperature of the water is near that of maximum density. How many calories are equivalent to 1 000 000 British thermal units?

APPENDIX

Art. 202. Hydraulic-Electric Analogies

It is well known that there are certain analogies between the flow of water in pipes and that of the electric current in wires, and some of these will here be briefly explained from a hydraulic point of view. The electric analog of a water pump is the dynamo, both being driven by mechanical power and both transforming it into other forms of energy. The analog of a water wheel is the electric motor, each of which delivers mechanical power by virtue of the energy transmitted to it through the water pipe or electric wire. While the water is flowing from the pump to the wheel much of its energy is lost in overcoming frictional resistances, whereby heat is produced; while the electricity is flowing from the dynamo to the electric motor some of its energy is lost in overcoming molecular resistances, whereby heat is produced. The steady flow of water corresponds to the continuous flow of electricity in one direction, or to the direct current, and the following discussion compares hydraulic phenomena with those of the direct electric current. The phenomena of the alternating current have also certain hydraulic analogies in the flow of water, but these will not be discussed here.

Let q represent electric current, R the electric resistance of a wire of length l, cross-section a, and diameter d, and p the electromotive force under which the current is pushed through the wire. Then Ohm's law gives, if s is the specific resistance of the material of the wire,

$$p = Rq = s\frac{l}{a}q = A\,\frac{l}{d^2}q \qquad (202)_1$$

in which A is a constant depending only on the material of the wire. This equation shows that the electric pressure p varies