Annexure 5.1: Design Calculations for Pumping Machinery

1.0 Data of Scheme

1.1	Daily demand of water	116 ML/day
1.2	Hours of pumping, considering loss of one hour due to tripping and other minor interruptions	23 hrs per day
1.3.1	Maximum (Top water level)	11.0 m
1.3.2	Mean WL	9.0 m
1.3.3	Minimum WL	7.0 m
1.4	Rising main	
1.4.1	Length	2,575 m
1.4.2	Diameter (Internal)	1,200 mm
1.4.3	H-W Coefficient for MS mortar lined pipeline	140
1.5	RL of point of discharge destination	59.0 m
1.6	No. of pumps (Based on the preliminary analysis)	
	1.6.1 Duty pumps	4
	1.6.2 Standby pumps	2
1.7	RL of ground level at the pumping station	8.25 m
1.8	RL of high flood level	10.5 m
1.9	Altitude of the site above mean sea level	1,250 m
1.10	Ambient temperature	40 °C

2.0 Size of pipes and fittings for the pumping system for Q = 0.35 m^3/s per pump

2.1	Inlet bell mouth for VT pump	
	Design velocity	1.4 m/s
	Bell mouth diameter	0.564 m
		Say 600 mm
	Actual velocity	1.24 m/s

2.2 Column pipes

Design velocity	1.75 to 2.75 m/s
	Say, 2.75 m/s (for higher flow)
Column pipe diameter	0.402 m
	Say 400 mm

Actual velocity 2.79 m/s

2.3 Delivery pipes and valves

Design velocity	2.0 m/s
Diameter of delivery pipe, delivery valve and NRV	0.472 m
	Say 500 mm
Actual velocity	1.78 m/s

2.4 Suction pipes and valves

Design velocity	1.5 m/s
Diameter of suction pipe and valve	0.545 m
	Say 600 mm
Actual velocity	1.24 m/s

2.5 Suction bell mouth for centrifugal pump

Design velocity	1.2 m/s
Diameter of bell mouth	0.609 m
	Say 700 mm
Actual velocity	0.91 m/s
2.6 Suction eccentric reducer 600 × 500 (assumed)	1.78 m/s

3.0 Pump head and required head range

3.1 Combined discharge of four pumps In parallel (116 ML/d × 24 hrs.)/23 hrs.	121.01 MLD
3.2 Rate of total flow with 23 hrs. running of pumps per day	1.4 m^3/s

3.3 Discharge of each pump 0.35 m^3/s

3.4 Mean static head (59 m - 9 m) 50 m

3.5 Frictional loss in straight pipeline for combined discharge 2.237 m

3.6 Minor losses in bends, valves, and other fittings and exit loss in rising main at 10% of (3.5) 0.2237 m

3.7 Station loss (head loss in suction, delivery piping, valves, manifold, etc., at pump house).

Table Showing head losses in piping and valves in the pumping station

S. No.	Fitting/Equipment	Size (mm)	Velocity (m/s)	K/C*	Head loss
A) Delivery side (VT/Centrifugal Pump)					
1	Enlarger	400 × 500	2.79	0.4	0.16
2	Tee for air valve	500 × 500 × 100	1.78	0.3	0.05
3	Non-return valve	500	1.78	2.5	0.40
4	Dismantling joint	500	1.78	0.3	0.05
5	Delivery valve (BFV)	500	1.78	0.4	0.06
6	Distance piece 1 m length	500	1.78	110	Negligible
7	Knife gate valve	500	1.78	0.3	0.05
8	30°/45° tee at manifold	500	1.78	0.8	0.13
9	Manifold (L = 14 m)	1,200	1.24	110	0.02
10	90° bend	1,200	1.24	0.75	0.06
11	NRV/DPCV	1,200	1.24	2.5	0.19
12	Dismantling joint/flange adopter	1,200	1.24	0.3	0.02
13	Isolation valve (BFV)	1,200	1.24	0.4	0.03
14	Flowmeter	1,200	1.24	0.1	0.007
				Total (A)	1.23 (rounded 1.3)
B) Suction side (centrifugal pump only)					
15	Bell mouth	700	0.91	0.1	0.004
16	90° bend	600	1.24	0.5	0.04
17	BFV	600	1.24	0.4	0.03

S. No.	Fitting/Equipment	Size (mm)	Velocity (m/s)	K/C*	Head loss
18	Suction piping/reducer	500 × 600	1.78	0.20	0.20 (assumed)
				Total (B)	0.28 (rounded 0.3)

*Head loss in valves and fittings = k x $\frac{v^2}{2g}$

Where k = Coefficient depending on valve/fitting

(Refer Chapter 6 Part A of Manual)

*H_f in pipes = $\frac{10.674}{D^{4.87}} \times \left(\frac{Q^{1.852}}{C}\right) \times L$

Where H_f = friction loss in straight pipeline

Q = discharge, m^3/s

L = Length of pipe

D = internal diameter, m

3.8 Design pump head for VT pump = (3.4) + (3.5) + (3.6) + Total A = 54.0 m (Rounded)

3.9 Design pump head for centrifugal pump = (3.4) + (3.5) + (3.6) + Total A + Total B

= 54.3 m (Rounded)

3.10 System head curve

Table showing co-ordinates of heads and discharge for system head curves at max WL, mean WL, and min WL (discharging RL at destination = 59.0 m).

Parameter		H_f	Total H at max WL (m)	Total H at mean WL (m)	Total H at min WL (m)
WL	-	-	11.0	9.0	7.0
Static head	-	-	48.0	50.0	52.0
Q(m^3/s)	0	-	48.0	50.0	52.0
	0.25	0.101	48.101	50.101	52.101
	0.50	0.365	48.365	50.365	52.365
	0.75	0.775	48.775	50.775	52.775
	1.00	1.314	49.314	51.314	53.314
	1.25	1.986	49.986	51.986	53.986
	1.50	2.783	50.783	52.783	54.783
	1.75	3.703	51.703	53.703	55.703

The system head curve is illustrated in Figure 5.1.A.1.

3.11 Head Range

As seen from the system head curve, actual head variations are from 48.0 m to 55.7 m which is -11.1% to +3.1%. However, as per guidelines and IS, the head range for the VT pump (which is finally selected) shall be +10% and -25%.

Hence, head range based on duty head of 54.0 m shall be 59.4 m to 40.5 m.

4.0 Selection of Type of Pumps - VT or Horizontal Centrifugal Pumps

4.1 VT Pump

VT pump shall be installed at a floor level above the maximum WL in the sump. The maximum WL is 11.0 m, which is above HFL at 10.5 m. There is no necessity for priming in case of VT pumps. Hence, a VT pump is feasible and generally suitable for any installation with sump and pump house located above sump.

4.2 Horizontal Centrifugal Pump

The pump shall preferably be a double suction, single stage, horizontal split casing so as to have a low NPSHr.

It is necessary to check the feasibility of a centrifugal pump installation to ensure that the pump mounting floor is not dangerously below GL and the risk of the motor getting waterlogged if any valve or piping inside the pump house burst. The feasibility checks shall cover the following two vital points.

i) The top of volute of the pump should be below minimum WL by the magnitude of head losses in suction piping and valve to ensure that the pump can be fully primed without any vacuum pump
ii) Suction specific speed (Nsss) of the pump at the site installation shall be below 145 (SI)/7,500 USCU as discussed in Chapter 5, subsection 5.8.4.

4.2.1 Pump installation level from the aspect of the case of priming

Size of pump volute and installation level
Figure 17.40 and the equation for impeller diameter in the book ‘Centrifugal Pump’ by Karassik can be referred to calculate impeller diameters. See Figure 5.1.A.2 reproduced. The dimensions are head in feet and Q in US gpm.

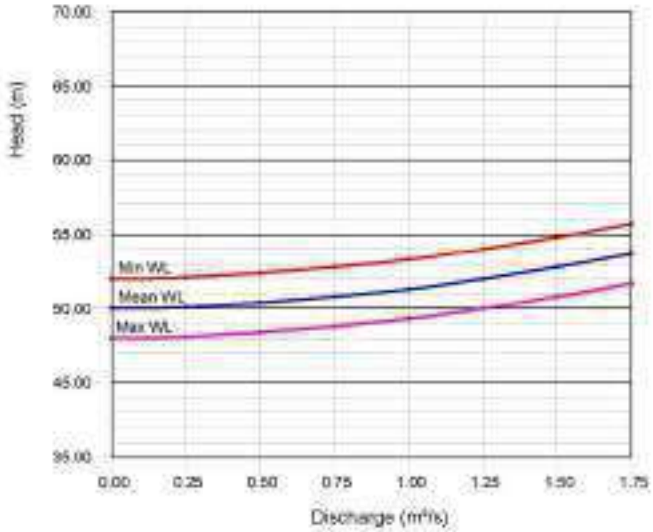

FIGURE 5.1.A.1 : SYSTEM HEAD CURVES

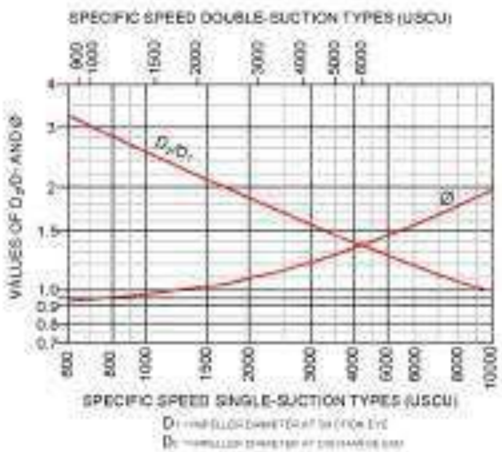

FIGURE 5.1.A.2 : VARIATION OF D_2/D_1 RATIO AND Ø WITH SPECIFIC SPEED

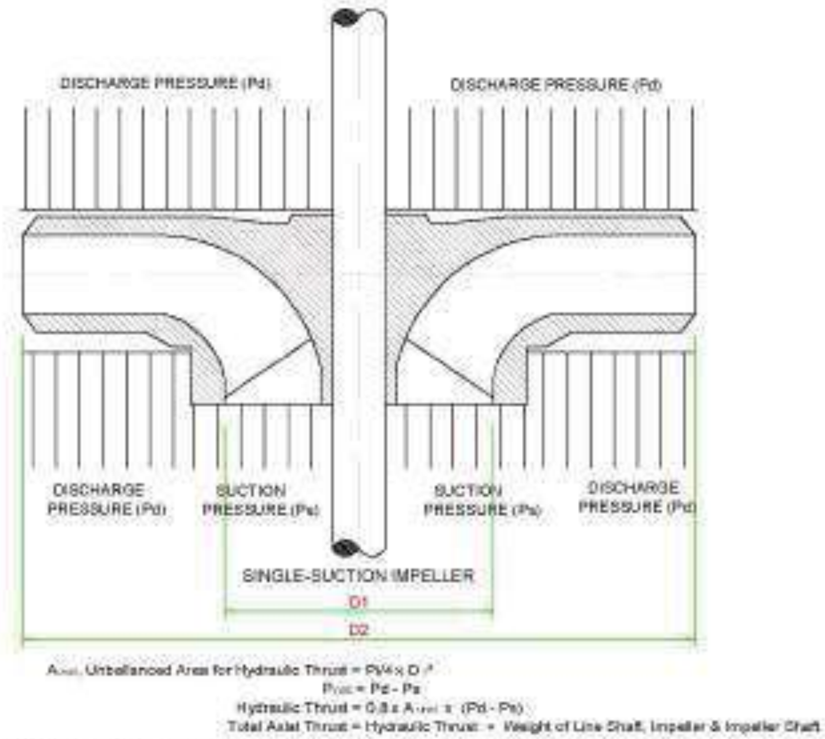

FIGURE 5.1.A.3 : ILLUSTRATION AND MAGNITUDE OF HYDRAULIC THRUST

$Q = 0.35\ m^3/s$ = 5551 US gpm
H = 54.0 m = 177.17 feet

Ns (specific speed) = $N\sqrt{Q}\ /H^{0.75}$

Where N = rpm
Q = US gpm
H = head, feet

If N = 980 rpm, Ns = 1,503 USCU

and N = 1,480 rpm, Ns = 2,271 USCU

Referring to subsection 5.7.1 of the Chapter Ns = 1,503 USCU is low and not suitable. Ns = 2,271 USCU is acceptable. Hence, the pump speed shall be about 1,480 rpm (1,500 rpm synchronous) if a centrifugal pump is selected.

From the Figure 5.1.A.2 for Ns = 2271 USCU (double suction pump)

$\phi = 1.03$

Using the equation given in the book,

$$H = \left(\frac{D_2 N}{1840\phi}\right)^2$$ …… H in feet and diameter of impeller, D_2 in inches

Substituting and on calculation

D_2 = 17.05 inches = 433 mm

Usually, volute outside diameter is 2 to 2.25 times of impeller diameter

Volute outside diameter = 975 mm

Assuming that the combined height of feet of pumps, base plate, and c.c. foundation as 500 mm,

Minimum height of the top of volute above pump floor level

= 975 + 500 = 1475 mm = 1.475 m = 1.5 m

The safe margin for assumptions and variation of the volute size of different pump manufacturers

= 200 mm = 0.2 m

Therefore, the design height of the top of volute above the pump floor

= 1.5 + 0.2 = 1.7 m

Head loss in suction piping, fitting, and valve (as per 3.7 above) = 0.3 m = 300 mm

Hence pump floor level = Min WL - 1.7 - 0.3

= 7.0 - 2.0

= 5.0 m RL

4.2.3 Pump Installation level from the aspect of safe suction specific speed (Nsss)

For cavitation-free operation,

Nsss ≤ 145 (SI)

Using the equation in subsection 5.8.4, Nsss = $\frac{N\sqrt{Q}}{NPSHa^{0.75}}$

Where, NPSHa = Net positive suction head available

Here for Nsss, half Q for double suction pump is to be considered. Solving,

NPSHa ≥ 6.92 m

NPSHa at site installation (refer to subsection 5.8.3)

= Ps - Hfs - $\frac{Vs^2}{2g} - Zs - V_p$

(The parameters are defined in the relevant subsection, hence not repeated).

Here Ps = 9.55 m (atmospheric pressure at an altitude of 1250 m above MSL)

Hfs = 0.3 m (as before)

$\frac{Vs^2}{2g} = \frac{1.78^2}{2xg}$ = 0.16 m (assuming 500 mm suction nozzle for 600ϕ suction pipe)

Vp = 0.427 m (at 30°C water temperature for 40°C ambient)

Hence,

Zs ≤ 9.55 - 0.3 - 0.16 - 0.427 - 6.92

≤ 1.74 m, say 1.75 m

+ve value of Zs implies the suitability of the pump for a 1.75 m suction lift.

Hence, RL at the eye of the impeller ≤ Min WL + Zs

≤ 7 + 1.75

≤ 8.75 m RL

Based on values calculated in 5.2.2 above,

Pump floor level ≤ 8.75 - $\frac{900}{2x1000} - \frac{500}{1000}$

≤ 7.80 m RL

4.2.4 Conclusion about pump installation and recommendation

As seen from 4.2.2 and 4.2.3, the lowest pump floor levels worked out are:

i) For priming 5.0 m RL
ii) For Nsss criteria 7.80 m RL, which necessitates a vacuum pump for priming and complex automation

Thus, design pump floor level shall be 5.0 m RL, i.e., lower of these two values. Ground level at site = 8.25 m RL (as per data).

It is seen that the pump floor level shall be 8.25 - 5.0, i.e., 3.25 m below GL.

In the event of any burst in delivery piping or valve, water will accumulate in the pump house up to at least 3.25 m causing submergence of motors necessitating motor rewinding or insulation, and other major repairs. Such mishappenings have taken place in some badly designed centrifugal pump installations.

Therefore, the selection of a centrifugal pump is with serious risk and not advisable.

5.0 Selection of number of stages and rotating speed (rpm) of VT Pump

Pump head as per 3.9, H = 54.0 m

Head loss at the entrance to 600 mm diameter bell mouth

$H_e = 0.1\ V^2/2g = 0.007$ m

Head loss in column pipe, assuming presently, 10 m length of the column and as per Figure (5) in IS 1710-1972 for 400 mm diameter,

$H_c = 0.25$ m

H_d, Head loss in discharge bend/tee = 0.20 m

(Note: k = 0.5, v = 2.79 m/s, head loss = $kV^2/2g$)

Hence, bowl head, $H = 54.0 + H_e + H_c + H_d = 54.457$ m, Say 54.5 m.

Six options on the basis of variation in the number of stages from 1 to 3 and speeds 980 and 1,480 rpm as per Table below, are evaluated.

Table showing options and evaluation of Ns and efficiency for variations in the number of stages and speed [$Q_{(pump)} = 0.35\ m^3/s$ and $H_{bowl} = 54.5$ m].

Option	Number of Stages	Head per stage (m)	rpm	Specific Speed (SI)	Graph efficiency	Remarks
1	1	54.5	980	28.9	0.86	Not accepted, unstable H-Q curve
2	1	54.5	1480	43.65	0.90	Acceptable
3	2	27.25	980	48.61	0.90	Acceptable
4	2	27.25	1480	73.41	0.87	Acceptable
5	3	18.17	980	65.87	0.88	Acceptable

Option	Number of Stages	Head per stage (m)	rpm	Specific Speed (SI)	Graph efficiency	Remarks
6	3	18.17	1480	99.49	0.85	Not acceptable: shut-off kW is higher than kW at BEP

It is seen from the above analysis that options 2 and 3 are to be compared judiciously for final selection.

Considering that wear and tear and noise level at 1,480 rpm shall be more than that of 980 rpm, the most appropriate option shall be Option 3 (980 rpm two-stage VT pump).

6.0 Sizes/Rating of other design components and parameters

6.1 Motor

i) Lowest bowl efficiency as per 4 above is 0.90

Allowing 5% variation for commercial offer design bowl efficiency is 0.85

ii) Input to bowl assembly (refer to subsection 5.11)

= 9.81 × 0.35 × 54.5 / 0.85

= 220.15 kW

iii) Thrust bearing loss (kW) = $\frac{0.0009225\ X\ T\ X\ N}{100\ X\ 50}$ = 1.08 kW

Where

T = thrust in kg = F = 5,000 kg (refer to 6.2 below)

N = operating speed rev/min = 980 rpm

iv) Line shaft bearing loss = 0.5 kW (as per IS 1710)

v) Input to pump [(ii) + (iii) + (iv)] 221.73 kW

Considering 10% margin of power in motor, rating of motor required 243.90 kW

Rounding to commercial rating 250 kW

Note: Calculations for motor rating are to be done to enable detailing specifications for associated electrical equipment. Motor rating should not be specified in the tender specifications.

vi) As the rating is 250 kW, as seen from subsection 5.20.5, HT motors at 3.3 kV can be adopted. In case of a generally clean, dry environment, Screen Protected Drip Proof (SPDP) or Open Drip Proof (ODP) motors (IP23) can be selected as they allow air to circulate freely through the windings for cooling and prevent water dripping from above entering the windings. This is normally adequate for most of the indoor pump sets. In case of dusty environment, frequent cleaning of the vents, (and fly and moth menace during nights which may be sucked into the motor) and risk of water splashes, select totally enclosed, fan cooled, IP44 (TEFC) motors which allow external air inside the motor through a fan, but are safe against not only water dripping from above, but also water splash.

vii) The motor shall be designed for Class F insulation with a temperature rise limit for Class B. As the altitude at the site is 1250 m RL, the temperature rise limit shall be reduced by 2.5 °C, i.e., at the rate of 1 °C for every 100 m higher altitude above 1,000 m RL.

6.2 Impeller shaft and line shaft diameter

i) IS 1710: Specification for vertical turbine pump stipulates the following formula for impeller shaft. The same can also be applicable to the line shaft.

$$S = \sqrt{\left(\frac{63.7F}{D^2}\right)^2 + \left(\frac{492p \times 10^6}{ND^3}\right)^2}$$

Where

S = combined shear stress in kgf/cm^2 shall be either a maximum 30 per cent of the yield stress or maximum 18 per cent of the ultimate tensile strength of the shaft steel used

F = axial thrust in kg of the shaft, including hydraulic thrust plus the weight of the shaft and all rotating parts line shaft, impeller shaft, impeller, etc., supported by it

p = power transmitted by the shaft in kW (advisable to adopt motor kW)

N = revolutions per minute (rpm)

D = shear diameter in mm at the root of threads or minimum diameter at undercut

ii) Calculations for hydraulic thrust

The first step is to determine, impeller diameters - outside diameter D_2 and inside diameter D_1.

Ns (specific speed) = $\frac{N\sqrt{Q}}{h^{0.75}}$

Where N = rpm

Q = discharge, m^3/s

h = head per stage, m

h = $\frac{54.5}{2}$ = 27.25 m, 89.41 feet

N = 980 rpm

Q = 0.35 m^3/s, 5551 USgpm

Hence, Ns = 48.61 (SI), 2512.5 (USCU)

From Figure 17.40 from the book Centrifugal Pump by Karassik, see reproduced Figure 5.1.A.2.

D_2/D_1 = 1.55 and ϕ 1.15

Using the equation from the above book,

$\phi = \frac{D_2 N}{1840 \sqrt{h}}$

Substituting for ϕ = 1.15, h = 89.41, N = 980 and solving,

D_2 = 20.416 inches = 0.518 m

Hence, D_1 = 0.335 m (Nominal)

Adding 2×6 mm (approx.) thickness of wearing ring, D_1 = 0.347 m

As per guidelines on Page 2.276 and 2.277 of Pump Handbook, 4th edition, refer to Figure 5.1.A.3 for an illustration of hydraulic thrust.

Unbalanced area = $\frac{\pi}{4} x D_1^2$

= 0.095 m^2 = 946 cm^2

Unbalanced pressure (neglecting and assuming zero pressure on the suction side of the eye of the impeller due to submergence being of very low magnitude).

= 54.5 m = 5.45 kg/cm^2

As per guidelines in the Pump Handbook, 70%-80% thrust is to be considered

Hence hydraulic thrust = 946 × 5.45 × 0.8

= 4125 kg

iii) Weight of 10 m long 65 mm diameter line shaft (and density 7,850 kg/m^3)

= $\frac{\pi \, x \, 0.065^2 x 10 x 7850}{4}$

= 261 kg say 300 kg

iv) Weight of impeller (assuming void factor as 0.5 and width as 33% of D_2)

= $\frac{\pi \, x \, D_2^2 x 0.33 D_2 x 7850 x 0.5}{4}$

= 141.4 kg say 200 kg

v) Weight of Impeller shaft = 50 kg (assumed)

Hence axial thrust, F = (ii) + (iii) + (iv) + (v)

= 4125 + 300 + 200 + 50

= 4475 kg, say 5,000 kg

SS 410/416 as MOC of impeller shaft and line shaft is selected

Yield stress = 28.2 kg/mm^2 = 2820 kg/cm^2

Ultimate strength = 49.3 kg/mm^2 = 4930 kg/cm^2

Hence safe stress S = 30% of 2820 kg/cm^2 or 18% of 4930 kg/cm^2
(whichever is lower)

= 846 kg/cm^2 or 887 kg/cm^2 (lower of two)

= 846 kg/cm^2

Substituting values in the equation, (F = 5,000, p = 250, N = 980, S = 992) and solving by trial and error (as the equation is not linear)

D = 53 mm at the root of screw threads or undercut due to sleeve

Adding 2 × 5 mm depth of screw threads and 4 mm corrosion allowance and tolerances

Nominal shaft diameter = 67 mm

6.3 Thickness of Column pipe

The column pipe will act as a closed pressure vessel when the pump is started under shut-off conditions. Considering the specific speed and pattern of pump characteristics, the shut-off head is likely to be 80 m.

The material of construction shall be mild steel.

Hence design pressure (at 1.5 times shut-off pressure)

P = (80/10) × 1.5 = 12 kgf/cm^2

For pressure vessel as per IS 2825-1969

$t = (PD)/(200f_i-p)$

Where p, design pressure = 12 kgf/cm^2

D, internal diameter = 400 mm

F, safe stress = 10kgf/mm^2

J, welding factor = 0.7

Hence, t = 3.45 mm

Adding 4 mm corrosion allowance as the pipe is subjected to corrosion from both inside and outside.

Thickness of column pipe = 7.5 mm = 8mm

Design rating of electrical equipment

7.1 Transformer

a) Total load of four pump motor sets 250×4 = 1,000 kW

Hence transformers kVA required at 0.85 P.F. and 20% margin

$= \frac{1000\times1.2}{0.85}$ = 1412 kVA

The next commercial ratings are 1,600 kVA and 2,000 kVA. in addition to a load of main pump motors, load due to ventilation system, indoor and external lighting, power for crane, surge control devices, and water treatment plant are necessary. Momentary starting overload of transformer on starting last working pump should not exceed 50% of transformer capacity.

Considering these aspects, a transformer of 2000 kVA, one duty + one standby is selected.

b) Check for loading under the last/4th motor starting

Momentary overloading of the transformer to be checked for condition N-1 number of pumps running and Nth pump started.

Here Number of duty pumps = 4

kW each motor = 250 kW

Hence , kVA load for three pump motors running (assuming worst PF 0.85 and motor efficiency = 0.90)

$= \frac{250\times3}{0.85\times0.9}$ = 980.4 say 981 kVA

Maximum starting kVA of 4th pump as six times being direct online starting

$= \frac{250\times6}{0.85\times0.9}$ = 1,960.8 kVA, Say 1,961 kVA

Hence kVA load when 4th pump started

= 981 + 1,961 = 2,942 kVA

Momentary overload = $\frac{2{,}942}{2{,}000} - 1$

= 47% (within the limit of 50%)

Hence acceptable

7.2 Breaker on incoming to transformer

The power supply authority's system is 22 kV (Design 24 kV) and the fault level is 500 MVA.

Therefore, S.C. current $\frac{500}{1.732\times24} = 12.03\ kA$

Full load current of transformer $\frac{2000}{\sqrt{3}\times 24}$ = 48.11A

Standard rating - 24 kV, 12.5 kA, 6,30A

7.3 Motor Control Gear

As the motor is 3.3 kV, vacuum circuit breaker (VCB) can be selected.

(i) Current at 0.85 P.F. and lowest voltage 3.3 kV - 10%, i.e., 2.97 kV

$= \frac{250}{\sqrt{3}\times2.97\times0.85\times0.9} = 63.53A\ (0.9\ motor\ efficiency)$

Specify minimum available rating is 400A as per IS 13118.

(ii) Short Circuit Current rating

Normal impedance with 10% tolerance as per IS 2026 = 6.25 × 0.9 = 5.6%

Z_{min} = 5.6%

Therefore, short circuit MVA of a transformer at 3.3 kV Bus

= (2000 × 100)/(5.6 × 1000) = 35.71 MVA

As motor contributes 10 times its normal full load during fault, contribution of four motors.

S.C. Current = 63.53 × 10 × 4 = 2.54 kA

S.C MVA = 3.3 × 2.54 × $\sqrt{3}$ = 14.52 MVA

Hence, total S.C. MVA = 50.23 MVA

Hence, short circuit breaking capacity of the breaker at 3.6 kV design voltage as per IS 13118

$$= \frac{50.23}{\sqrt{3}\times 3.6} = 8.05 \text{ kA}$$

Breaker rating: 3.6 kV, 400A, 10 kA as per IS 13118

7.4 Incoming breaker and bus-coupler breaker to HT Panel

Normal current = 63.53 × 4 = 254.12A

Specify 630A as per IS 13118. Considering margin for future.

Breaker rating: 3.6 kV, 630 A, 16 kA as per IS 13118

S.C. MVA shall thus be 99.8 MVA

Hydraulic design of Sump and General Arrangement of Pump House

8.1 Sump

As per guidelines in subsection 5.2.10.2,

a) Clearance between the bottom of sump and lip of inlet bell mouth,

C = D/2 = 600/2 = 300 mm

b) Submergence above bell mouth and water depth

Diameter of inlet bell mouth 0.6 m

V (actual) [Refer 2.1] 1.24 m/s

F_D, Froude number $\frac{v}{\sqrt{g}}$ = 0.511

S, minimum submergence required above Lip of bell mouth

= D × [1+2.3 F_D]

= 1.30 m

C (calculated as before) = 0.3 m

∴ Minimum depth of water below minimum WL (S+C)

= 1.60 m

c) Distance between rear wall and centre of bell mouth,

B_{max} = 0.75 D = 0.75 × 600 = 450 (maximum) = 450 mm

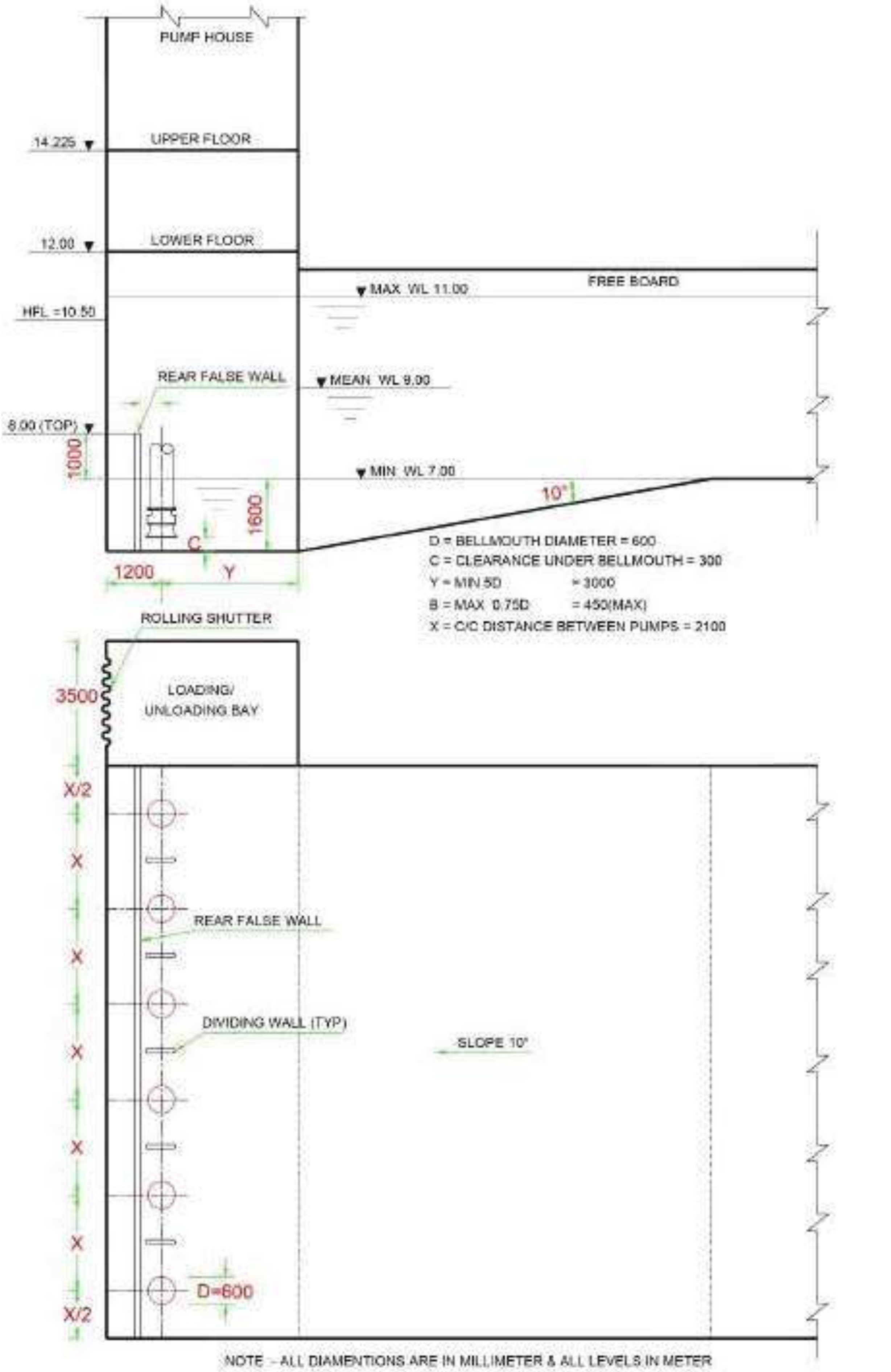

FIGURE 5.1.A.4 : DIMENSIONS OF SUMP FOR V.T. PUMPS FOR VORTEX-FREE OPERATION

d) Rear False Wall

The size of the base of the discharge head will be 1200 mm, i.e., 600 mm from the centre of the pump, whereas dimension B is 450 mm (max). Therefore, the column and rear wall of the sump will have to be located at least 1,200 mm away from the pump centre keeping a 600 mm margin for nut fastening, etc. Therefore, a rear false wall is necessary at a distance of 450 mm (clear) from the pump centre. A top or false wall will be up to 1000 mm above the maximum water level.

e) Spacing between pumps

Desirable spacing between pumps is 2.5 D, i.e., 1,500 mm. However, the size of the lower flange of headgear/discharge head (accommodating stuffing thrust bearing and flexible coupling would be approximately 3.0 times column pipe diameter, i.e., 1,200 mm. Keeping about 900 mm clearance, the spacing will be 2,100 mm.

f) Slope

As the seen minimum depth of water required is 1.60 m below the minimum WL. In order to minimise excavation cost, a permissible slope of 10° degree is taken. The slope will terminate upstream of the pump at a distance equal to 5 D, i.e., 3,000 mm from the pump centre.

g) Straight Approach

The portion under the pump will be flat from the line of termination of slope up to at least the rear false wall.

h) Baffles/Dividing Walls

Dividing walls will be constructed between pumps to avoid mutual interference. Both ends of each dividing wall shall be rounded. The front edge of the dividing wall shall be in line with the front edge of the suction bell mouth. At the rear end opening 150-200 mm size shall be kept at least up to minimum WL. The top of the dividing wall will be up to 1,000 mm above the minimum WL.

8.2 Pump House

a) Number of floors

As the delivery size is 500 mm, two floors shall be planned. The lower floor for pump installation and delivery piping and valve and the upper floor shall be the operating floor. The lower floor shall be 1.25 m above HFL of 10.5 m or keeping 0.6 m freeboard above max WL of 11.0 m, which works out 11.75 m RL, say 12.0 m.

Headroom between floors shall be 2250 mm.

b) Corbel level for EOT crane shall be 5.5 m above the upper floor

c) Headroom above corbel shall be minimum of 2.0 m for the bridge girder of the crane.

Capacity of EOT Crane

9.1 Design Basis

EOT crane should have a minimum 25% higher capacity to lift maximum load in the pumping station. The loads to be lifted are as follows:

i) The motor can be de-flanged from the discharge head and lifted as separate equipment.

ii) For dismantling column pipe or bowl assembly, the load to be lifted as a single load comprises the following:

- Discharge head
- Column assembly
- Bowl assembly
- Inlet bell mouth

Load (ii) shall be higher than load (i)

9.2 Load (ii) - Discharge head, column assembly, etc.

a) Weight of discharge head

The base of the discharge head shall generally be about 2.5 to 3.5 times column pipe diameter and height shall be about 1.25 times base and solid-void factor = 0.25. The density is 7850 kg/m^3.

Weight = $0.7854 \times (0.4 \times 3.5)^2 \times 1.25 \times 0.25 \times 7850$

= 3776 kg

b) Column assembly

Diameter = 0.4 m

Thickness = 8 mm = 0.008 m

Length = 10 m

The factor for the weight of flanges = 20%

line shaft, etc.

$\therefore$ Weight = $\pi \times D \times t \times L \times 7{,}850 \times 1.2$

= 947.0 kg

c) Bowl assembly

Impeller diameter (D_2) worked out above

= 0.518 m

Bowl diameter = 1.15% of impeller diameter

= 0.6 m

Height of bowl = 1.25 times bowl diameter

= 0.75 m

Solid-void factor = 0.5

Weight = $\frac{\pi}{4} \times 0.4^2 \times 0.75 \times 0.5 \times 7{,}850$

= 370 kg

d) Inlet bell mouth (VT Pump)

Weight	=	150 kg (assumed)
∴ Total load	=	(a) + (b) + (c) + (d)
	=	5,243 kg
	=	Say 5,500
Capacity of crane	=	1.25 × 5,500
	=	7,000 kg

Dynamic load of pump-motor set on Civil structure

10.1 Dead Loads

a) Due to the pump set

As worked out in 9.0 above the weight of pump set = 5,500 kg

b) Motor

As per enquiry,
Weight of motor = 3,000 kg

c) Weight of base frame and sole plate

= 500 kg (assumed)

d) Weight of water in column assembly

Volume of water 10 m long column assembly

$$= \frac{\pi}{4} \times 0.4^2 \times 10 \times 1{,}000$$

= 1,260 kg

10.2 Dynamic load for the design of civil structural members

As per IS 2974 Part IV, and as discussed in subsection 5.15(b) of Chapter 5 Part A, a dynamic factor of 3 is applicable for pumps and 2 for the motor. This dynamic factor is logical for a centrifugal pump mounted on a foundation. In the case of the VT pump, the bowl assembly is submerged in water and hence vibrations are dampened. Vibration in discharge head and column assembly are of low magnitude.

Considering the above, low dynamic factor 2 instead 3 is applied for dead weight bowl assembly, column assembly, and discharge head. For water in column assembly, a dynamic factor of 1.25 is considered. For motor, the factor shall be 2 as per IS 2974.

Hence, dynamic load = 5,500 × 2 + 3,000 × 2

$$+ 500 \times 2 + 1{,}260 \times 1.25$$

$$= 19{,}575 \text{ kg}$$

Adding 10% for inaccuracy in estimation,

Design dynamic load on civil structure $= 19{,}575 \times 1.1 = 21{,}532$ kg

$= 22{,}000$ kg (rounding off)