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Excel^{TM} in Centrifugal Pump Selection

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Abstract:- To achieve optimum centrifugal pump performance efficiency for liquid flow rates within the range of 4.32 cubic meters per hour - to - 1152 cubic meters per hour, the equations, and method outlined here can be applied to sizing or selecting a centrifugal pump for a process pumping need. The method and derived equations of the hydraulics analysis can also be extended to a typical process centrifugal pumping need. A Microsoft Excel ^{TM} computer program for carrying out and easing the sizing calculations is described.

Keywords:- Centrifugal pumps, Pumps, Pump hydraulics analysis, Centrifugal pump sizing, Centrifugal pump selection, Engineering Computer Programs, Process pumping solution, Pump selection, and Fluid machinery.

I. INTRODUCTION

The workhorse of the Hydrocarbon and Chemical process industry (HCPI), the Centrifugal Pump is usually the most common pump unit applied for liquid transfer in piping applications. The design types are many from several Manufacturers. From the single-stage, small flow applications to the multi-stage units suitable for large volume applications. Matching a Pump Manufacturer's available Centrifugal Pump types to the process operational system requirements, involves analysis of the total system as an integral operating unit. This matching process, often lengthy in analysis with several variables, goes by such names as, Centrifugal Pump Sizing, or Centrifugal Pump Selection, or simply Centrifugal Pump Hydraulics Analysis. A number of software packages are available as off-the-shelf packages to carry out these tasks. Here is how to develop one for your desktop (or Laptop) and Company's process plant needs.

II. BASIC MODEL, APPLICABLE EQUATIONS AND MATHEMATICAL FORMULATIONS

A. Basic Model

Typical installation arrangements for centrifugal pumps are modelled as static suction lift, with the reservoir tank below the pump, or as a static suction head with the reservoir tank above the pump [1]. In some installations the reservoir is inline with the pump suction piping. The fig. 1 shows the general model used for the pump hydraulics analysis. Reservoir tank (1) is the source location for liquid transfer by the pump to delivery tank (4). All reference position coordinates are to a common Datum, "D-L".

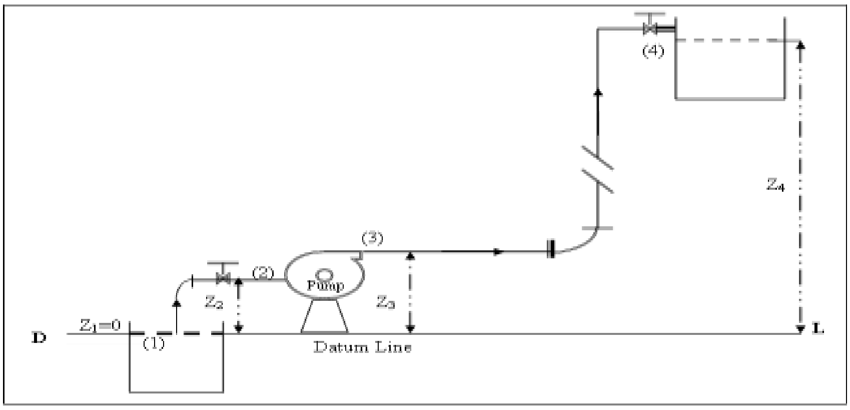


Fig. 1 Basic Model of Centrifugal Pump and Piping Configuration

B. Mathematical Background and Formulations

Mathematical evaluation and analysis of the developed basic model is guided by the Bernoulli and Darcy relations.

1) Bernoulli's Equation: The Bernoulli is concerned with energy balance, and the conservation and conversion to each other of pressure energy (uP), elevation (gZ), and kinetic ($V^2/2$) energies in a fluid stream [2]. In applying the conservation of energy principle to a steady, incompressible, frictionless fluid flow, the Bernoulli equation is stated as:

Mechanical energy = pressure energy + elevation (or potential) energy + kinetic energy

Thus, the Bernoulli equation is defined in terms of the constancy of energy along a fluid stream, as the sum of the pressure energy, elevation, and kinetic energies; and allows for the evaluation at any location point on the streamline where the pressure, elevation and velocity is known [3], [2].

The Energy balance equation is then written as:

$$uP + gZ + \frac{V^2}{2} = constant \tag{1}$$

The Energy Content per kg mass of Fluid at a location, say, (*i*), can then be evaluated from the relation (1a), [3]:

$$E_{F(i)} = P_i u_i + Z_i g + \frac{V_i^2}{2} \tag{1a}$$

For any two points on the *same streamline*, the constant in eq. (1), also allows for equating and comparing flow properties between the two locations, and determining any unknown parameter(s). Thus, for any two locations (*i*, *j*) on the same streamline, the Bernoulli equation is written as in eq. (1b) [3], [2]:

$$u_i P_i + gZ_i + \frac{V_i^2}{2} = u_j P_j + gZ_j + \frac{V_j^2}{2} \tag{1b}$$

However, a look at the left and right sides of eq. (1b), with $u_i = u_j = 1/\rho$, for steady, incompressible flow, and eq. (1b) written as eq. (1c), will show a pressure change or pressure drop – i.e. loss of pressure energy, since each term in eq. (1c) has a pressure unit; and the Bernoulli equation does not account for this loss of energy experienced between two streamlines as the value of the constant in eq. (1) is different for each streamline, and the flow is assumed steady, and frictionless [4], [3], [2].

$$P_i + \rho g Z_i + \frac{\rho V_i^2}{2} = P_j + \rho g Z_j + \frac{\rho V_j^2}{2} \tag{1c}$$

In practice, there is a decrease or loss in mechanical energy due to friction. Thus, enters the Darcy modification!

2) Darcy's Equation: Flow between any two locations ($i \rightarrow j$) in the arrow direction of the flow, experiences a loss of energy or decrease in pressure due to frictional effects. This is termed as pressure drop, $\Delta P_{f(i,j)} = (P_{f(i)} - P_{f(j)})$, and defined in terms of other flow parameters by the Darcy relation, given by eq. (2) [3]:

$$gh = \bar{u}\Delta P_{f-(i,j)} = \frac{\Delta P_{f-(i,j)}}{\bar{\rho}} = L_{e-(i,j)}\frac{fV^2}{2D} \tag{2}$$

By rearrangement, eq. (2) gives the pressure drop as,

$$\Delta P_{f-(i.j)} = \rho g h = L_{e-(i,j)}\frac{f\rho V^2}{2D} = f\left(\frac{L_{e-(i,j)}}{D}\right)\left(\frac{V}{2}\right)(\rho V) \tag{2a}$$

From a consideration of the shearing rate and fluidity [5] of the flow, it can be shown that the pressure drop can be written as in eq. (2b):

$$\Delta P_{f-(i.j)} = \left(\frac{f}{2}\right)\left(\frac{L_{e-(i,j)}}{D}\right)(\tau_w)\left(\frac{\rho V D}{\mu}\right) \tag{2b}$$

The shearing effect on the pipe wall boundary layers, denoted by the shear stress term, τ_w, relates to the friction velocity concept which incorporates wall shear and density; and by implication, relates to the time-averaged mean variation of the pipe radius – a measure of pipe roughness, whatever the flow regime or pipe wall boundary layer texture condition [6], [4], [2]. White [7] notes that the pressure loss is proportional to the wall shear stress regardless of flow boundary inclination – horizontal or tilted [7]. In as new condition, pipe material and fabrication method determine the degree of pipe roughness. However, time in service, changes in operating conditions, and deterioration through aging can pose adverse effects on the pipe walls, by the build-up of deposits [8]. Roughness is thus viewed as a relative concept, whereby the pipe wall surface is taken as sticking out of the boundary sub-layer - a protuberance [6], [2]. The concept of relative roughness allows for comparing the changes in the sub-layer to the original diameter, since this can have effect on the flow capacity.

The Darcy pressure drop, by functional notation, can thus, be written as in eq. (2c):

$$\Delta P_{f-(i.j)} = \psi\left(f, \frac{L_{e-(i,j)}}{D}, \frac{\varepsilon}{D}, \frac{\rho VD}{\mu} \right) \equiv \psi\left(f, L_{e-(i,j)}, D, \varepsilon, \rho, V, \mu \right) \tag{2c}$$

Where, the symbol, Ψ, denotes a functional. The first functional in eq. (2c) is a set of dimensionless relations. This can also be confirmed from a dimensional analysis of the second functional.

When friction effects is accounted for, the modified Bernoulli in terms of the pressure drop is as given by eq. (2d):

$$P_i + \rho g Z_i + \frac{\rho V_i^2}{2} = P_j + \rho g Z_j + \frac{\rho V_j^2}{2} + \Delta P_{f-(i-j)} \tag{2d}$$

Equation (2d) is often expressed as eq. (2e) [6] in terms of the friction head loss term in the Darcy relation:

$$\frac{P_i}{\rho g} + Z_i + \frac{V_i^2}{2g} = \frac{P_j}{\rho g} + Z_j + \frac{V_j^2}{2g} + h_{f-(i-j)} \tag{2e}$$

3) Applicable Piping Flow Equations: The parameters in eq. (2c), relate the flow condition to the geometry of the pipe. The flow regime is defined by the dimensionless relation, $\rho VD/\mu$. This is the familiar Reynolds Number, R_e, and allows for the flow to be characterized as, Laminar, Transition or Turbulent. Thus,

Reynolds Number, R_e:

$$R_e = \frac{\rho VD}{\mu} \tag{3}$$

Flow Velocity:

$$V = \frac{Q}{A} \tag{4}$$

Limitations set on flow line velocities for liquid pump suction and discharge are given by Walas [9]. Walas [9] also provides a centrifugal pump stage selection guide based on flow rate, developed head and efficiency limitations.

Flow-Pipe Cross-Sectional Area:

$$A = \frac{\pi D^2}{4} \tag{5}$$

Under certain flow conditions involving highly viscous liquids, flow control requirement for improved performance, can be obtained with increased Reynolds Number by expressing eq. (3) in terms of, Q, the flow rate [10].

4) Friction Factor: It must be noted that some of the quantities in the first functional of eq. (2c) have interdependence. This has also been shown by dimensional analysis; the friction factor has been confirmed to depend on the relative roughness and Reynolds number only [6].
Thus, by functional notation:

$$f = \psi(\varepsilon/D, R_e)$$

This applies in the turbulent flow region.

For laminar flow, the friction factor depends on the Reynolds Number only and defined by the relation:

$$f = \frac{64}{R_e} \tag{6}$$

The general friction factor chart seen in most fluids mechanics text is based on the work of Lewis F. Moody, after whom the chart is named. The Moody chart is based on the Colebrook-White eq. (6a).

$$\frac{1}{f^{\frac{1}{2}}} = -2Log\left\{ \frac{\left(\varepsilon/D\right)}{3.7} + \frac{2.51}{\left(\sqrt{f}\right)R_e} \right\} \tag{6a}$$

The eq. (6a) is a non-linear equation and an iterative solution method such as Newton-Raphson is required for the solution of the friction factor, f.

The difficulty with the solution for the friction factor, *f*, in the Colebrook-White relation, prompted researchers to find explicit friction factor equations to handle the transition - to - turbulent flow region. Gregory and Fogarasi [11] have listed a number of such equations. One such is the Swamee-Jain [12] relationship for turbulent flow within the following relative roughness and Reynolds number limits range respectively, i.e.

$10^{-6} \le (\varepsilon/D) \le 2 \times 10^{-2}$;

$3 \times 10^{3} \le R_e \le 3 \times 10^{8}$;

The Swamee-Jain relationship is often given in two forms as eq. (6b), and eq. (6c):

$$\frac{1}{f^{1/2}} = -4Log\left[\left(\frac{6.97}{\mathrm{Re}}\right)^{0.9} + \frac{\left(\varepsilon/D\right)}{3.7}\right] \tag{6b}$$

$$f = 0.25\left[Log\left\{\frac{\left(\varepsilon/D\right)}{3.7} + \frac{5.74}{\mathrm{Re}^{0.9}}\right\}\right]^{-2} \tag{6c}$$

Equation (6c) is a more simplified form with the friction factor, *f*, as the subject of the formula.

Pipe roughness for different pipe materials is available from fluid mechanics texts, piping and engineering handbooks [13], [4], [1].

5) Solution to the Colebrook Friction Factor using Microsoft Excel ™: Interpolation search iterative solution approaches based on the Newton-Raphson and the Conjugate Gradient methods are available in Microsoft Excel ™. These are Solver Add-in macro options, modelled after the trial-and-error type Goal Seek desired solution value search method. With the solution form to eq. (6a) written as eq. (7) in the Microsoft Excel ™ cells, the Solver Add-in option dialog box under the Tools menu, allows for desired constraints to be set as follows:

Set Target Cell:
Equal To:
Subject to: Guess value:

To avoid the risk of having a circular reference, i.e., repeated recalculation of particular cell values as input and output it is advised to apply the constraints with caution [14].

$$\frac{1}{f^{\frac{1}{2}}} + 2Log\left\{\frac{\left(\varepsilon/D\right)}{3.7} + \frac{2.51}{\left(\sqrt{f}\right)R_e}\right\} = 0 \tag{7}$$

Additional constraints by way of limitations on the number of iterations, the desired degree of precision, and the final answer convergence level (i.e. the decimal floating points) are provided for in Microsoft Excel ™ to allow for faster and more accurate solutions to non-linear type equations such as eq. (7). As a check, a tolerance percentage gives an indication of the error margin in the iteration calculation.

As an initial estimate in the solution of the Colebrook-White equation, Miller [15] suggests using the Swamee-Jain equation; with a single iteration giving a result within 1% of the Colebrook-White formula. This is applied in the pump program described. Another method which has been reported to give useful result within 5% of the Colebrook-White formula is given by Fox [16], based on an explicit equation derived by Moody from the original Colebrook-White equation.

6) Equivalent Length, the Concept: A typical pipe line will have valves and fittings for flow control and regulation purposes. In addition to the frictional flow resistance and eventual loss of mechanical energy (or loss in pressure energy) posed by the walls of the main, straight pipe section – often termed major loss, other sources of flow resistance due to flow separation/change in direction are the valves and fittings (pipe bends, globe valves, strainers, pipe tees, elbows, etc.) – these are termed minor losses. The industry practice is to reduce such valves and other fittings to an equivalent pipe length. Methods available to accomplish that, are the *K*-loss-factor coefficient method, the equivalent length-to-diameter ratio method, and applied strictly for valves only, is the C_v-valve loss coefficient method [17-21], [2], [1].Other reported recent research methods in the literature are the *2K*- and *3K*- loss factor coefficient methods [22]. Wilson [22] discusses the pros and cons in the slow take-up of applications of the *2K*- and *3K*- methods. The equivalent length-to-diameter ratio method is applied in the program described here.

The concept of equivalent length, allows experimentally derived length-to-diameter (L_e/D) ratios available for different fitting types to be handled in line with eq. (8) and eq. (9) for the pump pipe suction and discharge sections respectively:

System suction location:

$$L_{eS} = L_{pipe-suction} + \sum^{fittings}(L_e/D)D_s \tag{8}$$

System discharge location:

$$L_{eD} = L_{pipe-discharge} + \sum^{fittings}(L_e/D)D_D \tag{9}$$

Tables of values of the loss-factor coefficients, and equivalent length-to-diameter (L_e/D) ratios are available from the Hydraulic Institute, and the Crane Company Publications [18-20]. Preliminary values can be obtained from Engineering Handbooks and Fluid Mechanics text. Note that close observation will show that, values taken from different text can vary for same fitting types. Best practice is to source data from standard references in line with industry regulatory guide.

7) Pressure Loss due to Friction between two Locations, (i and j) [3]*:* Equation (2) can be used to obtain the loss of energy or pressure drop experienced in the pump piping suction and discharge sections of fig. (1) as follows:
(*a*) Between the reservoir and inlet to the pump; and
(*b*) Between the outlet of the pump and the delivery tank as follows:

Suction Side - between system locations (1) and (2):

$$H_{fS} = \bar{u}P_{(1-2)} = \frac{f_S L_{eS} V_S^2}{2D_S} \tag{10}$$

Discharge Side - between system locations (3) and (4):

$$H_{fD} = \bar{u}P_{(3-4)} = \frac{f_D L_{eD} V_D^2}{2D_D} \tag{11}$$

Equations (10) and (11) represent the Darcy loss of energy per unit mass (J/kg) of liquid transferred or specific energy (work done per unit mass of liquid flowing). Douglas *et al.* [23] note that, pumps develop the same specific energy irrespective of gravity.

Some piping engineers recommend adding equal *factors of deterioration* to the calculated suction and discharge head losses to account for aging and the likely decrease in pumping capacity [24 -25]. Suggestions on the application of scaled age multiplier factors for different pipe diameters, which varies incrementally with the age of the piping installation, have also been reported in the literature as allowance for adjusting frictional losses due to aging and capacity loss [26]. Coulson and Richardson [27], observe that in a typical pumping installation, pipe roughness will change with continued usage, and frictional losses cannot then be estimated accurately; with a further cautionary note to design pumping installations with ample excess capacity to guard against cases for corroded pipes, where values of roughness can increase tenfold.

8) Energy Content per unit Mass of Liquid [23], [3]*:*

(a.) At the Reservoir Tank (1):

$$E_{F(1)} = \frac{P_1}{\rho} + Z_1 g + \frac{V_1^2}{2} \tag{12}$$

At the reservoir, $V_1 = 0$, $Z_1=0$ (system reference position)

Therefore, eq. (12) reduces to eq. (12a),

$$E_{F(1)} = \frac{P_1}{\rho} \tag{12a}$$

(b.) At the Pump inlet Flange (2):

$$E_{F(2)} = E_{F(1)} - H_{fS} \tag{13}$$

(c.) At the Pump outlet flange (3):

$$E_{F(3)} = E_{F(4)} + H_{fD} \tag{14}$$

(d.) At discharge tank header (4) – section between Pump outlet flange and discharge tank header:

$$E_{F(4)} = \frac{P_4}{\rho} + Z_4 g + \frac{V_4^2}{2} \tag{15}$$

Where, flow velocity at discharge location, $V_4 = V_D$

9) Energy Input Required by Pump, E_{IRP}: Net input energy required to drive the pump is the difference between the energy content of liquid at the pump outlet and inlet flanges:

$$E_{IRP} = E_{F(3)} - E_{F(2)} \tag{16}$$

This is the energy that must be added to the liquid expressed as a potential energy of a column of liquid of height, H_T - the developed head [3].

10) Total Developed Head Required by Pump, H_T: This represents the energy per unit weight (Joules/Newton = Nm/N) or Head (m), defined by the relation of eq. (17):

$$H_T = \frac{E_{IRP}}{g} \tag{17}$$

C. System Relationships – for Analysing Pumping System Performance

A system analysis represents the overall hydraulic analysis of the total pump and connecting piping system configuration, and follows from the Total Pump Developed Head eq. (17):

$$H = \frac{E_{IRP}}{g} = \frac{\left(E_{F(3)} - E_{F(2)}\right)}{g} = \left(\frac{1}{g}\right)\left\{\left(\frac{P_4}{\rho} + Z_4 g + \frac{V_4^2}{2} + \frac{f_D L_{eD} V_D^2}{2D_D}\right) - \left(\frac{P_1}{\rho} + Z_1 g + \frac{V_1^2}{2} - \frac{f_S L_{eS} V_S^2}{2D_S}\right)\right\} \tag{18}$$

P_1 = pump suction pressure
P_4 = pump discharge/delivery pressure
At the reservoir, $V_1 = 0$; also velocity at discharge location, $V_4 = V_D$

Hence the above equation reduces to,

$$H = \frac{E_{IRP}}{g} = \frac{\left(E_{F(3)} - E_{F(2)}\right)}{g} = \left(\frac{1}{g}\right)\left\{\left(\frac{P_4}{\rho} - \frac{P_1}{\rho}\right) + \left(Z_4 g - Z_1 g\right) + \frac{V_4^2}{2} + \frac{f_D L_{eD} V_D^2}{2D_D} + \frac{f_S L_{eS} V_S^2}{2D_S}\right\} \tag{19}$$

From eq. (4):

$$V_D = \frac{Q}{A_D} \tag{20}$$

$$V_S = \frac{Q}{A_S} \tag{21}$$

Therefore, by substituting in eq. (19) for the suction and discharge piping liquid flow velocities we obtain eq. (22):

$$H = \frac{E_{IRP}}{g} = \frac{\left(E_{F(3)} - E_{F(2)}\right)}{g} = \left(1\!\!\diagup\!\!g\right)\left\{\left(\frac{P_4}{\rho} - \frac{P_1}{\rho}\right) + \left(Z_4 g - Z_1 g\right) + \frac{Q^2}{2A_D^2} + \frac{f_D L_{eD} Q^2}{2A_D^2 D_D} + \frac{f_S L_{eS} Q^2}{2A_S^2 D_S}\right\} \tag{22}$$

By rearrangement, eq. (22) simplifies to eq. (23):

$$H = \frac{E_{IRP}}{g} = \frac{\left(E_{F(3)} - E_{F(2)}\right)}{g} = \left\{\left(\frac{P_4}{\rho g} + Z_4\right) - \left(\frac{P_1}{\rho g} + Z_1\right)\right\} + \left\{\frac{1}{A_D^2} + \frac{f_D L_{eD}}{A_D^2 D_D} + \frac{f_S L_{eS}}{A_S^2 D_S}\right\}\left(Q^2\!\!\diagup\!\!2g\right) \tag{23}$$

The eq. (23) is of the form of eq. (24), i.e.

$$H = X_1 + X_2 Q^2 \tag{24}$$

This is the installed centrifugal pump/piping *operating system (or demand) curve equation* [2].

Where,

$$X_1 = \left\{\left(\frac{P_4}{\rho g} + Z_4\right) - \left(\frac{P_1}{\rho g} + Z_1\right)\right\} \tag{24a}$$

$$X_2 = \left(\frac{1}{2g}\right)\left\{\frac{1}{A_D^2} + \frac{f_D L_{eD}}{A_D^2 D_D} + \frac{f_S L_{eS}}{A_S^2 D_S}\right\} \tag{24b}$$

Thus, the system resistance or friction flow loss is made of a “static” constant elevation component, independent of the flow – X_1, and a dynamic component – X_2Q^2, incorporating the equivalent loss terms of frictional and separation losses in the suction and discharge as defined by the length, diameter and area [23]. By implication, eq. (24) indicates that, the friction loss varies as the square of the flow rate (Q^2), and by relation (19), the square of the velocity (V^2).

D. Static Pressure at Pump Inlet Flange, P_s

This represents the actual pressure of the liquid at the pump inlet flange (location 2), and is determined from a consideration of the energy content of a mass of liquid at the inlet flange, as given by the relation of equation (25).

$$E_{F(2)} = \frac{P_s}{\rho} + Z_2 g + \frac{V_S^2}{2} \tag{25}$$

Z_2 = pump suction flange location.

Or

$$\frac{P_s}{\rho} = E_{F(2)} - Z_2 g - \frac{V_S^2}{2} \tag{26}$$

Or

$$P_s = \rho\left(E_{F(2)} - Z_2 g - \frac{V_S^2}{2}\right) \tag{27}$$

Equation (27) is a way of expressing the conversion to pressure energy of the other energies [2].

E. Net Positive Suction Head (NPSH)

The net positive suction head is of two forms- net positive suction head available (NPSHA) and the net positive suction head required (NPSHR). In practice to deliver flow through the pump, the required condition is: (*NPSHA)* ≥ (*NPSHR)*, i.e. the NPSHA must always be equal to or greater than the NPSHR [28], [1]. NPSHR is usually supplied by the pump manufacturer. NPSHA is defined by eq. (28):

$$NPSHA = \frac{(P_s - P_v)}{\rho g} \tag{28}$$

Where, P_v = Vapour pressure

By eq. (28), the liquid static pressure must be greater than the pumping liquid vapour pressure. This helps prevent a loss of head and possible loss of flow, through flash vaporization or boiling of the liquid which can lead to the condition of cavitation, or vaporization at the pump eye inlet, which often causes unstable flow, can destroy impeller, and hence, the pump [28]. Thus, NPSH is the head required at the low pressure pump suction inlet to keep the liquid from vaporizing [7].

One way to keep *NPSHA>NPSHR* is to set the height of the pump, by adjusting, Z_2 in eq. (27) [7]. By rewriting eq. (28) as eq. (28a), the right-hand side can then be adjusted, to be equal to or greater than the left-hand side which is obtained from the Manufacturer in form of curves [7].

$$NPSH \equiv \frac{(P_s - P_v)}{\rho g} = \frac{1}{g}\left(E_{F(2)} - Z_2 g - \frac{V_S^2}{2}\right) - \frac{P_v}{\rho g} \tag{28a}$$

Other safe margin ways [29] to prevent unstable operation and the risk of liquid vaporization have been reported in the literature. Streeter [30] and Douglas *et al.* [23] discuss the usefulness of the concept of cavitation parameter in predicting the inception of cavitation. The likely effects of changes in seasonal climate and environmental conditions to stable operation without cavitation has been noted by Douglas *et al.* [23], citing as reason the dependence of vapour pressure on temperature.

F. Pump Efficiency, η_p

Corripio *et al.* [31] developed an efficiency relationship as a function of flow rate for useful flow range, within the range of application of most centrifugal pumping needs. This is defined by eq. (29):

$$\eta_P = 0.885 + 0.00824(LnQ) - 0.01199(LnQ)^2 \tag{29}$$

Range of Application: (0.0012 m^3/s – 0.32 m^3/s); (0.072 m^3/min – 19.2 m^3/min); (4.32 m^3/hr - 1152 m^3/hr)

An alternative pump efficiency relation which though applies for a limited flow range and developed head, but useful for computer application is given by Coker [32] and McAllister [21]. Dickenson [10] also gives a relationship, derived based on statistical evaluation of turbine driven pumps covering larger flow ranges. The Corripio *et al.* [31] method is applied in the program described.

G. *Pump Power, P_w*

In order to select a nominal driver size for the driven centrifugal pump, the brake power required to deliver the flow must be known. This is given by the relation of eq. (30):

$$P_W = \frac{\rho Q E_{IRP}}{\eta_p} = \frac{\rho Q g H}{\eta_p} \tag{30}$$

The driver power required, P_r, is then computed with eq. (30a):

$$P_r = \frac{P_w}{\eta_d} \tag{30a}$$

Where, η_d = driver efficiency.

For an electric motor driver, the driver (motor) efficiency can be estimated by a correlation given by Corripio *et al.* [31].

III. PUMP OPERATING SYSTEM CURVE

The Pumping System Head-Capacity relation is as defined by eq. (24). A good pump hydraulic analysis will usually result in calculation of the total possible head from the maximum permissible flow rate. Fluctuations or deviations as a result of variable flow in the system can be analysed for, by applying eq. (24)

whilst varying the flow within the range, zero-to-maximum flow, and then obtaining the corresponding values of Head. Such graphs are of the form of fig. (2). A table can be drawn up resulting in a graphical plot of Operating System Head *versus* Flow Capacity (system *H-Q* curve).

The Program can be prepared to automatically handle the table and graphical plot, through initial design using Excel formatting together with Microsoft Visual Basic for Applications (VBA) Subroutines codes.

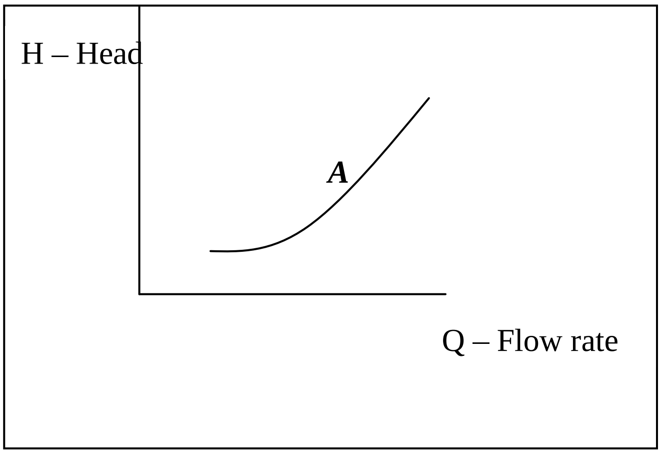


Fig. 2: Pump System Head Curve

IV. PUMP PERFORMANCE CURVES FROM MANUFACTURERS' CATALOGUES

Pump Manufacturers provide catalogues of available pump types; and pump performance curves in form of Head (*H*) *versus* Flow rate (*Q*) are usually provided in such catalogues. Pump Performance characteristic (or *supply curves*) [2] are experimentally derived based on certain tests conditions. These Manufacturers' supplied characteristic curves are usually of the form of fig. (3).

Talwar [33] gives the relationship between Head (*H*) and flow capacity (*Q*) for a centrifugal pump expressed in the form of the quadratic eq. (31). This is a parabolic curve fit model.

$$H = a + bQ + cQ^2 \tag{31}$$

Where, ***a***, ***b***, and ***c*** are called pump constants.

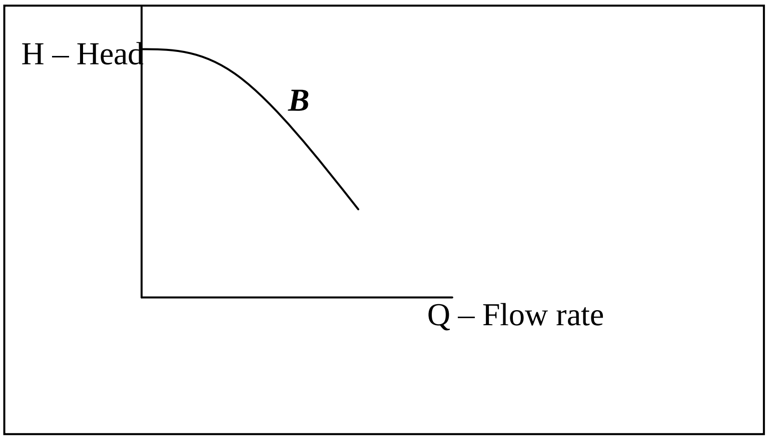


Fig. 3: Pump Characteristic Curve from Manufacturer's Catalogue

The manufacturers' test conditions are on instrumented test rig, usually at a measured specified constant speed, and allows for flow rate to be varied from zero-flow (shut-off) to maximum flow [34]; and flow rate-versus-head relationship plotted for different pump models in sales and marketing catalogues. The purpose of such pump manufacturers' information is to enable pump users to match available pump models to specific process pumping needs, which in most instances varies from one customer pump end-user to another.

V. ESTABLISHING RELATIONSHIP BETWEEN MANUFACTURER'S PUMP CURVE AND OPERATING SYSTEM CURVE – "DETERMINATION OF PUMP OPERATING POINT"

By analysing a pump manufacturer's curve as provided in the Pump catalogue of available pump types, and extracting any two corresponding Head-Flow Capacity (*H-Q*) points say:

$$H_{1y} = \beta_{1y} \qquad\qquad H_{2y} = \beta_{2y}$$
$$Q_{1x} = \alpha_{1x} \qquad \text{[and]} \quad Q_{2x} = \alpha_{2x}$$

Expressed as (*x-y)* coordinates, we have:

(Q_{1x}, H_{1y}) and $(Q_{2x}, H_{2y}) \equiv.$ $(\alpha_{1x}, \beta_{1y})$ and $(\alpha_{2x}, \beta_{2y})$ respectively.

Two equations are then obtained from substitutions for the Head (*H*) and flow-rate (*Q*) in eq. (31) as follows:

$$H_{1y} = a + bQ_{1x} + cQ_{1x}^2 \tag{31a}$$

$$H_{2y} = a + bQ_{2x} + cQ_{2x}^2 \tag{31b}$$

A third equation involves the system-head equation curve of eq. (24). Equations (24), (31a), and (31b) in the form of the three equations derived can be solved simultaneously to obtain the pump constants, ***a***, ***b*** and ***c*** in equation (31).

The derived values of the pump constants are defined by the equations (32), (33) and (34):

$$a = \beta_{1y} - \left(b\alpha_{1x} + c\alpha_{1x}^2\right) \tag{32}$$

$$b = \frac{\left(\beta_{2y} - \beta_{1y}\right) - c\left(\alpha_{2x}^2 - \alpha_{1x}^2\right)}{\left(\alpha_{2x} - \alpha_{1x}\right)} \tag{33}$$

$$c = \frac{\left(\beta_{3y} - \beta_{1y}\right) - \left[\dfrac{\left(\beta_{2y} - \beta_{1y}\right)\left(\alpha_{3x} - \alpha_{1x}\right)}{\left(\alpha_{2x} - \alpha_{1x}\right)}\right]}{\left(\alpha_{3x}^2 - \alpha_{1x}^2\right) - \left[\dfrac{\left(\alpha_{2x}^2 - \alpha_{1x}^2\right)\left(\alpha_{3x} - \alpha_{1x}\right)}{\left(\alpha_{2x} - \alpha_{1x}\right)}\right]} \tag{34}$$

Where, α_{3x}, and β_{3y} are points on the System-Head curve. In the program these can be preset to the calculated system maximum delivery Head from the known maximum permissible flow rate.

Thus, with the constants obtained, a Head (*H*) - Flow rate (*Q*) table can be drawn up *to simulate a plot* of the manufacturer's pump performance curve.

By plotting, *on the same graph* the pumping system – curve (***A***) shown in fig. 2, and *simulated manufacturer's pump curve* (***B***) of fig. 3, the desired operating point (best efficiency point - BEP) of the pump can be obtained from the intersection of the system-curve and the simulated manufacturer's pump curve. The form of the curve is as shown in fig. (4). A user-defined pump curve can also be plotted. White [7] notes that, in a typical performance curve, the head is approximately constant at low discharge flow, and then drops to zero at maximum flow. In plotting a user-defined chart, this is an important consideration. This will also guard against having a positive slope (see fig. 4), which White [7] notes, can be unstable in pumping operations leading to surge or oscillations in pumping conditions with the difficulty in locating an actual operating point – also referred to as a hunting or rough operational condition. The sloping effect relates to the pump impeller blade types and outlet blade angle [7], [23]; certain blade types are recommended.

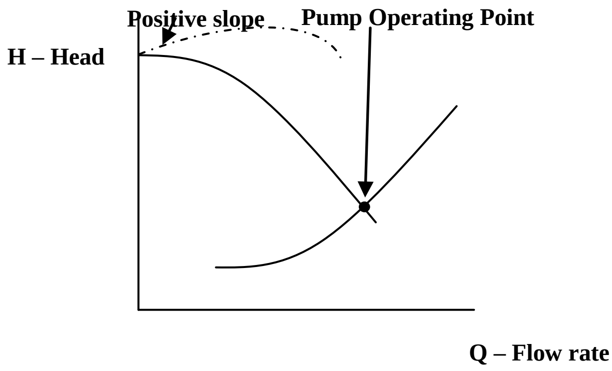


Fig. 4 Pump Performance Curve showing Operating Point

VI. PUMP POWER AND EFFICIENCY VERSUS FLOW RATE CURVES

For a given pump machinery, the head, power and efficiency characteristics are functions of the flow-rate [34]. Thus, in addition to the *H-Q* curve, other curves of interest are the brake-power *versus* flow-rate (P_w-Q), and the pump efficiency *versus* flow-rate curves (η_p-Q). These plots can be displayed as two graphs of (Head and Power) *versus* flow rate, and (Head and Pump Efficiency) *versus* flow rate. The plots can be programmed to be displayed on same graph using the global calculation link sheet (see numerical example fig.9). Such plots are shown in the Numerical example.

VII. PUMP SPEED

The data obtained in Manufacturers' catalogues of available pumps, are for centrifugal pumps tested at particular speeds. By matching a pump from the manufacturer's catalogue to the needs of the process operations, the nominal operating pump speed is thus specified. Pump operating conditions can change – speed, flow rate, impeller size, power .The affinity rules based on the concept of geometric similarity, allow for estimating pump performance conditions due to changes in speed, size and power; empirical correlations to achieving maximum efficiency due to changes in size and flow rate are available in the open literature [7]. Davidson [35] gives an approach to determining the likely limit on rotational speed based on the required NPSH. Davidson also states conditions under which it is cost effective to operate pumps with variable speed drives [35].

VIII. FLUID PROPERTIES FUNCTIONS

Fluid Physical properties, density, viscosity, and vapour pressure of typical pumping liquids can be estimated from the pumping liquid temperature using the derived relationships of Yaws [36-37]. Reid *et al.* [38] is also a good source for fluid properties data.

The Microsoft Excel ^{TM} functions category is useful in building a database of liquid properties' functions in line with the mathematical function, *function* (Temperature), i.e. Ψ(T).

When developing fluid properties functions data pack in Microsoft Excel ^{TM}, it is recommended to follow a structured naming convention to allow for an error free process in selection using the drop-down list button. One useful method given by ALIGNAgraphics [39-40] is:

Name of property_ (temperature)

As an Example, using water: **rhoWater**(temperature); **viscWater**(temperature); **Pvapwater(temperature).**

Where, ***rhoWater***, ***viscWater,*** and ***PvapWater*** are the function names for water density, liquid viscosity and vapour pressure respectively. Typical example liquid properties function sub-routine program coding for water density, viscosity and vapour pressure respectively in the Visual Basic module of the ALIGNAgraphics pump and piping programs [39-40] are:

Water Density Function:

```
Function rhoWater(temperature)
  If 0 <= temperature And temperature <= 374.15 Then
  rhoWater = 1000 * 0.3471 * 0.274 ^ (-(((1 - ((temperature + 273.15) / 647.3))) ^ (2 / 7)))
  Else
```

```
  sngRetVal1 = MsgBox(conMsg, conBtns, "ALIGNAgraphics limiting temperature range for density.")
  End If
End Function
```

The Yaws [36-37] derived curve-fitted density relationships are, for certain fluid types, functions of reduced temperature ($T_r = T/T_c$). Thus, a good data source for critical temperature (T_c) properties for different liquid types is required to build a liquid functions database.

Water Viscosity Function:

```
Function viscWater(temperature)
   If 0 <= temperature And temperature <= 374.15 Then
  viscWater = 0.001 * (10 ^ (-10.73 + (1828 / (temperature + 273.15)) + (0.01966 * (temperature + 273.15)) + (-
0.00001466 * ((temperature + 273.15) ^ 2))))
  Else
  sngRetVal1 = MsgBox(conMsg, conBtns, "ALIGNAgraphics limiting temperature range for viscosity.")
  End If
End Function
```

Water Vapor Pressure Function:

```
Function PvapWater(temperature)
   If 0 <= temperature And temperature <= 374.15 Then
  PvapWater = 133.3224 * (10 ^ (16.373 + (-2818.6 / (temperature + 273.15)) + (-1.6908 * (Log(temperature +
273.15) / Log(10))) + (-0.0057546 * (temperature + 273.15)) + (0.0000040073 * ((temperature + 273.15) ^ 2))))
  Else
  sngRetVal1 = MsgBox(conMsg, conBtns, "ALIGNAgraphics limiting temperature range for vapour pressure.")
  End If
End Function
```

The functions database can be extended to several liquid types. The described program has a functions list of 48 liquids. These are easily selected as click-and-drop functions as they form part of the other Microsoft Excel™ Functions pack. The program can also be prepared to handle user-defined liquid properties values for which no direct correlations exists.

IX. FLOW CHART FOR DEVELOPING A CENTRIFUGAL PUMP SELECTION PROGRAM

The chart (fig. 5) shows layout of requirements for developing a Centrifugal Pump Hydraulics Analysis/Selection computer program using Microsoft Excel ™. The chart is provided as a checklist guide of things to take into consideration to develop an effective centrifugal pump selection program.

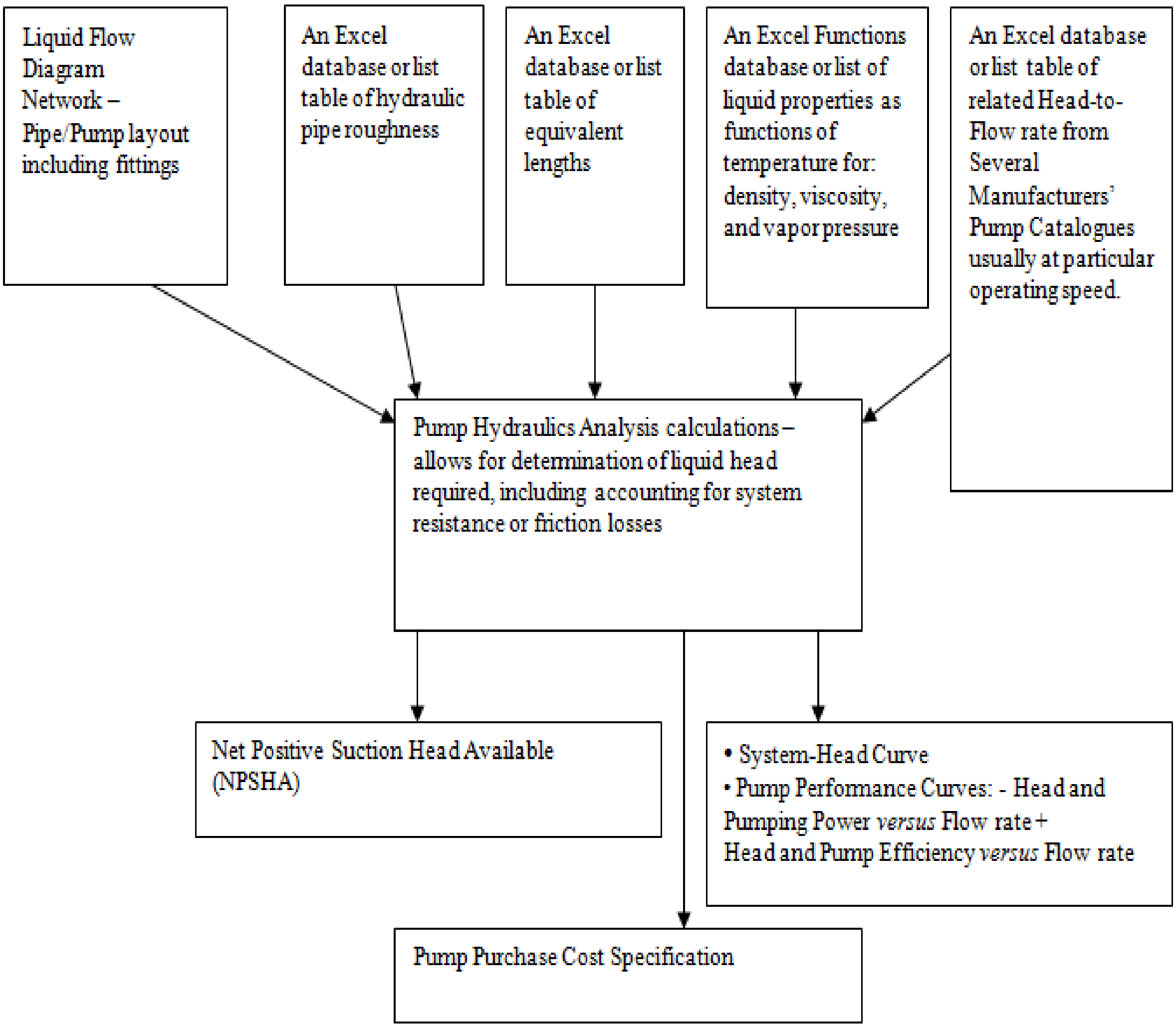


Fig. 5: Layout of Requirements for a Centrifugal Pump Selection Software Program Development

A detailed flow chart systems approach to centrifugal pump selection is given by Davidson [35].

X. Developing Internal Databases/Excel Data Lists and Linking Cells

Using as example, an abridged typical layout for the database of pipe roughness of commercial steel and cast iron pipe materials is given in Table 1:

Table 1: Typical Microsoft Excel data List for Pipe Roughness Values, ε

Pipe i.d	Type of Pipe Material	Pipe Roughness
1	Commercial Steel	0.04572
2	Cast Iron	0.25908

Data Source: Perry's Chemical Engineer's Handbook [1];
Data List Table Layout Source: ALIGNAgraphics Pumpro Program [39 - 40]

From the table, the button-linked cell is in the first column.
The method can be extended to a data list for equivalent length-to-diameter ratios; and head-versus-flow rate from pump manufacturers' catalogue as shown by the example of the abridged lists (Table 2 and Table 3) respectively as follows:

Table 2: Typical Microsoft Excel data List for selected equivalent Length-to-Diameter Ratios (Le/D)

Equiv. i.d	Fitting Type	Le/D
1	Gate Valve, fully open	13
2	90-degree long radius Elbow	30

(Le/D) Data Source: Crane [18] *and Hydraulic Institute* [20]*;*
Data List Table Layout Source: ALIGNAgraphics graphics Pumpro Program [39-40]

Table 3: Typical Microsoft Excel List for extracted points from Manufacturer's Pump Curve

i.d	Manufacturer/Pump Type	Pump type				
			Q1	H1	Q2	H2
1	User-Defined					
2	Manufacturer 1	1" at 2900 rpm	0	16	0.6	15
3	Manufacturer 2	1"/11/4" at 1450 rpm	5.46	22	6.84	20

Source: ALIGNAgraphics Pumpro Program [39- 40]

Note that it is good practice to keep the pipe roughness, equivalent length-to-diameter of fittings and valve data, and Manufacturers' Catalogue data on separate Microsoft Excel data list or database sheets, and dynamically link the sheets to the centrifugal pump hydraulics analysis global calculation sheet which is then again dynamically linked to the centrifugal pump selection/specification sheet. This allows for proper tracing of errors during the program development and upgrade phases. The VLOOKUP () function is useful in such VBA macro function code database linking. The INDEX() function can also be used. As an example, to link the Equivalent length Microsoft Excel data list, the following is used in the described program [40]:

VLOOKUP($C5,'EQUIVALENT LENGTHS OF FITTINGS'!$A$4:$C$40,2,FALSE)

Where, EQUIVALENT LENGTHS OF FITTINGS is the worksheet name of the database for the Equivalent Length of fittings and valves as in the layout shown in Table 2.

XI. OBTAINING DATA FOR INTERNAL DATABASE

Sources of required data for the program are vast. For very basic small programs, Fluids Mechanics, Chemical Engineering textbooks, and General Engineering Handbooks are good sources. For very extensive programs, fluids texts are additionally supplemented and supported with publications of piping standards and codes by the Hydraulic Institute and Engineering Societies plus published engineering journal articles.

XII. PUMP PURCHASING COSTS AND LIFE CYCLE MANAGEMENT

The costing method of Corripio *et al.* [31], with the prescribed flow rate limitations, can be adopted in estimating pump cost. Useful cost data sources are available from some publishers [41, 42]. The method of Jack and Lilly [43] can be applied to the life cycle cost management of a typical centrifugal pump installation. A collaborative executive summary report (44) by the Hydraulic Institute, Europump, and the United States Department of Energy on pump life cycle cost (LCC) is also a good reference.

XIII. APPLICATION EXAMPLE

A numerical example of the typical calculations requirements for selection of a centrifugal pump is shown below.

A. Numerical Example [40]

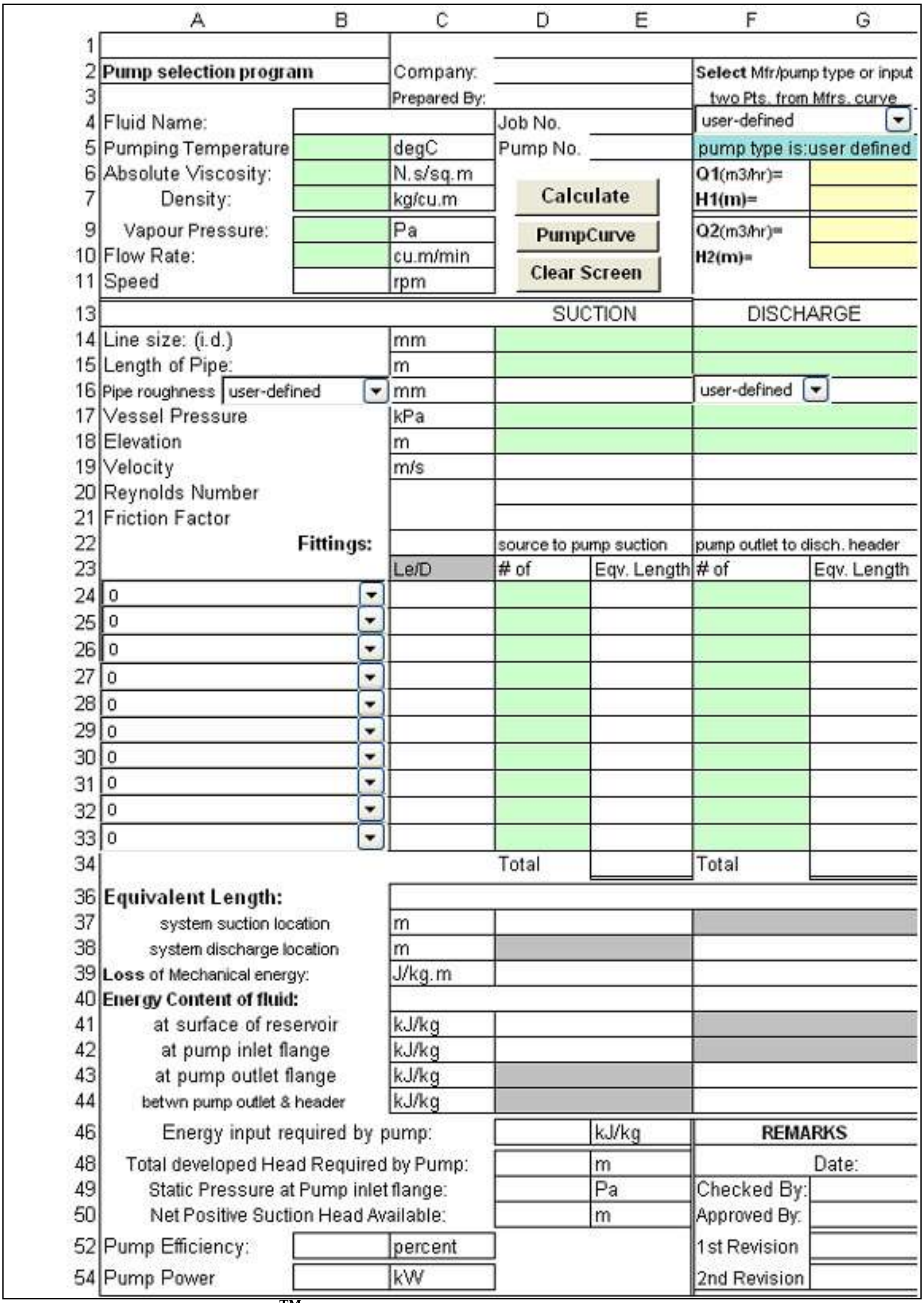
The pumping requirements of the utility process section of a major refinery are a piping system designed to provide 120 m^3/hr of water to a discharge header at a pressure of 440 kPa. The pumping water at 25°C is from a vessel at 90 kPa. Conduct a detailed analysis of the pumping system, if the discharge header and pump are to be maintained at 12 m and 0.5 m elevations respectively. The system consists of 20 m of 200 mm suction pipe and 60 m of 100 mm discharge pipe, both commercial steel. Also included:

Suction Side: ***Fittings in Suction Line:***

#1 Bellmouth entry; #4 90-degree standard radius elbows; #2 Gate Valves, #2 Check Valves; #1 Swing Check valve

Discharge Side: ***Discharge Side Fittings:***

Entrance (enlargement) into the delivery vessel, #4 90-degree standard radius elbows; #1 Gate Valve; #1 Swing Check Valve; #1 Globe Valve; #1 Tee, side outlet



	A	B	C	D	E	F	G
1							
2	**Pump selection program**		Company:			**Select** Mfr/pump type or input	
3			Prepared By:			two Pts. from Mfrs. curve	
4	Fluid Name:			Job No.		user-defined	
5	Pumping Temperature		degC	Pump No.		pump type is:user defined	
6	Absolute Viscosity:		N.s/sq.m			**Q1**(m3/hr)=	
7	Density:		kg/cu.m	Calculate		**H1(m)=**	
9	Vapour Pressure:		Pa	PumpCurve		**Q2**(m3/hr)=	
10	Flow Rate:		cu.m/min			**H2(m)=**	
11	Speed		rpm	Clear Screen			
13				SUCTION		DISCHARGE	
14	Line size: (i.d.)		mm				
15	Length of Pipe:		m				
16	Pipe roughness	user-defined	mm			user-defined	
17	Vessel Pressure		kPa				
18	Elevation		m				
19	Velocity		m/s				
20	Reynolds Number						
21	Friction Factor						
22		**Fittings:**		source to pump suction		pump outlet to disch. header	
23			Le/D	# of	Eqv. Length	# of	Eqv. Length
24	0						
25	0						
26	0						
27	0						
28	0						
29	0						
30	0						
31	0						
32	0						
33	0						
34				Total		Total	
36	**Equivalent Length:**						
37	system suction location		m				
38	system discharge location		m				
39	**Loss** of Mechanical energy:		J/kg.m				
40	**Energy Content of fluid:**						
41	at surface of reservoir		kJ/kg				
42	at pump inlet flange		kJ/kg				
43	at pump outlet flange		kJ/kg				
44	betwn pump outlet & header		kJ/kg				
46	Energy input required by pump:				kJ/kg	**REMARKS**	
48	Total developed Head Required by Pump:				m		Date:
49	Static Pressure at Pump inlet flange:				Pa	Checked By:	
50	Net Positive Suction Head Available:				m	Approved By:	
52	Pump Efficiency:		percent			1st Revision	
54	Pump Power		kW			2nd Revision	

Fig. 6: Microsoft Excel TM Formatted Centrifugal Pump Hydraulic Analysis Spec Sheet
Source: [ALIGNAgraphics Centrifugal Pump Hydraulic Analysis Program Manual] [40]

Pump selection program		Company:			**Select** Mfr/pump type or input	
		Prepared By:			two Pts. from Mfrs. curve	
Fluid Name:	Water		Job No.		user-defined	
Pumping Temperature	25	degC	Pump No.		pump type is: user defined	
Absolute Viscosity:	9.11E-04	N.s/sq.m			**Q1**(m3/hr)=	0
Density:	1.03E+03	kg/cu.m			**H1(m)**=	55
Vapour Pressure:	3.17E+03	Pa			**Q2**(m3/hr)=	4.32
Flow Rate:	2	cu.m/min			H2(m)=	55
Speed		rpm				

		SUCTION	DISCHARGE
Line size: (i.d.)	mm	200	100
Length of Pipe:	m	20	60
Pipe roughness	mm	0.04572	0.04572
Vessel Pressure	kPa	90	440
Elevation	m	0.5	12
Velocity	m/s	1.0610	4.2441
Reynolds Number		2.393E+05	4.786E+05
Friction Factor		0.01690	0.01743

Fittings:		source to pump suction		pump outlet to disch. header	
	Le/D	# of	Eqv. Length	# of	Eqv. Length
bellmouth inlet	4	1	0.80	0	0.00
sudden enlargement	45	0	0.00	1	4.50
90 deg standard elbow	30	4	24.00	4	12.00
Gate valve, fully open	13	2	5.20	1	1.30
swing check valve,fully open	135	1	27.00	1	13.50
Glo. valve,perp.to run,fully open	340	0	0.00	1	34.00
tee-branch flow	60	0	0.00	1	6.00
0	0		0.00		0.00
0	0		0.00		0.00
0	0		0.00		0.00
		Total	57.00	Total	71.30

Equivalent Length:			
system suction location	m	77	
system discharge location	m		131.30
Loss of Mechanical energy:	J/kg.m	3.73E-01	2.10E+01
Energy Content of fluid:			
at surface of reservoir	kJ/kg	87.59	
at pump inlet flange	kJ/kg	87.22	
at pump outlet flange	kJ/kg		554.94
betwn pump outlet & header	kJ/kg		575.95

Energy input required by pump:	488.74	kJ/kg	**REMARKS**	
Total developed Head Required by Pump:	49.82	m	Date:	
Static Pressure at Pump inlet flange:	83997.992	Pa	Checked By:	
Net Positive Suction Head Available:	8.02	m	Approved By:	
Pump Efficiency:	71.83	percent	1st Revision	
Pump Power	23.31	kW	2nd Revision	

Fig.7: Centrifugal Pump Selection Spec Sheet Results with input/output for the Numerical Example
Source: [ALIGNAgraphics Pumpro Manual] [40]

	Friction Factor		fittings buttons selector	
	suction	discharge	dd2	28
solverSolve	**solution**	**solution**	dd3	30
sngRecipfric	7.6922209	7.5740228	dd4	15
			dd5	6
Relative Roughness	0.00023	0.00046	dd6	12
			dd7	2
sngFricfunction	0.000366904	6.88209E-05	dd8	27
			dd9	1
sngFriction	0.016900381	0.017431983	dd10	1
			dd11	1
button Selector				
lstManufacturer	1			

Fig. 8: Buttons Selector and Friction Factor Solver Linked cells
Source: [ALIGNAgraphics Pumpro Program] [40]

				suction		discharge	
lstSuctRoughness	4	Commercial steel	0.04572				
lstDischRoughness	4	Commercial steel	0.04572	suction		discharge	
		Fitting type	Le/D	# of	Le	# of	Le
dd2	28	bellmouth inlet	4	1	0.8	0	0
dd3	30	sudden enlargement	45	0	0	1	4.5
dd4	15	90 deg standard elbow	30	4	24	4	12
dd5	6	Gate valve, fully open	13	2	5.2	1	1.3
dd6	12	swing check valve,fully open	135	1	27	1	13.5
dd7	2	Glo. valve,perp.to run,fully open	340	0	0	1	34
dd8	27	tee-branch flow	60	0	0	1	6
dd9	1	0	0	0	0	0	0
dd10	1	0	0	0	0	0	0
dd11	1	0	0	0	0	0	0
			Total (m)		57		71.3

			SYSTEM CHARACTERISTICS/PUMP CURVE: sys. curve fit : $H=X1+X2.Q2$			
ponits form			flow rate	system-**H-Q**	Mfrs-**H-Q** curve	efficiency
Mfrs. curve	Constants		(cu.m/hr)	(m)	**H**, (m)	%
& Calc Hp/Q	X_1 (m)	X_2 (hr^2/m)	Q	H		e
	46.72	2.1515E-04	4.32	46.7263	55.00	28.73
Q1	46.72	2.1515E-04	30	46.9159	54.71	57.07
0	46.72	2.1515E-04	60	47.4968	53.75	65.03
H1	46.72	2.1515E-04	90	48.465	52.12	69.14
55	46.72	2.1515E-04	120	49.820	49.82	71.83
	46.72	2.1515E-04	150	51.563	46.85	73.77
Q2			flow rate	system-**H-Q**	Mfrs-**H-Q** curve	Power
4.32	Eqn: $H=a+bQ+cQ2$		(cu.m/hr)	(m)	**H**, (m)	kW
H2	**Pump constants**		Q	H		P
55	**a**	180.4462	4.32	46.7263	55.00	1.97
Q3	**b**	0.00120	30	46.9159	54.71	6.90
120	**c**	-6.31E-05	60	47.4968	53.75	12.27
H3			90	48.465	52.12	17.66
49.82			120	49.820	49.82	23.31
		Mfrs-**H-Q** curve	150	51.563	46.85	29.36

Fig. 9: Results from Global Calculation Link Sheet for Spec Sheet for Example
Source: [ALIGNAgraphics Pumpro Program] [40]

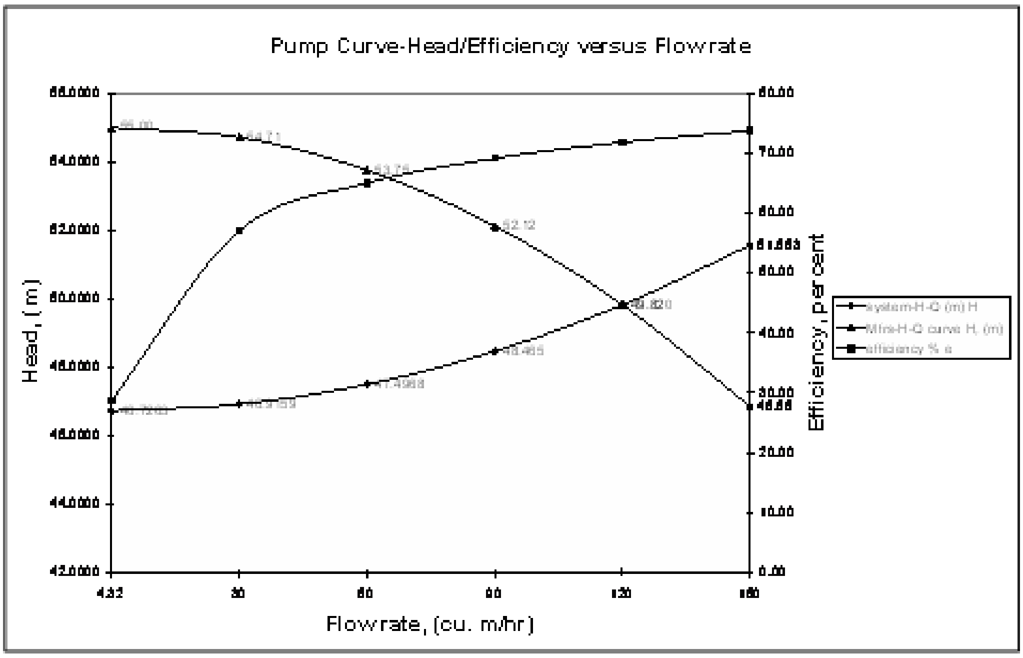


Fig. 10: Performance Curves of Head and Efficiency *versus* Flow rate for Example
Source: [ALIGNAgraphics Pumpro Program] **[40]**

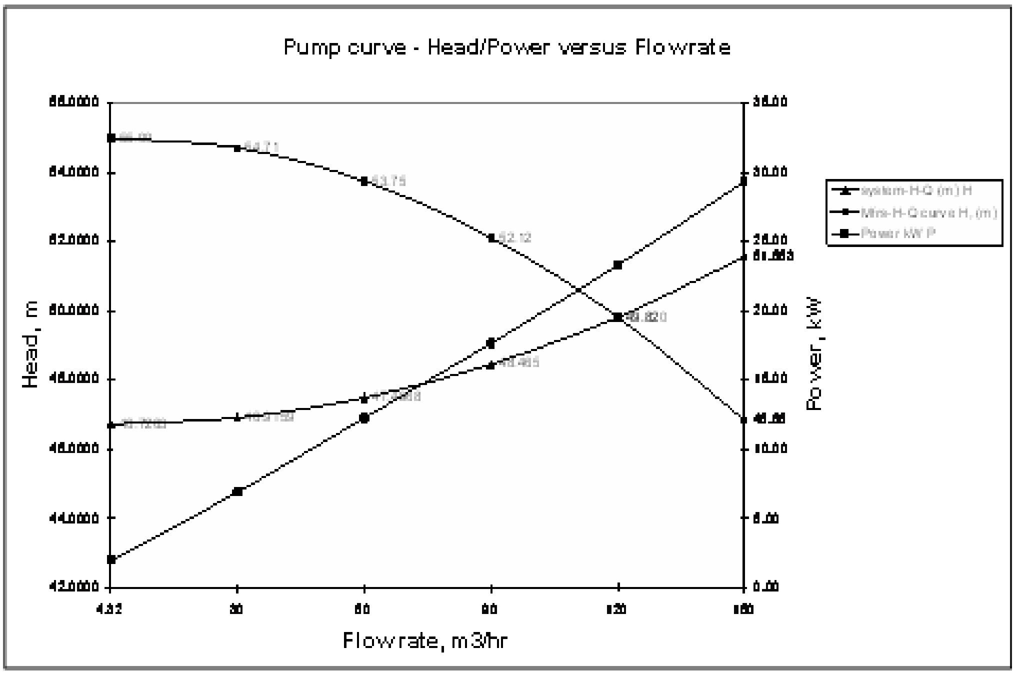


Fig. 11: Performance Curves of Head and Power *versus* Flow rate for Example
Source: [ALIGNAgraphics Pumpro Program] **[40]**

XIV. CONCLUSION

The limitation set on the described program is related to the lower and upper limits of the flow-rate as defined by the Corripio *et al.* [31] efficiency relationship. With correlations for extended applications, the program can be easily upgraded to handle larger flow-rates.

It can be argued that with pump manufacturers providing pump curves as electronic database documents backed with selection software, why an in-house program by end-users? The simple answer to that: with the exception of a few big manufacturers, most pump manufacturers for varied reasons, inclusive of costs, tend to operate in segmented markets. Thus, end users tend to resort to engineered consulting to handle the tedium and routine of matching a suitable pump for a pumping process need especially in times of retrofitting and process plant expansion requirements. Thus, specialised in-house programs can tend to bridge these gaps. Certain off the shelf packages also attempt to fill these needs.

ACKNOWLEDGMENT

This article is a disclosure of the contents and development features of the centrifugal pump hydraulic analysis and selection software program, "PUMPRO", developed by the now defunct ALIGNAgraphics Company, UK. The author thanks the Greater London Small Business Enterprise Initiative for the support granted the Company in its fledgling days. The author also thanks, the American Society of Mechanical Engineers (ASME) for the free of costs product feature placements in its Computers in Engineering pages, in the ASME Mechanical Engineering Magazine, and follow up leads from the wide global community of Engineers from a broad range of pump-user industries.

NOMENCLATURE

A = Pipe cross-sectional area [m^2]
D = Pipe internal diameter [m]
E_F = Total energy content of liquid [J/kg]
$E_{F(i)}$ = Total energy content of liquid at a location (i) [J/kg]
f = Friction factor
g = Gravity constant [m/s^2]
H_f = Head loss ≡ pressure loss [J/kg]
H_T = Total developed head required by pump [m]
H = Total developed process system operating head of pump, [m] = H_T
$L_{e(i,\, j)}$ = Equivalent length of pipe, valves and fittings between two locations (i and j) [m]
NPSH= Net Positive Suction Head [m]
P= Absolute pressure of liquid [Pa]
P_i = Absolute pressure of liquid at a location, (i), [Pa]
P_s = Static pressure [Pa]
P_v = Vapor pressure [Pa]
$\Delta P_{f(i,\, j)}$ = Pressure loss due to friction between two locations (i and j) in the system [Pa]
P_w = Pumping power [W]
Q = Flow rate [m^3/s]
u = Specific volume of liquid [m^3/kg]
u_i = Specific volume of liquid at a location (i) [m^3/kg]
$\bar{u}$ = *Average* liquid specific volume between two locations (i and j) in the system [m^3/kg]
V = Liquid flow velocity [m/s]
V_i = Liquid flow velocity at location (i) [m/s]
Z = Elevation or height of liquid [m]
Z_i = Elevation or liquid height at a location (i) [m]

Greek Letters

α, β = Corresponding Flow rate –to-Head points (Q-H) respectively from Pump Manufacturer's charts (obtained from Pump Catalogue) and flow hydraulics analysis System resistance curve.
Δ = Differential or difference
ε = Pipe roughness [mm]
η_p= Pump Efficiency [%]
μ= Liquid viscosity [N.s/m^2]
ρ= Liquid density [kg/m^3]
Ψ= Denotes a functional

Subscript

D = Discharge
S = Suction
1 = Reservoir or liquid source tank location
2 = Pump inlet location
3 = Pump outlet location
4 = Delivery tank location

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APPENDIX – Microsoft Excel VBA Centrifugal Pump Selection Program Sub-routines

```
Option Explicit
Dim T As Single, rho As Single, visc As Single, Pvp As Single, Q As Single
Dim L1 As Single, L2 As Single, Ds As Single, Dd As Single, Vs As Single, Vd As Single
Dim Vr As Single, Les As Single, Led As Single, Le1 As Single, Le2 As Single
Dim Re1 As Single, Re2 As Single, fs As Single, fd As Single, A1 As Single, A2 As Single
Dim e1 As Single, e2 As Single, P1 As Single, P2 As Single, hf1 As Single, hf2 As Single
Dim Power As Single, N As Single, Peff As Single, Hp As Single, Pstatic As Single
Dim Ef1 As Single, Ef2 As Single, Ef3 As Single, Ef4 As Single, Einput As Single
Dim Zr As Single, Zs As Single, Zd As Single, NPSHA As Single, sngRetVal1 As Variant
Dim intCount As Integer
Const conBtns = vbOKOnly + vbExclamation + vbDefaultButton1 + vbApplicationModal
Const conMsg1 = "the limiting flowrate is 0.072<=Q<=19.2 cu.m/min"
Const g = 9.81, Pi = 3.14159

'perform calculations

Sub Calculate()
  For intCount = 1 To 3 Step 1
  visc = ActiveSheet.Range("B6")
  rho = ActiveSheet.Range("B7")
  Pvp = ActiveSheet.Range("B9")
  Q = ActiveSheet.Range("B10")
  Ds = ActiveSheet.Range("E14")
  L1 = ActiveSheet.Range("E15")
  Les = Sheets("pumpbuttonCalcsheet").Range("G15")
  P1 = ActiveSheet.Range("E17")
  Zs = ActiveSheet.Range("E18")
  Dd = ActiveSheet.Range("G14")
  L2 = ActiveSheet.Range("G15")
  Led = Sheets("pumpbuttonCalcsheet").Range("I15")
  P2 = ActiveSheet.Range("G17")
  Zd = ActiveSheet.Range("G18")
'suction condition calculations
    A1 = Pi * (((Ds / 1000) ^ 2) / 4)
    Vs = (Q / 60) / A1
    Re1 = (rho * Vs * (Ds / 1000)) / visc
    ActiveSheet.Range("E20") = Re1
    Range("K21").GoalSeek goal:=0, changingcell:=Range("K17")
    fs = ActiveSheet.Range("K23")
    Le1 = (L1 + Les)
    hf1 = fs * (Le1 / (Ds / 1000)) * ((Vs ^ 2) / (2 * g))
    Zr = 0
    Vr = 0
    Ef1 = ((P1 * 1000) / rho) + (Zr * 9.81) + ((Vr ^ 2) / 2)
    Ef2 = Ef1 - hf1

  'discharge condition calculations

    A2 = Pi * (((Dd / 1000) ^ 2) / 4)
    Vd = (Q / 60) / A2
    Re2 = (rho * Vd * (Dd / 1000)) / visc
    ActiveSheet.Range("G20") = Re2
    ActiveSheet.Range("L21").GoalSeek goal:=0, changingcell:=Range("L17")
    fd = ActiveSheet.Range("L23")
    Le2 = (L2 + Led)
    hf2 = fd * (Le2 / (Dd / 1000)) * ((Vd ^ 2) / (2 * g))
    Ef3 = ((P2 * 1000) / rho) + (Zd * 9.81) + ((Vd ^ 2) / 2)
```

```
        Ef4 = Ef3 + hf2
        Einput = Ef4 - Ef2
        Hp = Einput / g
        Pstatic = (Ef2 * rho) - (Zs * g * rho) - (((Vs ^ 2) / 2) * rho)
        NPSHA = (Pstatic - Pvp) / (rho * g)

    If 0.072 <= Q And Q <= 19.2 Then
        Peff = 100 * (0.885 + (0.00824 * Log(Q / 60)) - (0.01199 * ((Log(Q / 60)) ^ 2)))
    Else
        sngRetVal1 = MsgBox(conMsg1, conBtns, "ALIGNAgraphics flowrate range")

    End If

    Power = ((rho * (Q / 60) * Hp * g) / (Peff / 100)) / 1000
'assign values to cells
    ActiveSheet.Range("E19") = Vs
    ActiveSheet.Range("G19") = Vd
    ActiveSheet.Range("E21") = fs
    ActiveSheet.Range("G21") = fd
    ActiveSheet.Range("A24") = Sheets("pumpbuttonCalcsheet").Range("D5")
    ActiveSheet.Range("A25") = Sheets("pumpbuttonCalcsheet").Range("D6")
    ActiveSheet.Range("A26") = Sheets("pumpbuttonCalcsheet").Range("D7")
    ActiveSheet.Range("A27") = Sheets("pumpbuttonCalcsheet").Range("D8")
    ActiveSheet.Range("A28") = Sheets("pumpbuttonCalcsheet").Range("D9")
    ActiveSheet.Range("A29") = Sheets("pumpbuttonCalcsheet").Range("D10")
    ActiveSheet.Range("A30") = Sheets("pumpbuttonCalcsheet").Range("D11")
    ActiveSheet.Range("A31") = Sheets("pumpbuttonCalcsheet").Range("D12")
    ActiveSheet.Range("A32") = Sheets("pumpbuttonCalcsheet").Range("D13")
    ActiveSheet.Range("A33") = Sheets("pumpbuttonCalcsheet").Range("D14")
    ActiveSheet.Range("C24") = Sheets("pumpbuttonCalcsheet").Range("E5")
    ActiveSheet.Range("C25") = Sheets("pumpbuttonCalcsheet").Range("E6")
    ActiveSheet.Range("C26") = Sheets("pumpbuttonCalcsheet").Range("E7")
    ActiveSheet.Range("C27") = Sheets("pumpbuttonCalcsheet").Range("E8")
    ActiveSheet.Range("C28") = Sheets("pumpbuttonCalcsheet").Range("E9")
    ActiveSheet.Range("C29") = Sheets("pumpbuttonCalcsheet").Range("E10")
    ActiveSheet.Range("C30") = Sheets("pumpbuttonCalcsheet").Range("E11")
    ActiveSheet.Range("C31") = Sheets("pumpbuttonCalcsheet").Range("E12")
    ActiveSheet.Range("C32") = Sheets("pumpbuttonCalcsheet").Range("E13")
    ActiveSheet.Range("C33") = Sheets("pumpbuttonCalcsheet").Range("E14")
    ActiveSheet.Range("E16") = Sheets("pumpbuttonCalcsheet").Range("E2")
    ActiveSheet.Range("G16") = Sheets("pumpbuttonCalcsheet").Range("E3")
    ActiveSheet.Range("E24") = Sheets("pumpbuttonCalcsheet").Range("G5")
    ActiveSheet.Range("E25") = Sheets("pumpbuttonCalcsheet").Range("G6")
    ActiveSheet.Range("E26") = Sheets("pumpbuttonCalcsheet").Range("G7")
    ActiveSheet.Range("E27") = Sheets("pumpbuttonCalcsheet").Range("G8")
    ActiveSheet.Range("E28") = Sheets("pumpbuttonCalcsheet").Range("G9")
    ActiveSheet.Range("E29") = Sheets("pumpbuttonCalcsheet").Range("G10")
    ActiveSheet.Range("E30") = Sheets("pumpbuttonCalcsheet").Range("G11")
    ActiveSheet.Range("E31") = Sheets("pumpbuttonCalcsheet").Range("G12")
    ActiveSheet.Range("E32") = Sheets("pumpbuttonCalcsheet").Range("G13")
    ActiveSheet.Range("E33") = Sheets("pumpbuttonCalcsheet").Range("G14")
    ActiveSheet.Range("E34") = Sheets("pumpbuttonCalcsheet").Range("G15")
    ActiveSheet.Range("G24") = Sheets("pumpbuttonCalcsheet").Range("I5")
    ActiveSheet.Range("G25") = Sheets("pumpbuttonCalcsheet").Range("I6")
    ActiveSheet.Range("G26") = Sheets("pumpbuttonCalcsheet").Range("I7")
    ActiveSheet.Range("G27") = Sheets("pumpbuttonCalcsheet").Range("I8")
    ActiveSheet.Range("G28") = Sheets("pumpbuttonCalcsheet").Range("I9")
    ActiveSheet.Range("G29") = Sheets("pumpbuttonCalcsheet").Range("I10")
    ActiveSheet.Range("G30") = Sheets("pumpbuttonCalcsheet").Range("I11")
    ActiveSheet.Range("G31") = Sheets("pumpbuttonCalcsheet").Range("I12")
    ActiveSheet.Range("G32") = Sheets("pumpbuttonCalcsheet").Range("I13")
    ActiveSheet.Range("G33") = Sheets("pumpbuttonCalcsheet").Range("I14")
    ActiveSheet.Range("G34") = Sheets("pumpbuttonCalcsheet").Range("I15")
    ActiveSheet.Range("E37") = Le1
    ActiveSheet.Range("G38") = Le2
    ActiveSheet.Range("E39") = hf1
```

```
    ActiveSheet.Range("G39") = hf2
    ActiveSheet.Range("E41") = Ef1
    ActiveSheet.Range("E42") = Ef2
    ActiveSheet.Range("G43") = Ef3
    ActiveSheet.Range("G44") = Ef4
    ActiveSheet.Range("D46") = Einput
    ActiveSheet.Range("D48") = Hp
    ActiveSheet.Range("D49") = Pstatic
    ActiveSheet.Range("D50") = NPSHA
    ActiveSheet.Range("B52") = Peff
    ActiveSheet.Range("B54") = Power
    ActiveSheet.Range("D54") = "Not for Resale"
    Next intCount
End Sub


Sub Clear()
    ActiveSheet.Range("D2:E2").ClearContents
    ActiveSheet.Range("D3:E3").ClearContents
    ActiveSheet.Range("E4").ClearContents
    ActiveSheet.Range("E5").ClearContents
    ActiveSheet.Range("G6:G7").ClearContents
    ActiveSheet.Range("G9:G10").ClearContents
    ActiveSheet.Range("B4:C4").ClearContents
    ActiveSheet.Range("B5:B7").ClearContents
    ActiveSheet.Range("B9:B10").ClearContents
    ActiveSheet.Range("E14:E21").ClearContents
    ActiveSheet.Range("G14:G21").ClearContents
    ActiveSheet.Range("C24:C33").ClearContents
    ActiveSheet.Range("D24:D33").ClearContents
    ActiveSheet.Range("E24:E34").ClearContents
    ActiveSheet.Range("F24:F33").ClearContents
    ActiveSheet.Range("G24:G34").ClearContents
    ActiveSheet.Range("E37").ClearContents
    ActiveSheet.Range("E39").ClearContents
    ActiveSheet.Range("G38:G39").ClearContents
    ActiveSheet.Range("E41:E42").ClearContents
    ActiveSheet.Range("G43:G44").ClearContents
    ActiveSheet.Range("D46").ClearContents
    ActiveSheet.Range("D48:D50").ClearContents
    ActiveSheet.Range("B52").ClearContents
    ActiveSheet.Range("B54").ClearContents
    ActiveSheet.Range("G49:G50").ClearContents
    ActiveSheet.Range("G52").ClearContents
    ActiveSheet.Range("G54").ClearContents
    ActiveSheet.Range("K28").Value = 1
    ActiveSheet.Range("N15:N24").Value = 1
    ActiveSheet.Range("A24:A33").ClearContents
    Sheets("pumpbuttonCalcsheet").Range("C2:C3").Value = 1
    ActiveSheet.Range("D54").ClearContents

End Sub
```