4.0 SELECTION OF PUMP TYPE AND PUMP

4.1 Classification of pumps

There are two main types of pumps used in the pumping of liquids, including water, namely:

1) Positive displacement pumps;
2) Rotodynamic pumps.

4.1.1 Positive displacement pumps

These types of pumps are built as reciprocating, or as rotary machines.

a) Reciprocating types

The main designs are:

1) Piston or plunger types such as those used in piston-type water pumps for shallow wells, and plunger types used in fuel injection pumps for internal combustion engines;

2) Membrane type pumps such as those used in fuel lift pumps in petrol engines or as chemical dosing pumps for water treatment works.

b) Rotary positive displacement pumps

The main designs are:

1) Gear wheel pumps commonly used in lubricating oil pumping in internal combustion engines;
2) Worm pumps;
3) Partition or vane pumps;
4) Mono pumps commonly used in the pumping mixtures of water and solids such as sewage sludge.

The primary character of positive displacement pumps is that an element of the pump displaces a fixed volume of fluid. Increasing the fluid compartment to suck in fluid, and thereafter reducing the fluid compartment to expel the fluid, achieves this positive displacement function.

Positive displacement pumps are capable of generating high pressures, limited only by the regulating or safety features installed on the delivery or discharge pipe. They are however only capable of low discharge flows, limited by the volume swept by the displacement element, and the speed of the element.

Positive displacement pumps are more complex mechanically, and therefore tend to be more expensive. For this reason, they are a second choice when Rotodynamic machines can accomplish the job.

4.1.2 Rotodynamic Pumps

Rotodynamic pumps are types of pumps where mechanical energy is first applied to an impeller in the form of rotary motion. This kinetic energy of a rotating wheel and blade is then transferred to the fluid to be pumped as fluid velocity. Thereafter, the kinetic energy represented by the fluid velocity is reconverted into fluid pressure by the stationary parts of the pump machine.

There are three types of rotodynamic pumps. The impeller shapes of the three types of pumps are shown in **Figures 1,2 & 3.** The types of pumps are:

1) Axial flow;
2) Mixed radial and axial flow.
3) Centrifugal or radial flow;

The fluid flow pattern through the three types of machines is also shown in **Figures 1,2 & 3**. The flow pattern is:

1) Centrifugal or **radial** flow pumps are those in which the fluid leaves the impeller in a radial direction.

2) Propeller or **axial** flow pumps are those in which the fluid leaves the impeller in an axial direction.

3) **Mixed flow** pumps are those in which the fluid leaves the impeller with a combination of radial and axial velocity.

The **performance range** for the pump types shown in **Figures 1,2 & 3** may be summarised approximately as below:

1) **Axial flow pumps**:

   a) Pressure generated Up to 10 mWH;
   b) Discharge flow rate Up to 20 cubic metres per second;

2) **Mixed flow pumps**:

   a) Pressure generated Up to 30 mWH;
   b) Discharge flow rate Up to 8 cubic metres per second;

3) **Centrifugal or radial flow pumps**:

   a) Pressure generated Up to 100 mWH;
   b) Discharge flow rate Up to 6 cubic metres per second.

4.1.3 Other Types of Pumps

Other types of pumps are:

1) **Positive displacement types**

   These are characterised by low discharge flow rates but can attain high pressures. Pressure relief valves that are usually fitted to the delivery side of the pump to provide safety protection against high pressures also limit maximum pressures possible.

   However, when built into multi-stage pumps, centrifugal or mixed flow pumps are also capable of delivering high pressures which would otherwise require positive displacement pumps.

2) **PUMPS with Free water surface**

   These are capable of low lifts only. Examples are:

   a) Water Wheels;
   b) Archimedean screws

4.2 PERFORMANCE OF ROTO-DYNAMIC PUMPS

Fluid flow in a Roto-dynamic machine (Radial or Axial flow)

The flow of fluid in a radial or axial flow machine is generated by the rotation of the pump's impeller blades. This rotation transfers kinetic energy from the impeller blade to the body of fluid inside the pump.

Figure 4 shows the case of the axial flow machine, while **Figure 5** shows the case of the radial flow machine.

Velocities at inlet and outlet of impeller blade

The velocities are illustrated at the impeller inlet and outlet locations shown in Figure 4(axial) and 5 (radial).

In the case of the axial flow machine, fluid flows through the machine at various radii, and the velocities shown therefore refer to a particular radius of the machine, and will vary in magnitude according to the radius location (r) considered.

In the case of the radial flow machine, fluid flows from the small impeller inlet radius to the large outlet radius.

In both cases (Axial and Radial), the inlet velocities are designated 1, while the outlet velocities are designated 2.

The inlet velocities and associated variables are:

β_1	Impeller blade angle at inlet
u_1	Tangential velocity of the impeller blade at inlet
v_1	Absolute velocity of fluid at impeller blade inlet
v_{1q}	Component of fluid velocity normal to inlet area
ω_1	Relative velocity of fluid in the direction of the impeller blade at inlet

In the velocities shown, the flow conditions in the machine are presumed to be at the point of maximum efficiency. Consequently, the fluid enters the inlet blades without shock, (with the relative velocity of fluid coinciding with the blade angle).

Under such conditions, the absolute velocity of fluid at impeller blade inlet (v_1) is normal to the inlet area. This absolute velocity of fluid is therefore equal to the component of fluid velocity normal to inlet area (v_{1q}), and the component of fluid velocity in the tangential direction is therefore zero at inlet ($v_{1u} = 0$).

The outlet velocities and associated variables are:

β_2 Impeller blade angle at outlet
u_2 Tangential velocity of the impeller blade at outlet
v_2 Absolute velocity of fluid at impeller blade outlet
v_{2q} Component of fluid velocity normal to outlet area
ω_2 Relative velocity of fluid in the direction of the impeller blade at outlet

Velocity triangles

From the velocities shown in the machines, velocity triangles are drawn in **Figures 4 and 5**, for an axial and a radial machine respectively.

For the axial flow machine, the tangential velocity of the impeller blade is the same at both inlet and outlet of the blade. This is due to the common radius at which both velocities are taken.

In the radial flow machine, the radius at inlet is smaller than that at outlet, and consequently, the tangential velocities u_2 is greater than u_1 .

Euler equation for pressure head generated

According to the Euler equation, the fluid pressure head generated by the machine with an **infinite number of blades** is given by the expression:

$$gH_{th} = v_{2u}u_2 - v_{1u}u_1$$

Where,

$H_{th} = The\ pressure\ head\ generated\ by\ the\ machine,\ assu\min g\ \inf inite\ number\ of\ blades$
$v_{2u} = Component\ of\ fluid\ velocity\ in\ the\ \tan gential\ direction\ at\ outlet$
$v_{1u} = Component\ of\ fluid\ velocity\ in\ the\ \tan gential\ direction\ at\ inlet$

But when the machines are considered at their maximum efficiency points, the component of fluid velocity in the tangential direction is zero at inlet ($v_{1u} = 0$). The resulting equation for pressure head generated then becomes:

$$gH_{th} = v_{2u}u_2$$

Head Vs Flow Performance Curve for Radial Flow Pump

For a particular machine, the velocities used in the Euler head can be expressed in terms of size, and rotational speed of the impeller blades.

Velocities at impeller blade exit

The tangential component of fluid velocity at impeller blade exit is a function of the factors shown in the exit velocity triangle at Figure 4 and 5 .

Radial component and relative velocity of fluid velocity at exit

Radial component of fluid velocity at impeller blade exit is given by

$$v_{2q} = \frac{Q(\text{fluid flow in } m^3/s)}{\text{Area of flow at impeller blade exit}}$$

$$v_{2q} = \frac{Q}{\pi DB}$$

Where

$B = \text{width of impeller blade at exit as shown in Figure 1 and 2}$

But from the velocity triangle, the radial component of fluid velocity is also given by

$$v_{2q} = \omega_2 \sin\beta_2$$

$$v_{2q} = \frac{Q}{\pi DB} = \omega_2 \sin\beta_2, \text{ and } \omega_2 = \frac{Q}{\pi DB \sin\beta_2}$$

Tangential component of fluid velocity at impeller blade exit

From the velocity triangle at impeller blade exit of the radial flow machine, the tangential component is given by

$$v_{2u} = u_2 - \omega_2 \cos\beta_2$$

Substituting for ω_2 *in the above equation*

$$v_{2u} = u_2 - \frac{Q}{\pi DB \sin\beta_2}\cos\beta_2$$

$$v_{2u} = u_2 - \frac{Q}{\pi DB \tan\beta_2}$$

Tangential velocity of impeller blade at exit

This is given by:

$$u_2 = \frac{\pi DN}{60} \ m/s$$

Where,

$D = \text{Outlet diameter of impeller blade}$

$N = \text{Rotational speed of impeller wheel in revs/minute}$

Euler head generated

This is given by

$$gH_{th} = v_{2u}u_2$$

Substituting for v_{2u} *and* u_2

$$gH_{th} = \left[u_2 - \frac{Q}{\pi DB\tan\beta_2}\right]u_2 = u_2{}^2 - u_2 * \frac{Q}{\pi DB\tan\beta_2}$$

$$gH_{th} = \left[u_2{}^2 - \frac{Q}{\pi DB\tan\beta_2} * u_2\right]$$

$$gH_{th} = \left[\left(\frac{\pi DN}{60}\right)^2 - \frac{Q}{\pi DB\tan\beta_2} * \frac{\pi DN}{60}\right]$$

Substituting

$$K_1 = \left(\frac{\pi DN}{60}\right)^2 \quad and \quad K_2 = \frac{}{B\tan\beta_2} * \frac{N}{60}$$

$$\underline{gH_{th} = K_1 - K_2Q}$$

Performance curve for Head (H) versus Flow (Q)

The performance curve of the machine as given by the Euler equation is therefore a straight line with a negative gradient, as shown in **Figure 6**.

This may also be expressed as:

$$H_{th} = \frac{u_2{}^2}{g} - u_2 * \frac{Q}{g\pi DB\tan\beta_2}$$

$$H_{th} = \frac{u_2{}^2}{g} - K_2 * Q$$

The curve H(t), shown in **Figure 7** is the theoretical performance curve for a real machine with a finite number of impeller blades.

4.2.5 Characteristic Head versus Discharge performance of the radial flow machine

The final pressure head generated by a **real radial flow impeller** will be further reduced by various losses shown in **Figure 7**, which occur in a the machine. These are:

a) Hydraulic friction losses occurring as fluid flows through the impeller and the pump's volute. Such losses increase with the velocity of flow, and therefore the discharge flow rate Q.

b) Entry losses at impeller blade inlet are minimum at a particular value of discharge flow rate Q and hence at a particular inlet velocity. At this one flow rate, the inlet velocity w_1 coincides with the blade angle at inlet. At lower or higher discharge flow rates different from this point of minimum entry losses, the fluid velocity, relative to the impeller blade, does not coincide with the blade angle at inlet, and losses caused by shock at entry increase. The discharge with minimum shock losses at entry, is the design discharge.

c) The third category of losses is caused by leakage of fluid from the high-pressure side to the low-pressure side. This leakage fluid is either returned to the inlet or lost without reaching discharge, for example, at the shaft seals.

The pressure head remaining after the losses for the real machine is taken into account is then the lowest curve in Figure 7 which shows **Total head** versus **Discharge** $(Q-H)$ performance of the pump.

4.2.6 Characteristic Power versus Discharge performance of the radial flow machine

Starting with the Head versus Discharge $(Q-H)$ curve shown at Figure 7, the Power versus Discharge performance curve $(Q-P)$, can be derived for the radial flow pump. The typical result is shown at **Figure 8**.

In Figure 8, the lowest curve shows how the fluid horsepower, computed as $\rho * g * QH$ varies with the discharge flow Q. This curve is obtained by computation from the head versus discharge $(Q-H)$ curve in Figure 7.

The fluid horsepower $\rho * g * QH$ is the net output power from the hydraulic machine. It excludes hydraulic and mechanical losses in the machine. The hydraulic losses, already identified as resulting in reduced head, also represent power losses in the machine. These power losses, also shown in 8, and are:

1) Hydraulic losses due to fluid friction within the impeller, and in the volute;
2) Entry losses due to shock at blade inlet;
3) Leakage losses occurring because pressurised fluid leaks back to inlet or out of the machine without reaching pump discharge.

Additional to these power losses due to the hydraulic efficiency of the machine, there are other mechanical losses which are not shown in Figure 8 such as:

1) Mechanical losses due to disc friction as the impeller wheel rotates in the fluid medium;
2) Mechanical losses at shaft bearings and at seals.

When these additional mechanical losses are added to the hydraulic losses, and to the fluid horsepower $(\rho * g * Q * H)$, the input brake horsepower (BHP), required at pump shaft is obtained. The resulting variation of brake horsepower (BHP) with the discharge flow (Q) is still similar to the lowest curve shown in Figure 8. This is the typical Power versus Discharge $(Q-P)$ characteristic performance curve for the radial flow pump. The curve then represents the power input required into the pump shaft, to generate the desired water horsepower $(\rho * g * Q * H)$.

4.2.7 Characteristic Efficiency versus Discharge performance of the radial flow machine

At each discharge value, the radial flow pump's overall efficiency is the ratio of the fluid horsepower $(\rho * g * Q * H)$ to the brake horsepower (BHP). This can be obtained by the computation of the fluid horsepower $(\rho * g * Q * H)$, followed by the measurement of the brake horsepower (BHP).

The resulting variation of efficiency with discharge flow $(Q - Eta)$, typical of the radial flow pump, is shown in **Figure 9**.

4.3 Design discharge

From the typical efficiency curve of the radial flow pump shown in Figure 9, it is seen that efficiency increases with discharge from zero to a point of maximum efficiency, and thereafter declines with increasing discharge. The discharge corresponding to the maximum efficiency, is the design discharge for the pump, and the point where the machine should be operated, when possible. At this design point, the power consumption per unit of discharge is minimum.

4.4. Summary characteristics for Radial flow pump

1) The Discharge versus Head $(Q-H)$ relationship shows a flat curve. The discharge varies rapidly with change of head.

2) The power requirement decreases with decreasing discharge, and increases with increasing discharge. Consequently, the power is minimum at minimum discharge Large models of this pump are therefore started intentionally with closed outlet valves or zero discharge to keep the starting power requirements at a minimum.

3) The pump housing is volute shaped to convert the kinetic energy of the fluid into pressure energy.

4.4.1 Summary characteristics for Axial flow pumps

1) The Discharge versus Head $(Q-H)$ relationship shows a steep curve. With an increasing head, the discharge decreases rather slowly. The power requirements also increase, and the efficiency decreases.

2) When head is decreased, the discharge increases while the power requirements and efficiency both decreases.

3) At zero discharge, which can occur when the pump is operated against a closed discharge valve, the power requirement is much greater than at design discharge. This type of pump should therefore never be started when the discharge valve is closed or throttled.

4) This type of pump has a guide vane fitted downstream of the impeller, to convert the kinetic energy of the fluid into pressure energy.

4.5 SPECIFIC SPEED

Specific speed is a parameter, which relates the performance of the rotodynamic machine at its peak efficiency, to the shape of the impeller wheel. This shape of the impeller wheel indicates the type of machine required. The parameter is applied to both pumps and turbines.

For a pump, the relevant performance criteria are discharge flow rate, and the total head to be overcome. A secondary criterion is the operating speed at which the machine is to be driven.

The specific speed of a Rotodynamic machine is then given by the expression:

$$n_s = \frac{N\sqrt{Q}}{H^{3/4}} = f(r/R)$$

Where,

$n_s = Specific\ speed\ in\ metric\ units$
$N = Rotational\ speed\ of\ the\ machine\ in\ revs/\min$
$Q = Flow\ discharge\ at\ peak\ efficiency\ in\ m^3/\sec$
$H = Total\ head\ generated\ at\ peak\ efficiency\ in\ metres\ of\ water$
$r = Inlet\ diameter\ of\ pump\ impeller$
$R = Outlet\ or\ wheel\ diameter\ of\ pump\ impeller$

The value of the group $\left[\frac{N\sqrt{Q}}{H^{3/4}}\right]$, the specific speed, is therefore an indicator of the shape of the impeller wheel, because it is a function of the ratio (r/R) of inlet diameter to outlet diameter of impeller wheel.

The value of the group $\left[\frac{N\sqrt{Q}}{H^{3/4}}\right]$ can therefore be used to predict the shape of impeller wheel required and hence the type of pump required.

The parameter of specific speed is useful because it makes it possible to compare, in detail, the performance of equally shaped impellers of different sizes.

From the relationship, it is seen that a large value of specific speed $\left[\frac{N\sqrt{Q}}{H^{3/4}}\right]$, implies a large value of the ratio (r/R) and vice versa. Furthermore, a large value of the ratio (r/R) implies an axial flow pump, while a small value of the ratio (r/R) implies a centrifugal or radial flow pump.

The variation of impeller shape or pump types with specific speed is shown qualitatively at **Figure 11**.

4.5.1 Specific speed and performance characteristic curves

Typical characteristic performance curves for radial types of pumps are shown in **Figures 10.** The point of maximum efficiency is taken as design point reflecting 100 % discharge, 100 % head, 100 % power, and 100 % efficiency.

Since the specific speed determines the pump type, it also determines the shape of the performance characteristic curves of the pump. **Figure 10** shows characteristic performance curves for various specific speeds or pump types. The curves are drawn with the design point of each pump representing 100% performance (i.e. Q=100 %, H=100%, P=100%, at the design point). The performance curves shown in Figure 10 represents machines ranging from a specific speed of $ns = 10\ \ to\ \ ns = 80$.

4.5.2 Variation of peak efficiency with pump type and size

Figure 12 shows the variation of peak efficiency with pump type and size. Peak efficiency increases with specific speed up to approximately 80 metric units, and thereafter declines with further increases in specific speed.

The reduction of peak efficiency is caused by increased losses due to fluid friction when the pump shape is very radial. For this reason, radial flow machines with a specific speed less than 10 metric units are not recommended.

Secondly, peak efficiency increases as the discharge flow through the pump increases, implying that larger pumps are more efficient even when they are of the same specific speed, and therefore of the same pump type.

Figure 12 shows the variation of efficiency with specific speed as well as with the size of pump, (the well-known Worthington plot).

Best efficiencies are usually obtained from pumps with specific speed between 29 and 58 metric units

From the variation of efficiency with the size of pump, denoted by flow capacity, it can be seen that for the same specific speed, the larger pump is more efficient.

4.6 Specific speed and choice of Pump Type

Specific speed is a useful concept for the comparison of pump types, in terms of their optimum performance. Specific speed is a function of a pump's discharge capacity, total head, and rotational speed at peak efficiency, such that for a given pump and impeller geometry/shape, specific speed remains constant over a range of capacities and heads.

Specific speed is used as a guide in selecting the most efficient pump type. Given a desired flow rate, total head to be overcome, and operating speed, specific speed can be computed, and the type of impeller chosen. A guide to this classification and selection of pump types is shown in **Figure 13.** The number *ns* obtained as specific speed indicates the impeller geometry or shape that will satisfy the specified requirements at the highest efficiency. The data from which the graphs at Figure 13 are drawn are shown in **Tables 1 and 2**.

4.6.1 Selecting a Centrifugal pump

At low specific speeds, the highest head per stage is developed. However, for best efficiency, a centrifugal pump's specific speed should be greater than 12.5, and less than 96 metric units. At low specific speeds, the impeller diameter is large, with high mechanical friction and high hydraulic losses. If the specific speed computed for a given set of design specifications drops below 12.5 metric units, a multiple stage pump should be selected.

4.6.2 Multiple stage Pumps

A multiple stage pump consists of two or more impellers within the same casing, such that the discharge of the first stage impeller becomes the suction of the second stage. In this manner, higher heads are achieved than would be possible with a single stage impeller. Both centrifugal and mixed flow pumps are built as multiple stage pumps

4.6.3 Double suction pumps

Radial flow and mixed flow pumps may be designed as either single or double suction. In single suction pumps, the liquid enters only one side of the impeller. In double suction, the flow enters each side of the impeller. Thus for an impeller of a given specific speed, a greater flow rate can be handled by a double suction pump. In addition, a double suction pump requires a lower NPSHR for a given flow than does a single suction pump.

Single suction centrifugal pumps have a specific speed of less than 97 metric units. Double suction centrifugal impellers have a specific speed of less than 116 metric units.

4.6.4 Selecting a Mixed flow Pump

As the specific speed increases, the ratio of the impeller diameter to the inlet diameter decreases. As this ratio decreases, the pump develops less head (since the impeller diameter is decreasing), but greater flow. Thus as specific speed increases, the energy added by the impeller becomes more and more the result of axial forces of the rotating vanes, and less the result of centrifugal forces.

If the specific speed is greater than 38 metric units, a mixed flow impeller is indicated. Best efficiencies are usually obtained from pumps with specific speed between 29 and 58 metric units. This is illustrated in Figure 12.

Specific speeds of mixed flow pumps range from 81 to 175 metric units.

4.6.5 Axial flow pumps

At specific speeds of 195 metric units or higher, the pump is no longer centrifugal, nor mixed flow but axial; and suitable for high flow rates but low discharge heads. This is illustrated in Figure 11.

Specific speeds of axial flow impellers are greater than 175 metric units.

TABLE 1: GUIDE FOR PUMP TYPE REQUIRED IN A WATER PUMPING SYSTEM-PUMP OPERATING SPEED OF 1450 R.P.M.

PERFORMANCE REQUIREMENTS TO BE MET SPECIFIED AS DISCHARGE FLOW DESIRED AND PRESSURE HEAD TO BE OVERCOME

SPECIFIC SPEED FOR VARIOUS DISCHARGE FLOWS AND PRESSURE HEADS IS USED AS AN INDICATOR OF PUMP TYPE

FOR ROTODYNAMIC MACHINES, MAXIMUM EFFICIENCY OCCURS IN SPECIFIC SPEED RANGE 29-58 (METRIC UNITS)

SPECIFIC SPEED OF	***<12.5***	***METRIC UNITS***	***MULTI-STAGE (RADIAL OR MIXED) FLOW***
SPECIFIC SPEED OF	***12.5-96***	***METRIC UNITS***	***RADIAL (CENTRIFUGAL) FLOW TYPE***
SPECIFIC SPEED OF	***81-175***	***METRIC UNITS***	***MIXED FLOW TYPE***
SPECIFIC SPEED OF	***>175***	***METRIC UNITS***	***AXIAL FLOW TYPE***

	DISCHARGE FLOW DESIRED IN M^3/SEC													
	0.0001	**0.001**	**0.01**	**0.1**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
HEAD mwH	**POSITIVE DISPLACEMENT OR MULTI-STAGE TYPE PUMPS INDICATED**					**CENTRIFUGAL TYPE PUMPS INDICATED FOR PERFORMANCE IN THIS RANGE**								
200	0	1	3	9	27	39	47	55	61	67	72	77	82	86
190	0	1	3	9	28	40	49	57	63	69	75	80	85	90
180	0	1	3	9	30	42	51	59	66	72	78	83	89	93
170	0	1	3	10	31	44	53	62	69	75	81	87	92	97
160	0	1	3	10	32	46	56	64	72	79	85	91	97	102
150	0	1	3	11	34	48	59	68	76	83	90	96	101	107
140	0	1	4	11	36	50	62	71	80	87	94	101	107	113
130	0	1	4	12	38	53	65	75	84	92	100	107	113	119
120	0	1	4	13	40	57	69	80	89	98	106	113	120	126
110	0	1	4	13	43	60	74	85	95	105	113	121	128	135
100	0	1	5	15	46	65	79	92	103	112	121	130	138	145
90	0	2	5	16	50	70	86	99	111	122	131	140	149	157
80	1	2	5	17	54	77	94	108	121	133	143	153	163	171
70	1	2	6	19	60	85	104	120	134	147	159	169	180	189
60	1	2	7	21	67	95	116	135	150	165	178	190	202	213
50	1	2	8	24	77	109	134	154	172	189	204	218	231	244
40	1	3	9	29	91	129	158	182	204	223	241	258	273	288
30	1	4	11	36	113	160	196	226	253	277	299	320	339	358
20	2	5	15	48	153	217	266	307	343	376	406	434	460	485
10	3	8	26	82	258	365	447	516	577	632	682	729	774	815
5	4	14	43	137	434	613	751	867	970	1062	1147	1227	1301	1371
1	15	46	145	459	1450	2051	2511	2900	3242	3552	3836	4101	4350	4585
	CENTRIFUGAL		**MIXED**	**AXIAL FLOW TYPE PUMPS INDICATED FOR PERFORMANCE IN THIS RANGE**										

TABLE 2: GUIDE FOR PUMP TYPE REQUIRED IN A WATER PUMPING SYSTEM-PUMP OPERATING SPEED OF 2900 R.P.M.

PERFORMANCE REQUIREMENTS TO BE MET SPECIFIED AS DISCHARGE FLOW DESIRED AND PRESSURE HEAD TO BE OVERCOME

SPECIFIC SPEED FOR VARIOUS DISCHARGE FLOWS AND PRESSURE HEADS IS USED AS AN INDICATOR OF PUMP TYPE

FOR ROTODYNAMIC MACHINES, MAXIMUM EFFICIENCY OCCURS IN SPECIFIC SPEED RANGE 29-58 (METRIC UNITS)

SPECIFIC SPEED OF	*<12.5*	*METRIC UNITS*	*MULTI-STAGE (RADIAL OR MIXED) FLOW*
SPECIFIC SPEED OF	*12.5-96*	*METRIC UNITS*	*RADIAL (CENTRIFUGAL) FLOW TYPE*
SPECIFIC SPEED OF	*81-175*	*METRIC UNITS*	*MIXED FLOW TYPE*
SPECIFIC SPEED OF	*>175*	*METRIC UNITS*	*AXIAL FLOW TYPE*

	DESIRED DISCHARGE FLOW IN M^3/SEC													
	0.0001	0.001	0.01	0.1	1	2	3	4	5	6	7	8	9	10
HEAD mwH	MULTI-STAGE TYPE OR POSITIVE DISPLACEMENT			CENTRIFUGAL TYPE PUMPS				MIXED FLOW TYPE PUMPS INDICATED						
200	1	2	5	17	55	77	94	109	122	134	144	154	164	172
190	1	2	6	18	57	80	98	113	127	139	150	160	170	179
180	1	2	6	19	59	83	102	118	132	145	156	167	177	187
170	1	2	6	19	62	87	107	123	138	151	163	174	185	195
160	1	2	6	20	64	91	112	129	144	158	171	182	193	204
150	1	2	7	21	68	96	117	135	151	166	179	191	203	214
140	1	2	7	23	71	101	123	143	159	175	189	202	214	225
130	1	2	8	24	75	107	130	151	168	185	199	213	226	238
120	1	3	8	25	80	113	139	160	179	196	212	226	240	253
110	1	3	9	27	85	121	148	171	191	209	226	241	256	270
100	1	3	9	29	92	130	159	183	205	225	243	259	275	290
90	1	3	10	31	99	140	172	198	222	243	263	281	298	314
80	1	3	11	34	108	153	188	217	242	266	287	307	325	343
70	1	4	12	38	120	169	208	240	268	294	317	339	359	379
60	1	4	13	43	135	190	233	269	301	330	356	380	404	425
50	2	5	15	49	154	218	267	308	345	378	408	436	463	488
40	2	6	18	58	182	258	316	365	408	447	482	516	547	577
30	2	7	23	72	226		392	452	506	554	599	640	679	715
20	3	10	31	97	307	434	531	613	686	751	811	867	920	970
10	5	16	52	163	516	729	893	1031	1153	1263	1364	1459	1547	1631
5	9	27	87	274	867	1227	1502	1735	1939	2124	2295	2453	2602	2743
1	29	92	290	917	2900	4101	5023	5800	6485	7104	7673	8202	8700	9171
	CENTRIFUGAL		AXIAL FLOW TYPE PUMPS INDICATED FOR PERFORMANCE IN THIS RANGE											