3.1 PUMP-FED VS GRAVITY WATER SUPPLY

Water pumping is required in situations where site conditions do not favour the use of gravity supply. This may occur in irrigation or water supply projects. In either case, gravity systems tend to involve high capital costs but low operating costs. On the other hand, pumping systems tend to require lower capital costs but high operating costs. The choice between gravity supply, and pump fed supply is therefore, an economic one. When the economic case is not obvious, then the economic viability of each alternative must be established, and the economically superior alternative chosen.

3.2 DESIGN SPECIFICATIONS FOR PUMPING SYSTEMS

The requirements to be met by any pumping system is specified as:

- A discharge flow rate for transfer of liquid from suction to discharge reservoir;
- A total pressure head to be overcome by pumping system.

3.2.1 Specification of the discharge flow rate required

The discharge flow rate required is stated in litres per second (l/s), or cubic metres per second \( (m^3/s) \). It is determined by a study of water demand.

Water demand is determined by segregating the total demand into categories such as:

1) Irrigation demand;
2) Domestic demand;
3) Industrial demand;
4) Commercial demand;
5) Institutional demand.

For irrigation, the water demand is derived from the total area to be irrigated and the water required per unit of area irrigated. The water demand required per unit area irrigated depends on the crop, the climatic conditions, and the soil conditions.

For categories of demand except irrigation, the population to be served and its per capita water consumption is estimated, and from this data, the aggregated water demand is computed. The water quality for categories of demand other than irrigation will generally be to human health standards.

In practice, demand for irrigation will usually be isolated and designed for separately, because the location where it occurs and the water quality demanded is often different.

3.2.2 Specification of the total Pressure head to be overcome by pumping system

Figure 1 & 2 shows a schematic drawing of typical water pumping systems.

The total pressure head \( (H) \) to be overcome by a pumping system is stated as metres of water, \( (\text{m of water}) \). This total pressure head is also referred to as the dynamic head. This is because it is the sum of the static head and the friction head. It is referred to as dynamic because it incorporates the head loss due to fluid friction in pipeline, which arises only during the dynamic conditions of fluid flow. The two components of pressure head are elaborated below:
FIGURE 1: BOREHOLE SCHEME WATER PUMPING SYSTEM

FIGURE 2: SPRING SOURCE WATER PUMPING SYSTEM

FIGURE 3: PROPOSED WATER SUPPLY SCHEME
Pumping systems design

Nyangasi

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The total static head therefore depends on the site conditions between the suction and delivery reservoir, and the location of suction and discharge points on the reservoirs. It is determined by a survey of site conditions.

b) Pressure loss due to friction head in pipeline

The friction head \( h_f \) is the total pressure head lost due to fluid friction which occurs as fluid flows through the pipeline. This friction head loss includes that in pipe-work and fittings starting from the suction inlet fitting, through to the discharge pipe outlet. For a given discharge flow rate, this friction loss depends on the pipe material, size, length, and the type and number of fittings. It can be computed once these pipeline specifications are determined.

c) Total (dynamic) head

The total pressure head to be overcome by pumping system is therefore given by the expression:

\[
H = hts + hf
\]

Where,

\[
H = \text{Total or dynamic pressure head to be overcome by pumping};
\]

\[
hts = \text{Total static head to be overcome by pumping};
\]

\[
hf = \text{Pressure head loss due to fluid friction in pipeline}.
\]

3.2.3 Preliminary design procedure for pumping system

The design of a pumping system therefore proceeds in three steps;

(a) Survey of site conditions;
(b) Selection of a pipeline;
(c) Selection of a pump.

(i) Survey of Site conditions

This step determines the opportunities and constraints of the environment at which the pumping system is to be located. The essential data to be determined in this physical survey of the site, is pipeline length and static head to be overcome by pumping.

(ii) Determination of Static Head

This is the level difference between suction reservoir and delivery reservoir.

(b) Pipeline selection

The pressure loss due to fluid friction in a pipeline depends on several factors: pipe size, material, length, fittings, and flow velocity. The total (dynamic) pressure head that the pump must overcome partly depends on the pressure loss due to fluid friction in pipeline (3.2.2.c), and hence on the pipeline data.
A preliminary selection of the pipeline is therefore made using a recommended flow velocity for water pipelines. This flow velocity recommended for preliminary design of water pipelines is chosen such that pressure losses due to fluid friction in pipeline are kept within acceptable limits. This ensures that pumping equipment size and costs are also kept within certain limits.

The recommended range of flow velocities for water pipelines, to be applied during preliminary design, is between 1 and 3 m/s. After this preliminary stage, the design specifications should guide further decisions.

(i) Selection of Pipe size

The selection of pipe size is the first step in system design. The pipe size is selected such that the flow velocity, when the pipeline delivers the design flow rate, remains within a specified range.

The flow velocity recommendation is an empirical guide, aimed at the compromise of ensuring that the pressure loss due to fluid friction in the pipeline is not too high, while the discharge flow rate through the pipeline is also not too low.

The trade-off is therefore between the size of pump, and the size of pipe. The total head to be overcome dictates the size of pump. The size of pump therefore depends on the friction loss in pipe; and this in turn varies inversely with the size of pipe. The size of pump therefore varies inversely with the size of pipe. For a given set of design specifications therefore, the smaller the pipe selected, the larger the pump required, and vice versa, all else being equal.

(ii) Selection of Pipe material

The pressure loss due to fluid friction in pipeline depends on the pipe size, material, length, fittings, and flow velocity. The second step in the selection of pipeline is therefore to select the pipe material. The available pipe materials for water pipelines are: UPVC (plastic), steel, cast iron, and ductile iron. In Kenya, only the first two alternatives are available.

(c) Selection of Pump

(i) Head loss due to friction in pipeline;

Once the selection of pipe size and material is made, the pressure head loss due to fluid friction in pipeline can be determined.

(ii) Total pressure head on pump;

\[ H = h_{ts} + h_f \]

Where,

- \( H \) = Total or dynamic pressure head to be overcome by pumping;
- \( h_{ts} \) = Total static head to be overcome by pumping;
- \( h_f \) = Pressure head loss due to fluid friction in pipeline.

(iii) Choice of pump to satisfy specifications

Finally, the designer can select the pump type and size required, by considering the total head to be overcome, in conjunction with the discharge flow rate required.
3.3 SUCTION DESIGN

Figure 3.2, and 3.3 illustrates typical schematic profiles of a pumping system's suction design. The scheme in Figure 3.2 shows a positive static suction arrangement, where the suction reservoir's water level is above the pump centre line. The positive static suction is shown as the positive head difference between the suction reservoir water level and the pump centre line.

Figure 3.2: Positive Static Suction Head Arrangement

The scheme in Figure 3.3 shows negative static suction arrangement where the suction reservoir water level is below the pump centre line. The negative static suction, often referred to as the suction lift, is shown as the negative head difference between the suction reservoir water level and the pump centre line.

Figure 3.3: Negative Static Suction Head Arrangement

The positive static suction arrangement is common in treated water pumping stations where the pump house level can be located below the treated water reservoir by excavation, if necessary.

The negative static suction arrangement is common in raw water pumping arrangements when the location of the pump house level below the river, lake or dam level would pose problems, such as flooding.
Suction problems are greatly reduced in positive static suction arrangement. This is because this arrangement increases the net positive suction head available at the pump inlet. The positive static suction arrangement is therefore preferred, whenever it is possible.

3.3.1 Net Positive suction Head (NPSH)

Liquid is not sucked into the inlet of a pump. A positive pressure head must exist at the inlet of a pump to push liquid into the pump inlet. Net positive suction head is the measure of this pressure head required at the inlet of the pump.

(a) Net Positive Suction Head Required (NPSHR)

Net positive suction head required (NPSHR) is a characteristic of a pump. The net positive suction head required by the pump is the minimum fluid pressure required at the inlet of the pump to enable it to operate satisfactorily. This characteristic of the pump depends on the pump's design, and is included in the manufacturer's performance specifications for each pump.

(b) Net Positive Suction Head Available (NPSHA)

Net positive suction head available (NPSHA) is the actual fluid pressure at the pump inlet, arising from a given suction design, at a particular geographical location. The balance between (NPSHA) and (NPSHR) is the variable factor that the designer seeks to control by suction design. The object of designing suction arrangements is to ensure that NPSHA at the pump inlet exceeds the NPSHR of the selected pump.

The (NPSHA) for a particular suction design is given by the expression:

\[
NPSHA = h_a + h_s - h_f - h_{vp} \quad (\text{for a design with positive suction head } h_s)
\]

\[
NPSHA = h_a - h_l - h_f - h_{vp} \quad (\text{for a design with negative static suction } h_l)
\]

Where,

- \(h_a\) = Atmospheric pressure operating at site in m of W,
- \(h_f\) = Friction head loss in suction pipework,
- \(h_{vp}\) = Vapour pressure loss at site.

Since NPSHA is fixed partly by site conditions, and NPSHR is a characteristic of each pump design, the two factors have to be considered carefully and reconciled where suction lift (negative static suction) is expected.

3.3.2 Cavitation

Cavitation is a phenomenon of pumping system malfunction, which is often associated with suction design. The avoidance of Cavitation is therefore an important consideration in the planning of a pumping system.

Cavitation is the result of the pumped fluid vaporising within the impeller, or even within the suction pipeline. If the fluid pressure is reduced below its vapour pressure, pockets of vapour will form or boiling within the fluid will occur. As the vapour pockets reach the surface of the impeller, the higher fluid pressure at that point causes them to collapse. The consequence of this is noise, vibration and possibly, structural damage of the pump.
3.3.3 Procedure for checking for Cavitation

The check for Cavitation is an important step in suction design. The object of this step is to ensure that the NPSHA at pump inlet exceeds the NPSHR of the selected pump, by the required margin of safety. The procedure below provides such a check:

1) Determine the NPSHR of the selected pump. A characteristic curve for NPSHR is included in the performance chart of each pump;

2) Calculate the NPSHA. This involves the following steps:
   a) Determine the atmospheric pressure and temperature. Table 3.1 shows the variation of atmospheric pressure with altitude, while Table 3.2 shows the variation of vapour pressure of water with temperature;
   b) Determine the temperature at the pumping site, and hence the vapour pressure of water, using Table 3.2.
   c) Convert the atmospheric pressure into metres of fluid to be pumped;
   d) Convert the vapour pressure into metres of fluid to be pumped;
   e) Solve for NPSHA from the expression:

\[
NPSHA = h_a + h_s - h_f - h_{vp} \quad \text{(for a design with positive suction head \(h_s\))}
\]
\[
NPSHA = h_a - h_l - h_f - h_{vp} \quad \text{(for a design with negative static suction \(h_l\))}
\]

Where,
- \(h_a\) = Atmospheric pressure operating at site in moW,
- \(h_f\) = Friction head loss in suction pipework,
- \(h_{vp}\) = Vapour pressure at site.

3) If the NPSHA obtained from step (2) is greater than the NPSHR obtained from (1), no Cavitation will occur. A good safety margin is 1 metre head of fluid.

4) If NPSHA is insufficient, certain steps may be taken to remedy the problem. The two options for this are: 1) Increase NPSHA or 2) Reduce NPSHR:

   a) **Increasing NPSHA**

   Given an existing suction piping network, little can be done to increase the NPSHA. However, the few possibilities that exist are:

   (i) Increase the free liquid level of the suction sump with respect to the pump, thus increasing the static suction head or reducing the static suction lift. For raw water pumping from river, lake or dam, the free liquid level of the suction sump cannot be increased. The only possibility is therefore the lowering of the pump's mounting;

   (ii) Reduce the distance between the suction sump and the pump, thereby reducing the length of the suction pipe-work, and the pressure loss due to fluid friction in suction pipe:
(iii) Eliminate as many fittings as possible between the suction sump and the pump to reduce minor friction losses in the suction pipe. In this connection, you should select an inlet fitting, and configuration for the suction pipe that has lowest friction loss;

(IV) Reduce the temperature of the fluid pumped to reduce vapour pressure. This may not be practical for most water pumping systems, but may be possible for industrial applications.

b) Reducing NPSHR

NPSHR may be reduced by:

(i) Placing a throttling valve in the discharge line. This will increase the total head to be overcome by the pump, and thereby reduce the discharge flow rate of the pump. This effectively moves the operating point of the pump to a position of lower discharge flow rate where the NPSHR for the pump is lower. The NPSHR is a characteristic of a centrifugal pump, which increases with flow rate.

(ii) A double suction pump substituted for a single suction requires less NPSHA for a given flow.

Table 3.1: Variation of atmospheric pressure with altitude

<table>
<thead>
<tr>
<th>Altitude in Metres</th>
<th>0</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>1500</th>
<th>2000</th>
<th>3000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Atmospheric pressure (MWH)</td>
<td>10.33</td>
<td>10.0</td>
<td>9.75</td>
<td>9.20</td>
<td>8.60</td>
<td>8.00</td>
<td>7.00</td>
</tr>
</tbody>
</table>

Table 3.2: Variation of Vapour pressure with temperature for water

<table>
<thead>
<tr>
<th>Temperature in °C</th>
<th>10</th>
<th>15</th>
<th>20</th>
<th>30</th>
<th>40</th>
<th>50</th>
<th>90</th>
<th>100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vapour pressure of water in (MWH)</td>
<td>0.12</td>
<td>0.17</td>
<td>0.23</td>
<td>0.43</td>
<td>0.77</td>
<td>1.26</td>
<td>7.30</td>
<td>10.33</td>
</tr>
</tbody>
</table>

3.4 OTHER FACTORS IN THE PUMPING SYSTEM DESIGN

Other factors to be considered in design of pumping systems for water are:

(i) Nature of liquid to be pumped in terms of solid matter content and the possible effect of this on the clogging and wear of the pump;

(ii) Economic viability of the installation.
Power required to pump water is determined by the flow rate, and the total head generated as shown below:

\[ P_w = \rho g Q H \text{ watts,} \]

Where,
\[ \rho = \text{Density of water in Kg/m}^3; \]
\[ g = \text{Gravity constant } m/s^2; \]
\[ Q = \text{Flow rate in } m^3/s; \]
\[ H = \text{Total pumping head in metres of water mW}. \]

Taking \( \rho = 1000 \text{ Kg/m}^3 \) and transforming into kilowatts,
\[ P_w = gQH \text{ Kw} \]
\[ P_w = 9.81 \times QH \text{ Kw}. \]

The water horse-power that the pump must inject into the water is therefore fixed once the design specifications of the pumping system are determined.

**Example 1:**

Estimate the water horsepower required to irrigate 10 hectares of rice, in a location where the water demand for that area has been determined as 22.2 litres per second. The total or dynamic head to be overcome by pumping has also been computed as 5 mW.

**Solution:**

\[ P_w = 9.81 \times QH \text{ Kw} \]

But \[ Q = \frac{22.2}{1000} \text{ m}^3/s, \text{ and } H = 5 \text{ m}. \]

**Substituting for Q and H,**
\[ P_w = 9.81 \times \frac{22.2}{1000} \times 5 \text{ Kw} \]
\[ P_w = 1.09 \text{ Kw}. \]

**3.5.2 Pump shaft-power**

The power that must be injected into the pump shaft by the prime-mover includes the water horsepower as well as other losses, namely;

a) Hydraulic losses in the pump;

b) Mechanical losses in the transmission shaft and the coupling between the pump and the prime-mover.
The input power required at the pump shaft is therefore given by:

\[ P_s = \frac{P_w}{\eta_p \eta_c} \]

Where,

\( \eta_p = \text{Overall pump efficiency. Indicative value for horizontal/centrifugal at 1450 rpm} \)
\( \eta_p = 0.55 \text{ for ratings of up to 5 Kw} \)
\( \eta_p = 0.65 \text{ for ratings of 5 – 10 Kw} \)
\( \eta_p = 0.70 \text{ for ratings of 10 – 20 Kw} \)
\( \eta_p = 0.75 \text{ for ratings of 20 – 30 Kw} \)
\( \eta_p = 0.78 \text{ for ratings of 30 – 40 Kw} \)
\( \eta_p = 0.82 \text{ for ratings above 40 Kw} \)

\( \eta_c = \text{Efficiency of transmission coupling. Indicative values are:} \)
\( \eta_c = 1.0 \text{ for direct coupling;} \)
\( \eta_c = 0.95 \text{ for V belt or gear coupling;} \)
\( \eta_c = 0.80 \text{ for flat belt drive.} \)

**Example 2:**

The water horsepower estimated in 3.5.1, (Example 1) is to be provided by a centrifugal pump, close coupled to an electric motor, and operated at a speed of 1500 rpm. Determine the power absorbed from the motor by the pump shaft.

**Solution:**

From 3.5.2, \[ P_s = \frac{P_w}{\eta_p \eta_c} \]

From 3.5.1, \( P_w = 1.09 \text{ Kw}, \)

Therefore, \( \eta_p = 0.55, \text{ and } \eta_c = 1.0; \)

Substituting,

\[ P_s = \frac{1.09}{0.55 \times 1.0} = 1.98 \text{ Kw.} \]

\[ P_s = 1.98 \text{ Kw.} \]

**3.5.3 Power Sources**

The prime movers encountered in small-scale water pumping duties, in irrigation and water supply projects are petrol engines, diesel engines, and electric motors.

1) **Petrol engines**

The availability of petrol driven stationary engines is limited to a maximum power demand of approximately 7.5 Kilowatts. Petrol engines are less reliable than diesel engines and tend to be built with shorter economic lives. To their advantage, their initial costs are lower.
Diesel engines are available in a wide range of power capacities. Standard products of up to 25 Kilowatt capacity are available as stand alone. Locally, larger capacities are only available already assembled in other products such as generating sets, agricultural machines, and earth moving equipment.

3) Electric motors

Electric power is available from the national grid in those locations where the electric power lines already reach.

3.6 Electric Power

3.6.1 Electric Power Supply in Kenya

Electric power from the national grid is available in several parts of Kenya, and is being extended constantly.

When electric power is not available on site, then the cost of bringing it to the site is part of the capital cost of the project. Consequently, when additional cost has to be incurred to extend the power-line to the site, other alternatives such as the use of diesel engine must be examined.

3.6.2 Standard motor sizes and speeds

Electric motors are available in sizes and speeds shown below:

a) Standard motor sizes (bhp);

1/2, 3/4, 1, 11/2, 2, 3, 5, 71/2, 10, 15, 20, 25, 30, 40, 50, 60, 75, 100, 125, 150, 200, 250.

b) Standard motor speeds (Full load, 50 cycles per second local power supply)

2900, 1450, 960, 720 rpm

3.6.3 Electric motor Power requirements

Electric motor speeds can be chosen to make direct coupling to the pump shaft appropriate. Assuming the transmission efficiency is 1, the power output required from the motor then equals the power absorbed by the pump shaft.

Table 3.3 shows typical variation of efficiency of electric motors with altitude, and ambient temperature. The test results are obtained at an operating temperature of 40 degrees, and an altitude of 1000 metres. The temperature derating factor can therefore be ignored when the operating environment is close to the 40 degrees C. This is a common situation in a country such as Kenya.
TABLE 3.3: VARIATION OF FULL LOAD, CONTINUOUS DUTY, ELECTRIC MOTOR EFFICIENCY, AND ALTITUDE

<table>
<thead>
<tr>
<th>ALT. IN METRES</th>
<th>AMBIENT TEMPERATURE (DEGREES CENTIGRADE) AND % MOTOR EFFICIENCY</th>
</tr>
</thead>
<tbody>
<tr>
<td>XXXXXX m</td>
<td>30  35  40  45  50  55  60</td>
</tr>
<tr>
<td>1000</td>
<td>107 104 100 96 91 86 80</td>
</tr>
<tr>
<td>2000</td>
<td>101  98  94  90  85  81  75</td>
</tr>
<tr>
<td>3000</td>
<td>92   90  86  83  79  75  69</td>
</tr>
<tr>
<td>4000</td>
<td>82   80  77  74  70  66  61</td>
</tr>
</tbody>
</table>

From the variation of efficiency with altitude shown in Table 3.3, it can be concluded that a 1% reduction in electric motor efficiency should be applied for every 100 metres above 1000 metre elevation.

Considering the foregoing, the power input required by the electric driving motor can be determined from input power required by pump shaft as follows

\[ P_m = \frac{P_s \times S_f}{\eta_m \times A_f} \]

Where,
\[ P_m \] = Power input required by motor;
\[ P_s \] = Power input required by pump shaft;
\[ A_f \] = Altitude derating factor,
\[ A_f = 1\% \text{ reduction for every } 100 \text{ m above } 1000 \text{ m}; \]
\[ S_f \] = Safety factor;
\[ S_f = 1.50 \text{ for ratings up to } 1.5 \text{ Kw}; \]
\[ S_f = 1.30 \text{ for ratings of } 1.5 - 4 \text{ Kw}; \]
\[ S_f = 1.20 \text{ for ratings of } 4 - 8 \text{ Kw}; \]
\[ S_f = 1.15 \text{ for ratings of } 8 - 15 \text{ Kw}; \]
\[ S_f = 1.10 \text{ for ratings above } 15 \text{ Kw}; \]

\[ \eta_m \] = Efficiency of motor (1500 rpm)
\[ \eta_m = 0.7 \text{ for ratings of } 1 - 2 \text{ Kw}; \]
\[ \eta_m = 0.8 \text{ for ratings of } 2 - 10 \text{ Kw}; \]
\[ \eta_m = 0.85 \text{ for ratings of } 10 - 50 \text{ Kw}; \]
\[ \eta_m = 0.90 \text{ for ratings above } 50 \text{ Kw}; \]
Example 3: The pumping installation at 3.5.2 (Example 2), is located in the Lake Victoria Basin, with an altitude estimated at 1500 m., and an ambient temperature of 35 degrees C. Determine the electric power absorbed by the motor driving the pump.

Solution:

From 3.6.3, \[ P_m = \frac{P_s \cdot S_f}{\eta_m \cdot A_f} \], and \[ A_f = 0.95 \] for an altitude of 1500 metres;

For the location, ambient temperature = 35 degrees C., and derating is ignored;
For the selected pump, \( P_s = 1.98 \text{ KW} \) and therefore, \( \eta_m = 0.7 \) and \( S_f = 1.3 \);
Substituting,
\[ P_m = \frac{1.98 \cdot 1.3}{0.7 \cdot 0.95} \text{ Kw.} = 3.90 \text{ KW}. \]
\[ P_m = 3.90 \text{ Kw.} \]

The electric power computed above is the maximum power that the motor will require, for example during starting. It can be used in planning the switch gear and controls for the motor. It can also be used to forecast the unit power consumption of the pumping system as shown below.

Electric power consumption of pump = 3.50 Kw.
Therefore power consumption of pump per hour = 3.5 Kwh.

Water delivered = \[ \frac{22.2}{1000} \text{ m}^3/\text{s} = \frac{22.2 \cdot 60 \cdot 60}{1000} \text{ m}^3/\text{hour} \]

Therefore, Power consumed per unit of water delivered is given by:

Unit power consumption = \[ \frac{3.5 \cdot 1000}{22.2 \cdot 60 \cdot 60} \text{ Kwh}/\text{m}^3 = 0.044 \text{ Kwh}/\text{m}^3. \]

Unit power consumption = 0.044 Kwh/m$^3$.

Applying a power charge of Kshs. 4.10 per Kwh,
Pumping power cost per unit of water delivered = Kshs. 0.18 per cubic metre.

It should be noted that after the pumping installation is built and is operating, the pumping power cost should be computed from records of actual water delivered and actual power consumed and paid for.

3.7 Engine Power Requirements

For diesel and petrol engines, output speeds may not match the speed requirements of the pump, and the use of transmission coupling should therefore be considered. The transmission efficiency of the common V-belt may be taken as 0.95, and this value is then used to adjust output power from the engine.
The actual power of the engine to be chosen is further adjusted by other factors as shown below:

\[ \text{Engine Power requirement } P_E = \frac{P_s \times S_f}{0.95 \times A_f \times T_f} \]

Where,
- \( P_s \) = Pump shaft power requirement before adjusting for transmission losses;
- \( A_f \) = Altitude derating factor;
- \( A_f = 1\% \) reduction for every 100 metres above sea level;
- \( S_f \) = Safety factor, taken as 1.2 for engines;
- \( T_f \) = Ambient temperature derating factor;
- \( T_f = 2\% \) for every 5 degrees centigrade above 30 degrees;

**Example 4:**

Considering the pump installation in 3.5.2 (Example 2), what is the engine power capacity required to drive the pump. Assume the installation is at the Lake Victoria Basin, with an altitude of 1500 metres and an ambient temperature of 35 degrees centigrade.

**Solution:**

From 3.7.1, Altitude derating factor for the location becomes
\( A_f = 0.85 \), for an elevation of 1500 metres at the location

Similarly, temperature derating factor for the location becomes
\( T_f = 0.98 \) for the ambient temperature for 35 degrees centigrade at the location

Hence, engine power requirement becomes;

\[ P_E = \frac{P_s \times 1.2}{0.95 \times 0.85 \times 0.98} \text{ Kw} \]

Substituting for \( P_s \) from 3.7.1

\[ P_E = \frac{1.98 \times 1.2}{0.95 \times 0.85 \times 0.98} \text{ Kw} = 3.00 \text{ Kw.} \]

\[ P_E = 3.00 \text{ Kw.} \]

The above Engine power requirement is the capacity of the engine required to drive the pump in 3.5.2 (Example 2).
3.7.2 Specific Fuel Consumption of Diesel Engines

The fuel consumption of the Lister/Petter engine models (TS1, TS2, TS3) at full load, continuous duty, are shown in Table 3.4. These are indicative for diesel engines of that design.

**TABLE 3.4: SPECIFIC FUEL CONSUMPTION AT FULL LOAD AND CONTINUOUS DUTY FOR VARIOUS MODELS OF LISTER/PETTER ENGINES:**

<table>
<thead>
<tr>
<th>MODEL</th>
<th>Speed in revs. Per minute</th>
<th>2500</th>
<th>2000</th>
<th>1800</th>
<th>1500</th>
</tr>
</thead>
<tbody>
<tr>
<td>TS1</td>
<td>Power in Kw.</td>
<td>7.3</td>
<td>6.0</td>
<td>5.4</td>
<td>4.5</td>
</tr>
<tr>
<td></td>
<td>Fuel consumption in litres per hour (l/hour)</td>
<td>2.31</td>
<td>1.77</td>
<td>1.58</td>
<td>1.31</td>
</tr>
<tr>
<td></td>
<td>Specific fuel consumption in litres per Kwh)</td>
<td>0.32</td>
<td>0.30</td>
<td>0.29</td>
<td>0.29</td>
</tr>
<tr>
<td>TS2</td>
<td>Speed in revs. Per minute</td>
<td>2500</td>
<td>2000</td>
<td>1800</td>
<td>1500</td>
</tr>
<tr>
<td></td>
<td>Power in Kw.</td>
<td>15.6</td>
<td>13.0</td>
<td>11.6</td>
<td>9.5</td>
</tr>
<tr>
<td></td>
<td>Fuel consumption in litres per hour (l/hour)</td>
<td>4.62</td>
<td>3.64</td>
<td>3.19</td>
<td>2.57</td>
</tr>
<tr>
<td></td>
<td>Specific fuel consumption in litres per Kwh)</td>
<td>0.30</td>
<td>0.28</td>
<td>0.28</td>
<td>0.27</td>
</tr>
<tr>
<td>TS3</td>
<td>Speed in revs. Per minute</td>
<td>2500</td>
<td>2000</td>
<td>1800</td>
<td>1500</td>
</tr>
<tr>
<td></td>
<td>Power in Kw.</td>
<td>23.4</td>
<td>19.5</td>
<td>17.4</td>
<td>14.2</td>
</tr>
<tr>
<td></td>
<td>Fuel consumption in litres per hour (l/hour)</td>
<td>6.93</td>
<td>5.47</td>
<td>4.78</td>
<td>3.86</td>
</tr>
<tr>
<td></td>
<td>Specific fuel consumption in litres per Kwh)</td>
<td>0.30</td>
<td>0.28</td>
<td>0.27</td>
<td>0.27</td>
</tr>
</tbody>
</table>

It can be seen that at 1500 rpm, the specific fuel consumption of the three engine sizes range from 0.27 to 0.29 litres per Kwh. The specific fuel consumption can therefore be used as a performance indicator to forecast pumping costs, as has already been done for electric power.

3.7.3 Diesel Engine Power Costs

The costs of diesel fuel were Kshs. 24 per litre in 1994, and reached Kshs. 45 per litre in 2001. Using the example of a 15.6 Kw diesel engine, with a specific fuel consumption of 0.28 litres per Kwh, the comparative costs of electric and diesel fuel energy are as shown in the Table 3.5 below:

**Table 3.5: Comparison of National Grid Electric Power Supply and Diesel Engine Power Costs**

<table>
<thead>
<tr>
<th>YEAR</th>
<th>Electric Energy cost Kshs per Kwh</th>
<th>Diesel Fuel costs Kshs per litre</th>
<th>Diesel Fuel energy costs Kshs per Kwh</th>
</tr>
</thead>
<tbody>
<tr>
<td>1994</td>
<td>4.1</td>
<td>24</td>
<td>6.7</td>
</tr>
<tr>
<td>2001</td>
<td>11.0</td>
<td>45</td>
<td>15</td>
</tr>
<tr>
<td>2005</td>
<td>11.0</td>
<td>70</td>
<td>19.6</td>
</tr>
</tbody>
</table>

It is seen that the costs of energy from diesel engines when compared with electric power are higher, and has not changed with time.

The diesel engine also has higher maintenance costs thereby further increasing its operating and maintenance costs.

Furthermore, the capital cost of a diesel engine is much higher than that of the equivalent electric motor, while its reliability is lower.

It is therefore apparent that the diesel engine power option is not economically justifiable, where electric power from the national grid is available.