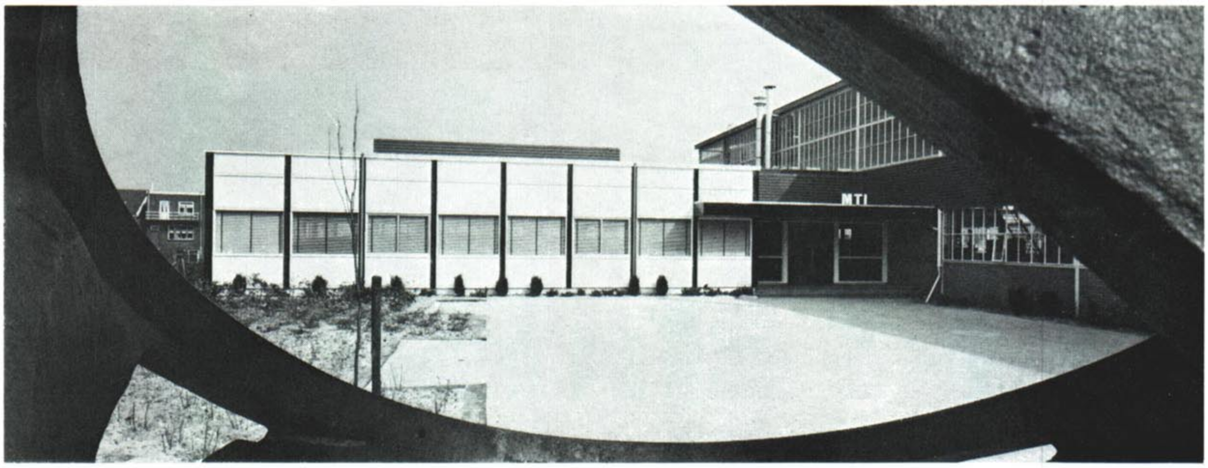
Centrifugal dredgepumps - II

By Ir. S. E. M. de Bree



This is the second in a series of articles devoted to centrifugal dredgepumps. The first appeared in issue No. 77 of *Ports and Dredging*.
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General

The previous article dealt with three technical features of the centrifugal pump as produced for dredging purposes, viz. its large bore; the simplicity of design and construction of the components, and the ease with which these can be replaced; and the robustness of the pump as a whole.
In addition to these features, every centrifugal pump — and thus every dredgepump of this type — possesses a number of physical features which are reflected in its characteristics.
These characteristics, in conjunction with those of the driving mechanism and the pipeline system, determine the performance of the complete installation as a means of transport. The performance, however, is subject to a number of constraints which are related to the nature of the soil to be transported.

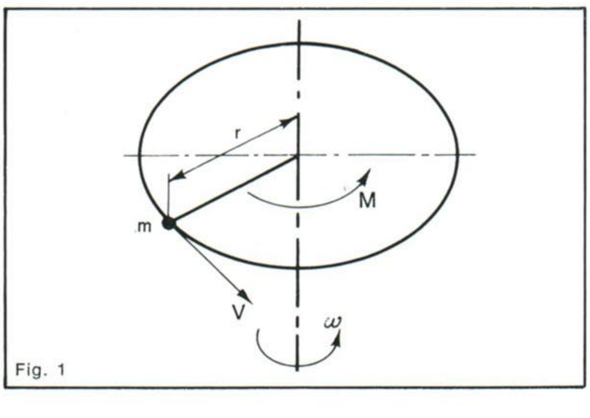

Fig. 1

In this article, the characteristics of pumps will be discussed with reference to a number of theoretical principles, namely the impulse-moment equation, the mathematical model of Euler and the translation of this into the centrifugal pump.

The impulse-moment equation,

For a rotating mass, the impulse-moment equation, states that the moment

$$M = \frac{\text{change of momentum}}{\text{unit of time}} = \frac{d\,(mv.r.)}{dt} \quad \text{(See Fig. 1)}$$

The mass remains constant, so that:

$$M = \frac{d\,(m.v.r.)}{dt} = \frac{md\,(v.r.)}{dt}$$

The mathematical model of Euler

The mathematical model constructed by Euler gives the picture reproduced in Fig 2. Here, the impulse-moment equation is applied with A_u serving as the outgoing control area, and A_i as the incoming control area. Under conditions of loss-free, steady flow, the theoretical moment M_{th}, which is exerted on the fluid (externally) is equivalent to the difference between the impulse moment flow which enters at A_i and which leaves at A_u.

$$M_{th} = \int A_u\,(V_n\,.\,dA\,.\,r\,.\,V_t)_u - \int A_i\,(V_n\,.\,dA\,.\,r\,.\,V_t)_i$$

$$= \rho_u\,.\,V_u\,.\,\sin\alpha_u\,.\,2\pi\,.\,r_u\,.\,b\,.\,r_u\,.\,V_u\,.\,\mathrm{Cos}\,\alpha_u\,-$$

$$-\,\rho_i\,.\,V_i\,.\,\sin\alpha_i\,.\,2\pi\,.\,r_i\,.\,b\,.\,r_i\,.\,V_i\,.\,\cos\alpha_i$$

The flowrate

$$Q = 2\,\pi\,.\,r_u\,.\,b\,.\,V_u\,.\,\sin\alpha_u = 2\,\pi\,.\,r_i\,.\,b\,.\,V_i\,.\,\sin\alpha_i$$

The specific mass is assumed to be constant, so that $\rho_u = \rho_i = \rho$ constant

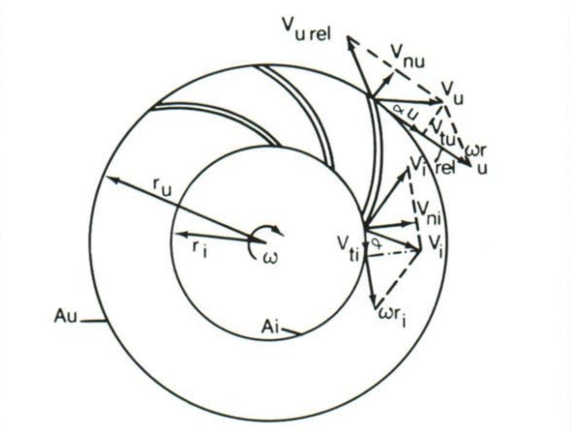

Fig. 2 Impeller width = b

It follows from this that the theoretical moment
$M_{th} = Q \,.\, \rho \; (r_u \,.\, V_u \,.\, \cos\alpha_u - r_i \,.\, V_i \,.\, \cos\alpha_i)$ (1)

The power exerted thus becomes:
$N = M_{th} \,.\, \omega = Q \,.\, \Delta p$ (2)
in which Δp is the pressure head.

It follows from (1) and (2) that the flowrate Q is equal to

$$Q = \frac{M_{th} \,.\, \omega}{\Delta p}$$

thus

$$Q = \frac{Q \,.\, \rho}{\Delta p} \,.\, (r_u \,.\, V_u \,.\, \cos\alpha_u - r_i \,.\, V_i \,.\, \cos\alpha_i) \,.\, \omega$$

The pressure head Δp of the model thus becomes:
$\Delta p = \rho \; (r_u \,.\, \omega \,.\, V_u \cos\alpha_u - r_i \,.\, \omega \,.\, V_i \,.\, \cos\alpha_i)$
$= \rho \; (U_u \,.\, V_u \,.\, \cos\alpha_u - U_i \,.\, V_i \,.\, \cos\alpha_i)$ (3)

As
$V_u \cos\alpha_u = U_u - V_{u\,rel} \cos\alpha_u$ (4)

$$V_{u\,rel} = \frac{V_{nu}}{\sin\beta_u} \qquad (5)$$

$$V_{nu} = \frac{Q}{2\pi\, r_u\, b} \qquad (6)$$

Incorporating equations (4), (5) and (6) in equation (3) shows that the pressure head is equal to:

$$\Delta p = U_u \,.\, \left(U_u - \frac{Q \,.\, \cot\!g\, \beta_u}{2\pi\, r_u\, b}\right) \,.\, \rho - U_i \,.\, \left(U_i - \frac{Q \cot\!g\, \beta_i}{2\pi\, r_i\, b}\right) \,.\, \rho$$

in which ρ is constant, and r_u and r_i, b, β_u and β_i are design constants.
If we assume a uniform motion with a steady flow, the speed of rotation n will be constant, and thus U_i and U_u will also be constant.
If we now introduce two constants C_1 and C_2, where

$$(U_u^2 - U_i^2) \,.\, \rho = C_1$$

$$\left(\frac{U_u \cot\!g\, \beta_u}{2\pi\, r_u\, b} + \frac{U_i \cot\!g\, \beta_i}{2\pi\, r_i\, b}\right) \,.\, \rho = C_2,$$

the relationship between the pressure head Δp and the flowrate Q at the constant speed n will be:
$\Delta p = C_1 - C_2 Q$ (7)
This relationship is shown in Fig. 3.

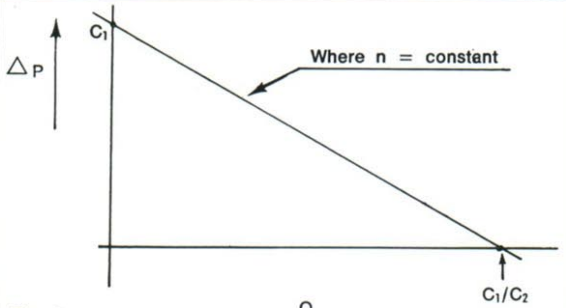

Fig. 3

It follows from equations (2) and (7) that, at a constant speed n, the power can be expressed in the equation
$N = C_1 Q - C_2 Q^2$
The power N, as a function of the flowrate Q, is reproduced in Fig. 4.

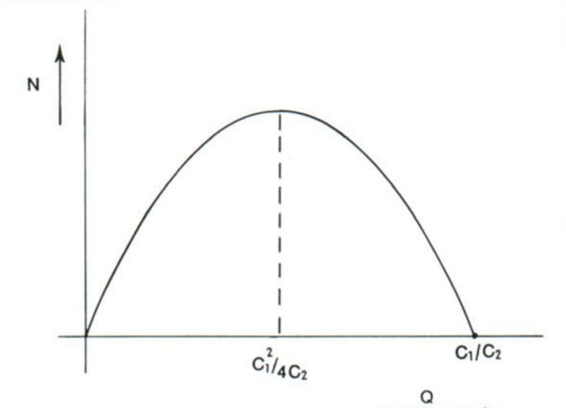

Fig. 4

Translation of the Eulerian model into the centrifugal pump

If we now translate the Euler model into reality, as represented by a centrifugal pump, we see that a number of corrections must be applied to the theoretical curves (see Fig. 5).

a) A correction for the incongruity of impeller blades and flow, the finite number of blades spaced around a circular impeller, the thickness of the impeller blades and the internal friction of the fluid.

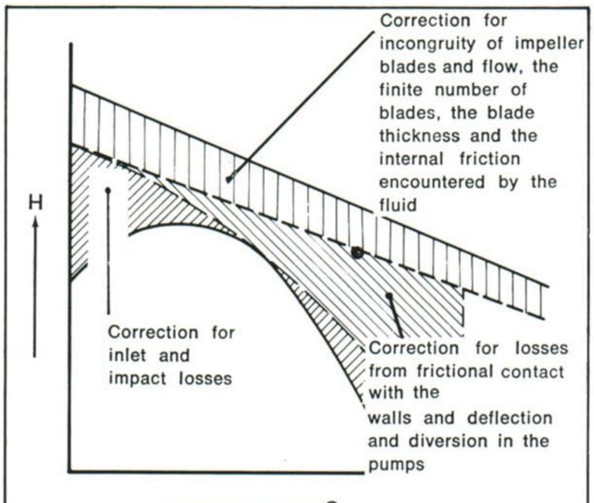

Fig. 5

b) A correction for losses resulting from frictional contact with the walls and deflection and diversion at the pump inlet, in the impeller and in the pump housing. The higher the flowrate of the pump, the greater is the correction required. As the flowrate increases, the role of these losses assumes greater importance, until eventually the greater part of the energy applied to the pump is absorbed in overcoming these and other losses arising at the pump inlet or caused by impact.

c) Correction for inlet and impact losses. For a given pump, the minimum values of these losses coincide with a particular flowrate. Above and below this, losses occur which increase as the flowrate rises or falls.

The characteristics of a dredgepump

The characteristics of a dredgepump generally conform to the pattern shown in Fig. 6. In thus, the pressure head Hman, the efficiency and the driving power of a dredgepump running at constant speed are shown as functions of the flowrate. The relationship between pressure head Hman, efficiency and driving power of a dredgepump can be determined in several ways, e.g. by taking measurements on the pump concerned or an identical unit, while pumping water, or with the aid of calculations.

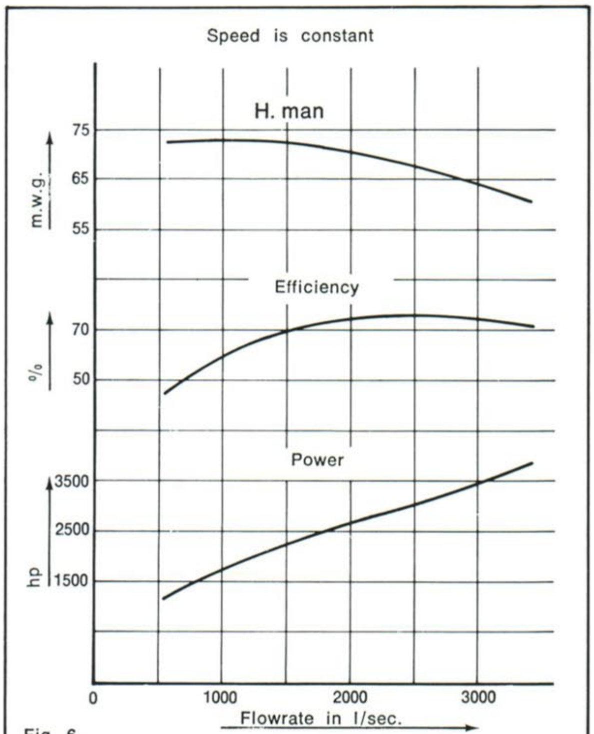

Fig. 6

In the case of IHC standard dredgepumps, the method of calculation has, over a period of many years, been assayed against, and matched to, measurements taken by the Mineral Technological Institute on model pumps and, to an even greater extent, full-size units.

It is of importance that the curve representing the pressure head should be steady, rising slightly as the flowrate decreases. This is desirable in order, on the one hand, to prevent major variations in flowrate at differing mixture concentrations, the other circumstances remaining constant), and on the other to ensure that a small reserve of pressure is available if a tendency towards silting should arise under the influence of a decline in flowrate. The maximum value of the pressure head for a single pump is 8-9 kgf/cm^2.

Attempts to achieve a higher pressure with one pump must be discouraged, since these will produce a sharp drop in efficiency. Moreover, operating at higher pressures will result in heavy wear.

In order to cope with fluctuations in solids concentration and flowrate, and changes in the pipeline arrangement, a dredgepump must have a fairly wide operating range. The efficiency curve must therefore be as flat as possible and, where attain-

able, its highest point must coincide with the design point of the operating range in order that its efficiency throughout this range shall be high. In practice, this is frequently difficult to achieve.
The maximum efficiency of a dredgepump seldom exceeds 75-80%.
If we have the Q-H curve of a pump and its efficiency curve at a given speed, we can calculate from these the corresponding characteristics at a higher or lower speed.
If the variations in speed are of a relatively minor nature, it may be assumed that the flowrate and the pressure head will vary as follows:

1) The flowrate Q is virtually proportional to the speed n,
   thus $\frac{Q1}{Q2} :: \frac{n1}{n2}$

2) The pressure head Mman is virtually proportional to the square of the discharge velocity,
   thus $\frac{H1}{H2} :: \left(\frac{n1}{n2}\right)^2$

3) For such relatively minor variations in speed, it may be assumed that the efficiency will remain constant,
   thus $\eta_2 \approx \eta_1$

This, however, is not strictly true, because the efficiency reaches its peak at the optimum pump speed, and declines at speeds above or below the optimum.
Where the variation in speed is of a relatively minor nature, the assumption that the efficiency remains virtually constant, i.e. that $\eta_1 \approx \eta_2$, is quite tenable.

In support of this statement, the results of test conducted on a model pump at speeds of between 800 and 1,800 rev/min. are reproduced in Fig. 7. The efficiency of the pump in question ranged from 78% to 82%, depending upon the speed.
The maximum efficiency of the model pump will be attained at approximately 1,600 rev/min.
A similar graph relating to a model pump with an impeller of smaller diameter is reproduced in Fig. 8.
It is clear that the maximum efficiency of the pump can be attained with the speed range 1,400 - 1,600 rev/min. But in both cases the speed range is very great, while the variation in efficiency is very small.

4) The necessary power N is proportional to H x Q and may thus be assumed to be virtually proportional to the third power of the speed n
   thus $\frac{N1}{N2} :: \left(\frac{n1}{n2}\right)^3$

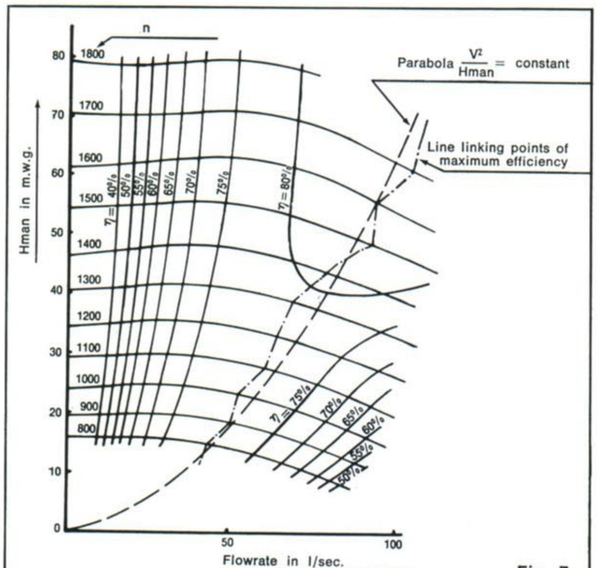

Fig. 7

Fig. 9 shows how the characteristics of the pump change if the speed is varied.
The line connecting the points of maximum pump efficiency is shown in Figs. 7 and 8.
This line, which closely resembles a parabola, passes through the O-point of the graph; it is described by the equation

$$\frac{V^2}{Hman} = \text{constant}$$

The numerical value of the constant in this equation is determined by the dimensions and shape of the pump. The constant will be greater in the case of a wide pump than a narrow one; in other words, where the Hman values are identical, the maximum efficiency of a wide pump coincides with a higher flowrate.

Within their operating range, the so-called low-pressure (hopper) pumps combine a high flowrate with a small pressure head. The value of the constant in the foregoing parabola equation will accordingly be large.

In order to achieve the highest possible efficiency throughout the operating range, low-pressure pumps must therefore be as wide as is practicable. With high-pressure (delivery) pumps, on the other

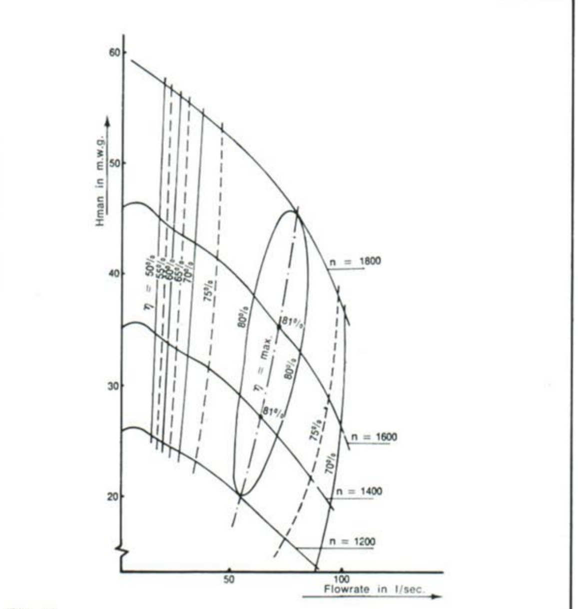

Fig. 8

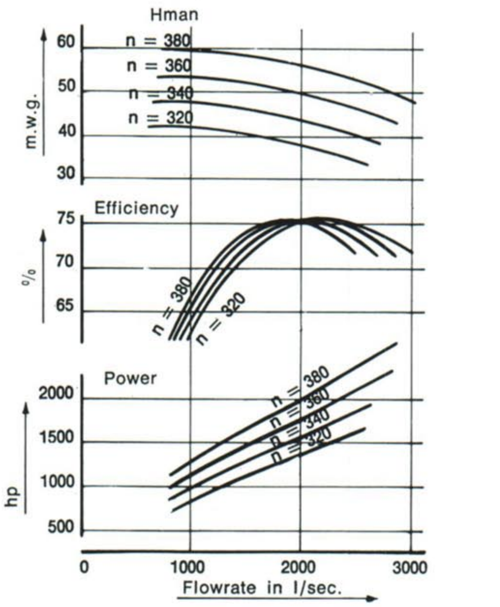

Fig. 9

hand, the flowrate is comparatively low and the pressure head large. The value of the constant in the equation

$$\frac{V^2}{\text{Hman}} = \text{constant}$$

will thus be smaller than is the case with low-pressure pumps.

In order to achieve the highest possible efficiency throughout the operating range, the width of a high-pressure pump must be kept to a minimum. But for reasons connected with the design of the pump — which relate both to wear and to the bore required — the internal diameter of the impeller is in most cases not less than 0.5 times the diameter of the suction inlet.

Computing the pump characteristic for a given drive system characteristic

With the aid of the characteristics shown in Fig. 9, the curves for a pump driven by a constant output, variable-speed engine can be computed fairly simply.

The line drawn through the Q-H curves in Fig. 10, which is determined by way of the power curves at constant speed, is the characteristic for a constant driving power of 1,500 hp.

To determine the characteristics for a pump driven by an engine with constant torque, it is necessary to establish the power available at the various

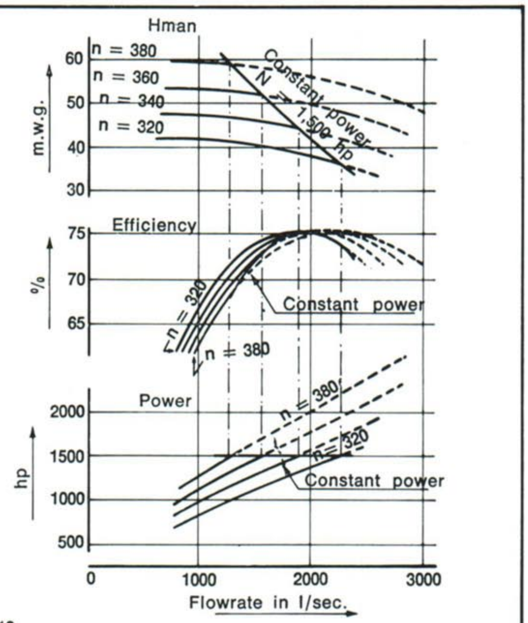

Fig. 10

speeds in the range. This is proportional to the speed of the driving unit.

The power values found to be available at the various speeds are now entered, as points, on the appropriate power curves. The points are then transferred to the appropriate Q-H curves, and are linked.

The line which connects them is the characteristic for constant torque.

In Fig. 11, a diesel engine having a nominal output of 1,500 hp at 380 rev/min formed the starting point. The torque of this engine is 2825 kg/m.
This implies that the output at 360 rev/min.

$$\text{will be } \frac{360}{380} \times 1500 = 1{,}421 \text{ hp.}$$

By the same equation, we arrive at outputs of 1,342 hp at 340 rev/min. and 1,263 hp at 320 rev/min.
The power values thus calculated are now plotted on the power curves.

The points thus obtained are then transferred to the Q-H curves for the relevant engine speeds, after which they are joined by a line, the so-called constant torque line.

It is now a comparatively simple matter to calculate the pump curves for any drive characteristic.

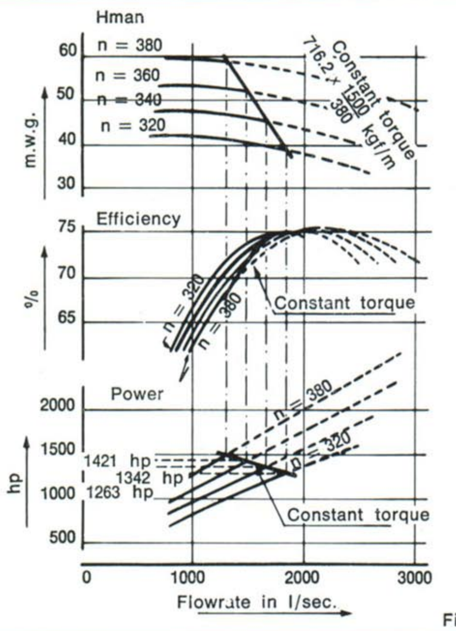

Fig. 11

Varying the impeller diameter
Because the pressure head is governed by the peripheral velocity of the impeller U_u ($= \omega \cdot r_u$), it is influenced not only by the speed of the pump, but also by the outside diameter of the impeller blades ($d_u = 2 \cdot r_u$). Reducing the impeller diameter is therefore a means whereby the pressure head, at constant speed, of an existing dredge-pump can be lowered, enabling the installation to be matched to changing circumstances, e.g. diminished delivery distance.

The nature of the changes in the pump characteristics which result from reducing the impeller diameter is shown in Fig. 12.

The pressure is approximately proportional to the square of the impeller diameter, and the power approximately proportional to the third power of the impeller diameter.

The diameter of an impeller can be reduced by machining or burning metal from the blades.

From the flow technology point of view, however, it is preferable to replace the impeller with one having blades of suitable size and shape. This, however, applies only where the outside diameter of the impeller blades has to be reduced by more than about 10%.

Generally speaking, impellers with a high ru/ri ratio (greater than 2) can be satisfactorily reduced

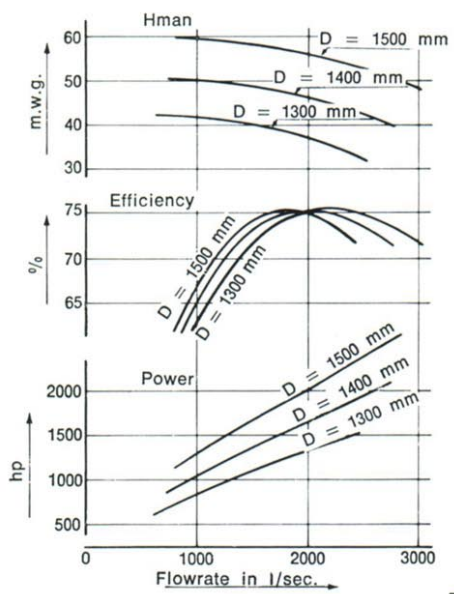

Fig. 12

to 70-80% of the original diameter by machining or flame-cutting. The feasibility of the operation is governed not only by this ratio, but also by other factors relating to the construction of the impeller.

Where conditions require that an impeller with an ru/ri ratio of less than 2 be significantly reduced in diameter, it is usually preferable not to attempt modification, but to replace the unit with one having blades of suitable shape.