ANNEX B

Pumps and Pumping

1 Introduction

The centrifugal pump is now used for most applications and systems on ships. In the machinery space it provides a much more reliable service than the steam reciprocating pumps that were still being installed in the “fifties” as auxiliary boiler feed and fuel pumps for example. These reciprocating pumps required regular maintenance and, if neglected, they needed constant attention to keep them functioning. The general use of centrifugal pump helped to make the unmanned machinery space viable.

2 General pumping system characteristics

A pump divides its pipe system into two distinct parts, each with different characteristics. These are the suction and discharge sides. On the suction side the drop in pressure that can be produced by a pump is limited to that of an almost perfect vacuum. On the discharge side there is theoretically, no limit to the height through which a liquid can be raised.

3 Suction conditions

If a liquid to be pumped is in a tank which is open to atmosphere and it is also at a height above the pump (see Figure B.1) then the liquid will flow into the pump because of its head and due to the effect of atmospheric pressure on its surface. The pump in this case only adds to the energy of the system.

When an open tank containing the liquid to be transferred is at a lower level than the pump, the energy required to bring the liquid to the pump, is provided by the atmospheric pressure on its surface but the pump must create the drop in pressure which makes the atmospheric pressure effective.

A discussion of the relevant factors of the suction side of a pumping system must include not only the height of the liquid surface above or below the pump and the effect of the atmospheric pressure, but also the effect of a vacuum or zero pressure on the liquid surface, vapor pressure and the characteristics of the suction pipe. Liquid flow through a pipe is impeded by friction over its length, by valves or other restrictions and by changes of direction.

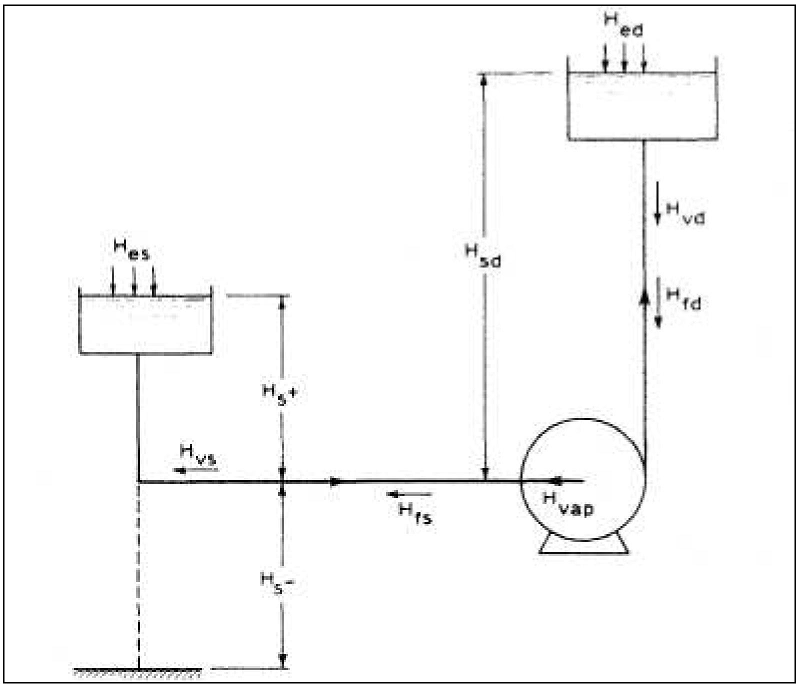

Fig. B.1 – A simple pumping system.

Figure B.1 shows pressure head H_{es} acting on the liquid surface at the suction inlet. The vertical distance of the pump centre H_s from the surface of the liquid will affect the head available at the pump and must be added algebraically to H_{es}. If the pump is below the liquid level then H_s will be positive, if it is above the liquid level H_s will be negative. The pipe will have some frictional resistance resulting in a loss of the pressure head H_{fs}. A further head loss H_v, due to the velocity of the liquid will also occur but, except for very high velocities, it is negligible.

Providing that the sum of these head losses $H_v + H_{fs} + H_s$ is less than H_{es}, the suction condition at the pump might be thought to be adequate. There are two further factors to take into consideration however. These are the vapor pressure of the liquid being pumped and the amount of remaining positive suction head required at the pump suction to effect the designed delivery rate. This factor is known as the required *NPSH* (net positive suction head).

Every liquid has a pressure at which it will vaporize and this pressure varies with temperature. If the combination of pressure and temperature within the suction pipe is such that vaporization occurs, the efficiency of the pump deteriorates and a condition can be reached where the pump will cease to function. The vapor pressure H_{vap} is thus usually shown as a suction head loss.

The summation $H_{es} + H_s - (H_{fs} + H_{vs} + H_{vap})$ is known as the available head loss (net positive suction head). In application to systems neglecting the velocity head the expression becomes:

$$NPSH_a = \frac{10.2}{\rho}\left(P_{bar} + P_{es} - P_{vap}\right) - H_{fs} \pm H_s \qquad \text{(B.1)}$$

, where:

ρ = density of liquid at maximum operating temperature, [kg/lt];

P_{bar} = barometric pressure at the pump, [bar];

P_{es} = minimum pressure on the free liquid level at the suction inlet of the pump (negative when under vacuum), [bar gauge];

P_{vap} = vapor pressure of the liquid at the maximum operating temperature, [bar abs];

H_s = height of liquid free surface above the centre line of the pump (negative when level is below the pump), [m];

H_{fs} = friction head losses in suction piping system, [m].

In application, the $NPSH_a$ must always be greater than the required *NPSH*. The former may be calculated knowing the details of the suction piping while the latter may be obtained from the pump manufacturer.

It follows that suction lift (when a pump drawing from a negative suction head) should be as small as condition allows and that for water temperatures above 75ºC the suction head must be positive or if this is impossible the suction pipe must be short, straight, free from interference and the speed of flow must be low, say less than 1 [m/s].

4 *Discharge conditions*

Some of the energy fed into the pump will be dissipated as heat due to mechanical inefficiencies. The remainder will be converted into pressure rise and fluid velocity. Some of the pressure head generated will be lost in overcoming the friction of the discharge pipe H_{fd}, some in the static head of the pipe system H_{sd}, and some in the pressure head acting on the free surface at the terminal point H_{ed}. There will also be a velocity head loss but, as in the case of suction line, for mist practical purposes this can be neglected.

5 *Pump power*

The total work done by the pump, neglecting losses within the pump itself, will be proportional to the equivalent head difference between the points of suction and discharge. This is known as total head H_{tot}:

$$H_{tot} = H_{fs} + H_{fd} + H_{vap} + H_{sd} \pm H_s \qquad \text{(B.2)}$$

The power absorbed by the pump, P_a, the becomes:

$$P_a = \frac{Q.H_{tot}\,w}{101.94\ \ _s} \qquad \text{(B.3)}$$

, where:

P_a = power absorbed, [KW];

Q = pump flow rate, [lt/s];

w = density of the liquid, [kg/lt].

The input power P_i to the pump required from the prime mover is:

$$P_i = P_a \times \frac{1}{\eta_p} \qquad \text{(B.4)}$$

, where:

η_p = pump mechanical efficiency.

For an electric driven pump, the power consumed P_c is:

$$P_c = P_a \times \frac{1}{\eta_p} \times \frac{1}{\eta_m} \qquad \text{(B.5)}$$

, where:

η_m = electric motor efficiency.

6 Friction losses

Friction losses on pipes depend on the flow rate, pipe dimensions and roughness, and the properties of the fluid.

To the head loss presented in equation (8.5) must be added the loss due to bends each equivalent to from 3 to 6 meters of straight pipe, depending upon the radius of the bend. Table B.1 is usually provided by manufacturers to find losses due to bends and fittings.

Drawings or prints are supplied with pumps by the manufacturers, giving sizes and particulars of flanges, positions of foundation bolts and other information necessary for the arrangement of pipe connections. These must be exactly adhered to; it is little use installing a highly efficient pump if the power is dissipated and increased by the use of unsuitable pipes and fittings or by poor layout.

Tab. B.1 – Loss for fittings in equivalent lengths of straight pipe [m].

Bore of pipe mm	Vel. head for ordinary pipe	Vel. head for bell-mouthed entry	Bend	Foot valve	Non-return valve	Delivery valve full open	Strainer	Tess and elbows
25	1.37	0.82	0.76	0.24	0.305	0.24	0.091	0.83
38	2.2	1.31	1.13	0.36	0.49	0.36	0.152	1.31
50	3.0	1.8	1.52	0.52	0.7	0.52	0.214	1.8
65	3.8	2.31	1.95	0.64	0.85	0.64	0.275	2.31
75	4.75	2.86	2.44	0.79	1.06	0.79	0.305	2.87
100	6.4	3.96	3.3	1.10	1.43	1.10	0.427	3.96
125	8.5	5.2	4.27	1.43	1.86	1.43	0.58	5.2
150	10.7	6.4	5.27	1.76	2.31	1.76	0.70	6.4

7 *Pipe connections*

Connecting pipes of the correct bore should be fitted and this is of special importance for the suction pipe where restriction causes cavitation. Pipe connections should be as direct as possible, sharps, bends and lops such as that shown in Figure B.2 must be avoided. The loop could increase turbulence and be the location of an air pocket.

A common fault, causing a great deal of trouble, is shown in Figure B.3. Here the suction pipe connected to the pump slopes downstream to the pump, with the result that an air pocket is formed.

A similar fault is found when reducing pipes are used to connect an oversize pipe to the pump inlet. The type of reducing pipe shown at A (figure B.4) in conjunction with a 90º bend, creates an air pocket. Such a pipe is quite satisfactory on the discharge side of a pump when used to increase the diameter of the piping. At the suction side the type of reducer shown at B, could be used.

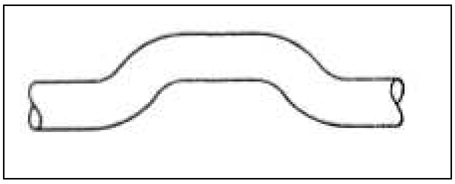

Fig. B.2 – Loop to be avoided.

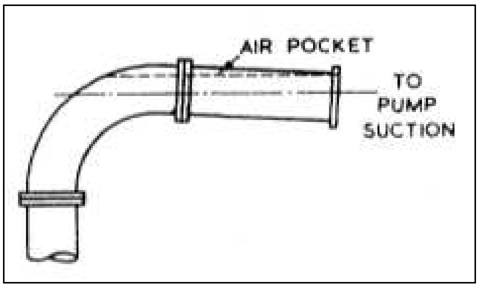

Fig. B.3 – Suction pipe sloping upwards from pump to be avoided.

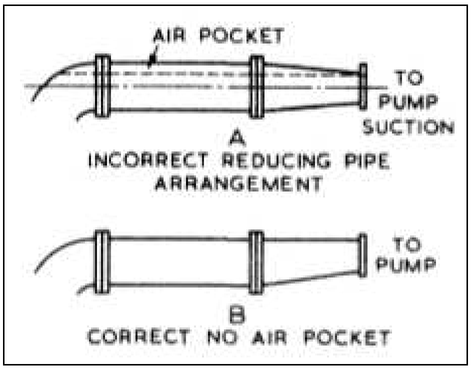

Fig. B.4 – Reducing pipes.

8 *Centrifugal pump discharge characteristics*

From a mathematical consideration of the action of a centrifugal pump it can be shown that the theoretical relationship between head, *H*, and flow rate, *Q*, is a straight line (see Figure B.5), with minimum flow rate occurring when head is maximum. Because of shock and eddy losses caused by impeller blade thickness and other mechanical considerations there will be some head loss, increasing slightly with flow rate. These losses, together with friction losses due to fluid contact with the pump casing and inlet and impact losses, result in the actual *H/Q* curve shown in Figure B.5. The shape of this curve, shows the discharge characteristic which varies according to the design and features of the particular pump. The discharge characteristic is obtained for a pump type, by measuring flow rate (*Q*) through increase of head (*H*) during a test at constant speed. The actual discharge characteristic provides important information for the designer of a pumping system; it also explains why the flow rate of a centrifugal pump alters with discharge head or back pressure. A slow rate of discharge by a centrifugal cargo pump can be explained by increasing head due to a restricted or very long discharge pipe, high viscosity of the liquid, discharge to a tank storage sited at high level or even partly open valve on the discharge line.

Depending on application, centrifugal pumps can be designed with relative flat *H/Q* curves or if required the curve can be steep to give a relative large shut-off head.

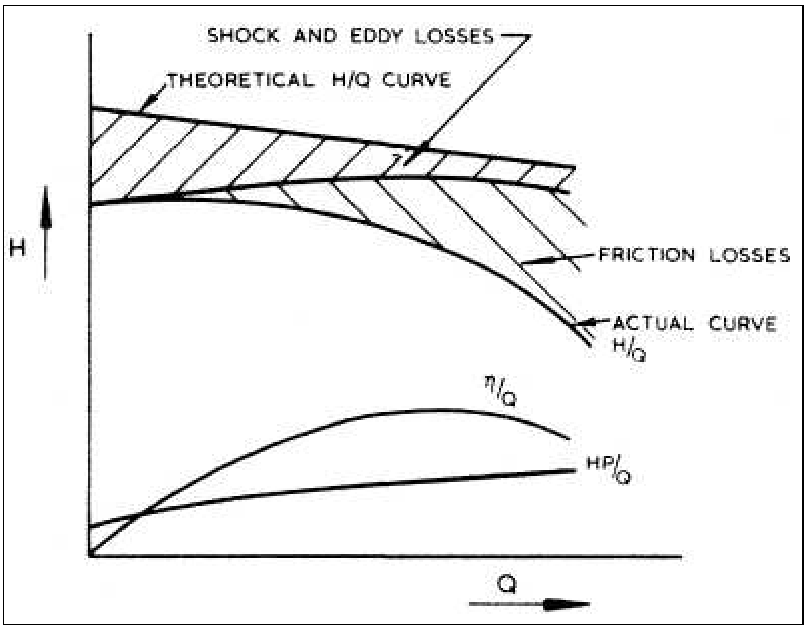

Fig. B.5 – Centrifugal pump H/Q curves.

From Figure B.5 and from *HP/Q*, the power curve, that minimum power is consumed by the pump when there is no flow and when the discharge head is at its highest. This equates to the discharge valve being closed. Because maximum pressure with the discharge closed is only moderately above working pressure, a relief valve is not necessary for a centrifugal pump.

It will be noticed that the efficiency curve for the pump is convex which mean a maximum efficiency occurs at a point somewhere between maximum and minimum discharge head and flow rate conditions.

In case of a variable speed pump:

a) Head varies as the square of the speed: $H = \frac{u_2 V_2 \cos\alpha_2}{g}$, where V_2 is the impeller outer speed at a position r_2 (usually taken at the blade tip), and u_2 is the impeller peripheral speed;

b) Capacity varies directly as the speed: $Q = 2\pi r_2 b_2 V_{r2}$, where V_{r2} is the impeller radial speed;

c) Power varies as the cube of the speed since it is a function of head times capacity: $P = QH\gamma$;

In case of a constant speed pump:

a) Head varies as the square of the diameter:

b) Capacity varies as the diameter:

c) Power varies as the cube of the diameter.

If the head in a given installation is known, then the following formula could be used to calculate the necessary speed of the pump:

$$N = \frac{95HC}{D} \tag{B.6}$$

, where:

N = impeller speed, in [rpm];

D = diameter of the impeller over blade tips, in [m];

H = total head, in [m];

C = constant, [1/min]

The value of C varies considerably with pump shape but is generally between 1.05 and 1.2; the higher value being taken for pumps working considerably beyond their normal duty, or for pumps with impellers having small tip angles.