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Optimal selection of variable-speed turbine pump and analysis of the effect of the rotational speed[1]

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ABSTRACT

A computer implementation is presented to analyze irrigation networks with sprinkler systems and to select the most economical pump among a wide range of variable-speed vertical turbine pumps. The program output consists of a network analysis, a system demand curve, a list of the infeasible pumps with the reason for rejection, and a list of the feasible pumps. The feasible pumps are listed in ascending order with respect to their input power. The selection of the pumps can be based on a range of pump rotational speeds rather than a single speed. An application example based on a real-world data is presented. The program has a built-in database that contains characteristic curves for about 100 different vertical turbine pumps obtained from international pump manufacturers. The user can modify the database and can deactivate or activate the available pumps in the database. Using the computer program, the effect of pump rotational speed on the pump selection is studied. It is found that the rotational speed is a very important factor to consider when selecting a pump. A pump with the least input power at a given rotational speed may become among the worst pumps at another rotational speed for the same application. Furthermore, it is found that as the number of rotational speeds considered increases, the chance of selecting the pump with the optimal operational cost increases.

Keywords: Irrigation systems; pump selection; turbine pumps; vertical pumps.

INTRODUCTION

Vertical turbine pumps are also known as deep-well pumps, bore hole pumps and turbine well pumps. These pumps, used to pump water from deep wells dug in confined or unconfined aquifers, are usually multistage pumps and are driven either by diesel engines or electric motors via a vertical shaft. The shaft can be easily raised or lowered from the top to allow proper positioning of the impeller in the bowl. Vertical turbine pumps can handle capacities from 2 m^3/hour (10 gpm) to 5700 m^3/hour (25000 gpm) with heads up to 300 m (1000 ft) (Karassik *et al.* 1986). The most common applications for vertical turbine pumps are pumping well-water for irrigation and other agricultural purposes, for industrial water supply and

[1] The computer program developed in this study is available from the author at no cost.

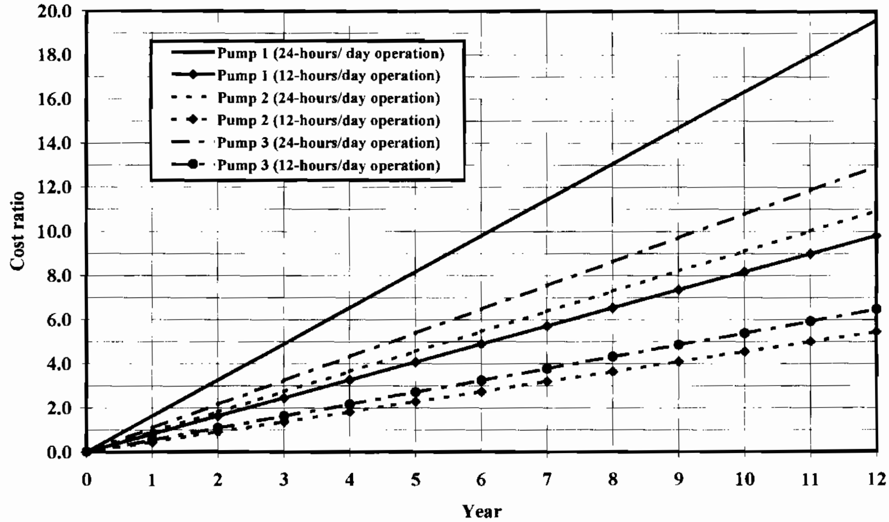

Fig. 1. Ratio of the operational cost to initial cost versus the service life of the pump for three vertical turbine pumps.

for municipal water supply. This study is concerned with variable-speed vertical turbine pumps that can be used for any of the above applications, but the specific application presented relates to irrigation networks with sprinkler systems.

The operational cost for a vertical turbine pump is high when compared to the initial cost, especially in the long run as is obvious from Fig. 1, which is based on a survey (unpublished data). The survey was carried out as an initial study for the work presented in this paper. The figure shows years of use versus the ratio of operational cost to initial cost for three actual vertical turbine pump models available in the market today. The initial cost considered was taken from actual quotations provided by pump dealers. It is clear from Fig. 1 that for 24-hours/day operation, the operational cost for each of the three pumps exceeds the initial cost in only one year, and the operational cost exceeds the initial cost in two years for 12-hours/day operation. Further investigation of this cost ratio for other pumps reveals the same finding. As different pumps perform differently under different applications, especially with regard to the operational cost, a precise pump selection for irrigation is very important to reduce overhead cost and optimize profits.

The literature on vertical turbine pump selection is limited. Some papers discuss the procedure for an economic pump selection for water-distribution systems in the presence of varying operational conditions (Tarquin & Dowdy 1989, Chase & Ormsbee 1993, Ulanicki *et al.* 1993, Tullis 1994). Roth (1984) discusses variable-speed pump selection for irrigation networks, and how to cover a wide range of operational conditions using a pump capacity envelope bounded by the minimum and the maximum pump characteristic curves. Literature on variable-speed pump selection seems to have ignored the sensitivity of the accuracy of the selection to the pump rotational speed. This paper deals with the variable-speed vertical turbine pump selection for a single operational point, which is the case for most irrigation networks with sprinkler systems.

Vertical turbine pump selection for water irrigation networks involves a tedious and lengthy process with a great many computations. The selection procedure involves analyzing the pipe network using an iterative procedure, then trying all competitive pumps for the system under consideration. Since no software has been presented in the published literature to evaluate the selection of such pumps, a computer program was developed for this purpose. This paper describes the computer program to select variable-speed vertical turbine pumps for agricultural applications with sprinkler irrigation systems. The computer program has been named **PUMPICK**. The program can also be used to select vertical turbine pumps for other applications. The program analyzes the irrigation network, then it evaluates the performance and suitability of all the competitive pumps for the network and ranks all the accepted (feasible) pumps for the application in ascending order according to their input power. The effect of considering a range of pump rotational speeds rather than a single speed on the accuracy of selecting the pump of the minimum input power is also presented to demonstrate the use of the **PUMPICK** computer program. The analysis is performed for a number of pump rotational speeds given by the user. The output of **PUMPICK** includes a list of feasible pumps listed in ascending order with respect to their input power for each rotational speed.

FORMULATION

Figure 2 shows a schematic diagram for an irrigation network consisting of one vertical turbine pump, pumping water from a deep well to an irrigation network with eight pipes and three center-pivot irrigation systems (laterals). A lateral is fitted with impact, spinner, or spray nozzle sprinkler to spread the water uniformly over the circular field (Keller & Bliesner 1990). These nozzles are actually small orifices.

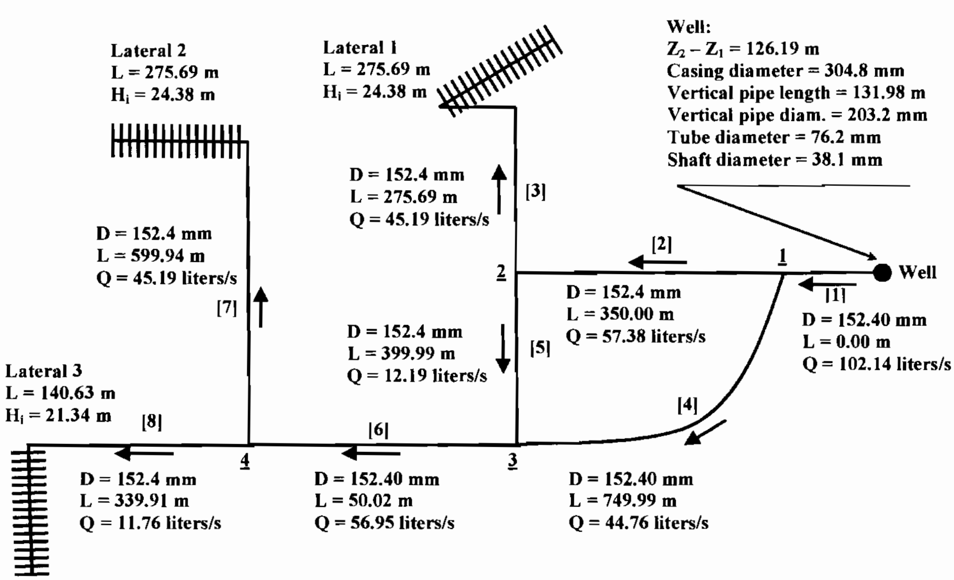

Fig. 2. Schematic diagram for a typical irrigation network.

For a speedy hydraulic analysis, they are theoretically replaced by a single large orifice (Keller & Bliesner 1990). Depending on its length, each lateral requires a minimum discharge (Q_{rl}) and a minimum inlet head (H_i). The network flow distribution can be found using any of the network analysis schemes after imposing the required discharge for each lateral via an upstream control valve. With the final flow distribution established, the total required pump head (H_{rp}) can be found using the equation:

$$H_{rp} = (Z_2 - Z_1) + h_{vp} + \sum_{i=1}^{n}(CQ^{1.852})_i + \sum_{i=1}^{n}(K_m Q^2)_i \tag{1}$$

where Z_2 is elevation of a representative sprinkler along the lateral, Z_1 is elevation of the steady-state well-water level, h_{vp} is head loss along the vertical pipe and is obtained from tables, C is friction loss coefficient, Q is discharge through a given pipe and K_m is minor loss coefficient.

The first term on the right hand side of Eq. 1 stands for the elevation difference between the water demand point (the representative nozzle) and the steady-state water level in the well. The third term represents the total frictional loss along an arbitrary path starting from a point of known hydraulic gradeline (e.g., any of the laterals) to the top of the well, and the last term is the total minor loss along this path (i refers to the pipe number and n refers to the number of pipes along the selected path). One may note that Eq. 1 considers the Hazen-Williams equation for frictional loss. If the Darcy-Weisbach equation for frictional loss is used, the only difference will be in the third term on the right hand side where the exponent will be 2.0 instead of 1.852.

Thus, for a given discharge through the pump, the corresponding required pump head is found using Eq. 1 after establishing the final flow distribution in the network. To simulate the actual hydraulic behavior of the irrigation network, the sprinkler system can be treated as a giant orifice rather than one consisting of small individual nozzles or orifices (Keller & Bleisner 1990). Knowing that the head loss through the orifice is simply the lateral minimum inlet head (H_i), the constant for this giant orifice is calculated at the required discharge; it is then added to the minor loss coefficient (K_m) and saved for subsequent use in the analysis. To obtain the system demand curve, several discharge values through the pump are assumed, and for each discharge, the corresponding required pump head (H_{rp}) is obtained using Eq. 1. This resulting curve is superimposed on the pump head-discharge characteristic curve, which is derived from the original single-stage characteristic curves using the affinity law as follows (Potter & Wiggert 1991):

$$\frac{W_{p2}}{W_{p1}} = \left(\frac{N_2}{N_1}\right)^3 \left(\frac{D_2}{D_1}\right)^5 \tag{2}$$

$$\frac{H_{p2}}{H_{p1}} = \left(\frac{N_2}{N_1}\right)^2 \left(\frac{D_2}{D_1}\right)^2 \tag{3}$$

$$\frac{Q_{p2}}{Q_{p1}} = \frac{N_2}{N_1}\left(\frac{D_2}{D_1}\right)^3 \tag{4}$$

where W_p is pump input power, N is pump rotational speed, D is pump diameter, H_p is pump head and Q_p is pump discharge.

Subscript 1 refers to the conditions at which the original pump curves were established, and subscript 2 refers to the conditions at which the pump curves are to be derived. If the derived pump characteristic head-discharge curve intersects with the system demand curve, then the pump under consideration is a feasible alternative for the application. The discharge and the head at the point of intersection are the design head and the design discharge, or the head and the discharge that the pump would supply if installed for the application under consideration. The pump design input power and efficiency are read at the point of intersection by drawing a vertical line from the point of intersection to the corresponding power and efficiency curves, respectively.

If the pump rotational speed is changed, a new pump head-discharge characteristic curve is obtained, and consequently, different design discharge, head, input power and efficiency would be obtained. Thus, it is important not only to consider different pumps for analysis, but also to consider different values for the rotational speed in order to determine the most economical pump for the application. Figure 3 shows a sketch for a system demand curve intersecting four generated pump head-discharge characteristic curves for four different rotational speeds. The vertical and horizontal bold dashed lines show the required pump discharge and head, respectively. Clearly, if the pump runs with the lowest rotational speed shown, it is not a feasible alternative as it will not supply the required discharge although

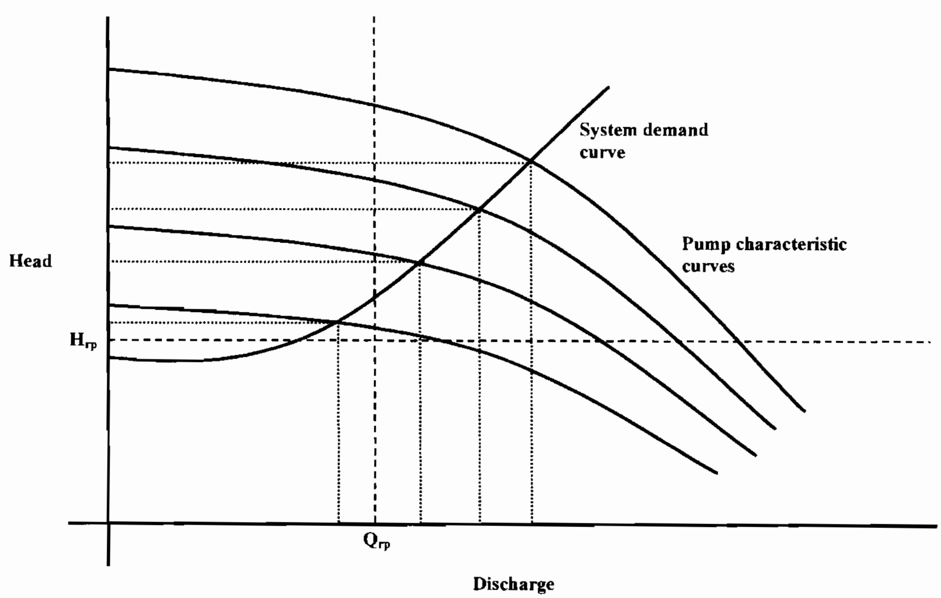

Fig. 3. Intersection of system demand curve with pump characteristic curves.

it satisfies the required head. The other three rotational speeds result in the pump being feasible as the design discharge and head will exceed the required corresponding quantities. In the same fashion, a family of characteristic curves is generated for each pump by considering a set of different rotational speeds. Thus, the user ends up with a number of possible, feasible alternatives, and selection of the most economical pump becomes possible.

IMPLEMENTATION

General

The computer program (**PUMPICK**) was written to perform selection of variable-speed vertical turbine pumps for irrigation networks with sprinkler systems (e.g., center-pivot laterals). The program supports both SI and the English systems of units and has a built-in database containing characteristic curves of discharge-head and discharge-power for about 100 vertical turbine pumps. These pumps are from major international pump manufacturers and all the pumps were tried in the selection process. The user can easily add other pumps by converting the pump curves to data points and listing them in the database with information on the pump, including the model, the manufacturer and the rotational speed at which the characteristic curves were generated. The program was written in FORTRAN 77 high level language so that it can be compiled by most of the available FORTRAN compilers. Figure 4 shows a brief flowchart for the computer program.

Input Files

The program requires two input files to be prepared by the user for a specific problem. The first input file contains information on the irrigation network, including pipe lengths, diameters, roughness factors, minor loss coefficients, water viscosity and specific gravity, pipe and node numbers, interior and pseudo loops. The first input file also requires information on laterals, including the minimum required inlet heads. Figure 5 shows a sample input file for the system shown in Fig. 2, which is for one of the real-world cases considered in this study. The initial assumed flow distribution based on the required lateral discharges is shown with all the information needed for the input file. Although lateral lengths are not needed in the program, they can be used to estimate the required discharge through each lateral. The minimum lateral inlet head (H_l) is also shown and is obtained from tables. The program deals with a single pipe from the well. Thus, Pipe 1 is used as a dummy pipe of zero length to be consistent with the requirements of the program.

The second input file contains data for the characteristic curves (discharge vs. head, discharge vs. pump input power) for the pumps to be considered in the selection process. This file includes single-stage characteristic curves data for the pumps, and it currently has information on about 100 different vertical turbine pump models. The user can easily modify the database by including or excluding pumps of his choice. The third input file contains information on the vertical shaft power loss for different pump rotational speeds and shaft diameters.

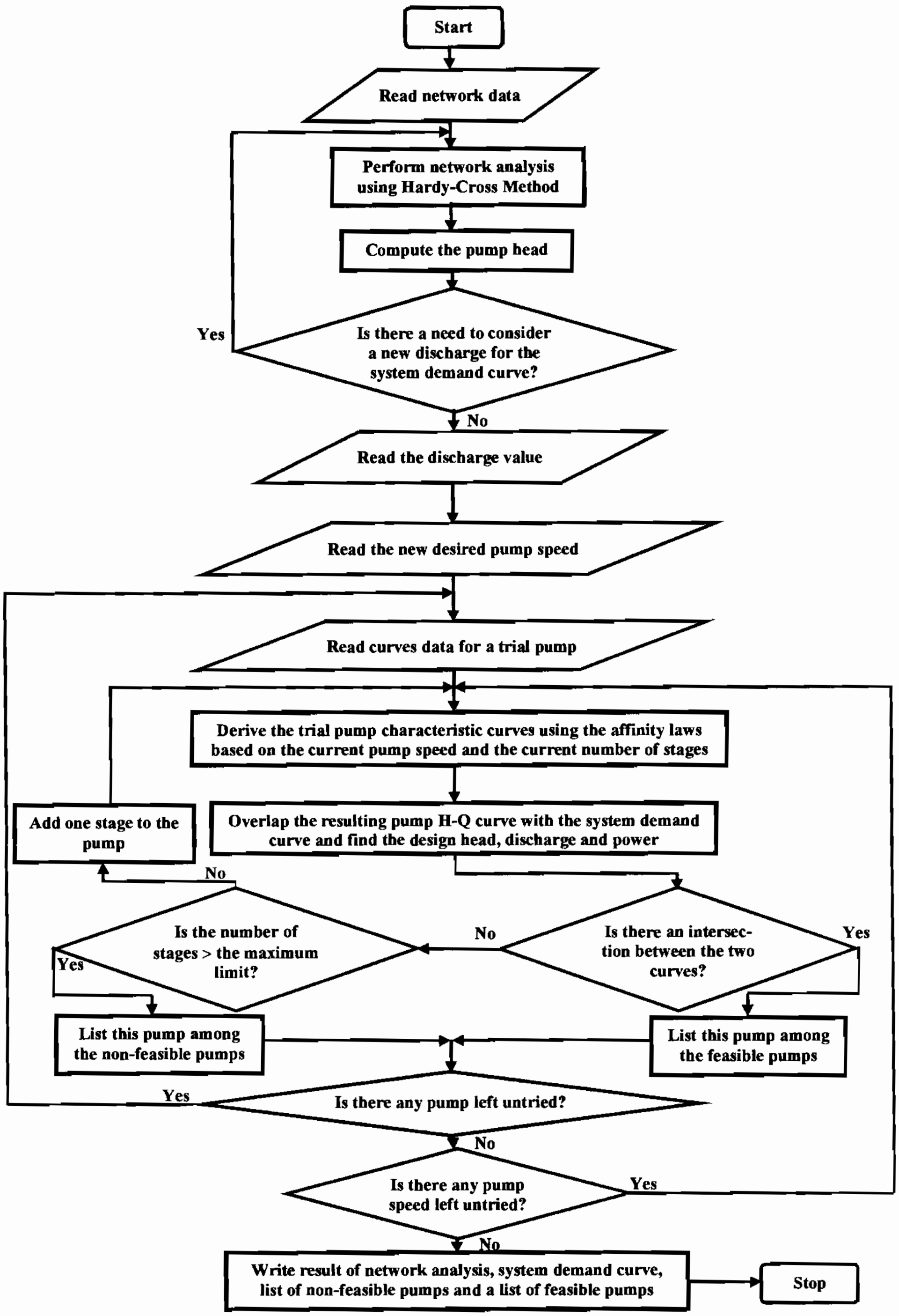

Fig. 4. A brief flowchart for the computer program **PUMPICK**.

```
'Case 10',8,4,0,'SI','hw',1.0e-05,1.0,1000,1000,.0001
1,0,203.2,140,0,102.14
2,350.0,152.4,140,0,57.38
3,275.69,152.4,140,0,45.19
4,749.99,152.4,140,0,44.76
5,399.99,152.4,140,0,12.19
6,50.02,152.4,140,0,56.95
7,599.94,152.4,140,0,45.19
8,339.91,152.4,140,0,11.76
100,1,1,1,2,1,5,1,6
0,3,4,-5,-2
0,4,7,-3,5,6
0,2,-7,8
0,2,-7,8
3,3,24.38,7,24.38,8,21.34
3,2,126.19
1.8,.3,-.1
9,1500,1600,1700,1800,1900,2000,2100,2200,2300
0.01,0.06,0.05
38.1,131.98,0.0,0.0938
```

Fig. 5. PUMPICK sample input file for a typical irrigation network.

Methodology

The program reads irrigation network data from the first input file and assumes that the control valves upstream of each lateral are fully open. It then performs network analysis using the Hardy-Cross method of analysis. After establishing an initial flow distribution in the network based on the required discharge through each lateral, the flow into each lateral is compared to the required quantity (Q_{rl}). If the two quantities do not closely match each other, control valves are theoretically adjusted by changing their coefficients in a certain manner, and the analysis is run again. The procedure is repeated until the resulting discharge into each lateral is close enough to the required quantity. The valve coefficients for the control valves are saved for subsequent use with other discharge values.

Once the final flow distribution at the required discharge is achieved, Eq. 1 is used to compute the corresponding pump head using an arbitrary path selected by the user and extending from a point of known hydraulic gradeline (e.g., a lateral) to the network-vertical pipe junction. To establish the system demand curve, different discharge values are assumed into the network. For each discharge, the corresponding flow distribution and the pump head computation are carried out in the same manner as before, except that the coefficients for the control valves are not altered. Thus, this set of discharge values and the corresponding heads represent the system demand curve.

The program then evaluates different pumps for the current application. With a user-determined set of different pump rotational speeds, the program tries each pump with the first rotational speed, then the second rotational speed, etc., until all the desired rotational speeds have been tested on all pumps. Since desired pump rotational speeds are different from those at which the pump characteristic curves were established, new characteristic curves are derived from the original single-stage curves stored in the data input file using the affinity laws [see Eqs. 2, 3 and 4]. The derived head-discharge, single-stage curve is overlapped with the system demand curve. If there is an intersection between the two curves, then the head and discharge at the intersection are compared to the required quantities. If one or both of them are less than the required quantities, then another stage is added to the pump and new characteristic curves are derived for the current number of stages. The curves are overlapped again, and the process is repeated until achieving the number of stages that would give a head and discharge at the point of intersection equal or greater than the required quantities. This same process is repeated for all the pumps at each desired speed, until covering all the set of the desired pump rotational speeds. The pump that results in an intersection between its head-discharge curve and the system demand curve is considered a feasible selection. Thus, the program usually ends up with a list of feasible pump alternatives that are stored in the output file in ascending order with respect to their input power. The pump with no possible intersection with the system demand curve for any number of stages at any rotational speed is considered a non-infeasible alternative for the application and is stored in a file (with the reason for rejection) for possible view by the user.

Output

There are two output files. The first output file has all the important output data, including the results of network analysis, system demand curve, and a list of the feasible pumps for the application ranked according to their input power. The number of stages and efficiency are also given for each feasible pump. Rejected pumps and reasons for rejection are also listed. The second output file contains the derived characteristic curves for all the pumps tried for the application and for all the desired pump rotational speeds. Figures 6 and 7 show different parts of a sample output for the computer program. The output shown is for the system shown in Fig. 2. Figure 6 shows the output of network analysis, including the flow distribution based on the initially assumed flow distribution shown in Fig. 2. Figure 7 shows the system demand curve, a partial list of the non-infeasible (rejected) pumps at 1500 RPM and a partial list for the feasible (accepted) pumps at 1500 RPM.

APPLICATION EXAMPLE

In the following sections, an application example is presented to explain how the computer program can be used as a tool for the selection of the most economical variable-speed vertical turbine pump for irrigation networks with sprinkler systems. Figure 2 shows a schematic diagram for a real-world irrigation network with a sprinkler system (center-pivot lateral system). All the information needed for the program input is shown on the figure where D is pipe diameter, L is pipe or lateral length, Q is the initially assumed discharge, Lateral refers to a center-pivot sprinkler system rotating about one end to irrigate a circular area, H_i is the minimum lateral inlet head and the quantity $Z_2 - Z_1$ gives the elevation difference between a representative nozzle sprinkler along the lateral and the steady-state well-water level.

The well information is as shown in Fig. 2. The well casing isolates the well from the surrounding environment and imparts the vertical pipe, which transports water from the well to the irrigation network and to which the turbine pump is attached.

```
PROJECT: Case 10
 NUMBER OF PIPES =            8
 NUMBER OF NODES =            4
 METRIC SYSTEM OF UNITS [ DISCHARGE= l/s,HEAD= m, POWER= kW  ]
 FRICTIONAL LOSS FORMULA USED:HAZEN-WILLIAMS

 OUTPUT OF NETWORK ANALYSIS

   1-WHEN THE FLOW RATE IS  100.0% OF THE REQUIRED FLOW RATE

 TRAIL NO. VERSUS FLOW CHANGE FOR HARDY-CROSS ANALYSIS
 TRIAL NO.   RELATIVE FLOW CHANGE    AVG. FLOW CHANGE
    1              .000                     .012
    2              .000                     .012
    3              .000                     .001

 NETWORK FLOW DISTRIBUTION
 PIPE NO.         FLOW RATE
     1              102.140
     2               60.317
     3               45.240
     4               41.823
     5               15.077
     6               56.900
     7               45.090
     8               11.810

  HYDRAULIC GRADELINE AT NETWORK NODES
   NODE NO.         HYD. GRADE LINE
      1              100.000
      2               79.367
      3               77.558
      4               74.911

 MINOR LOSS FACTOR
   PIPE NO.         FACTOR(K)
      3              126.9800
      7               77.9095
      8             2055.3740

  SELECTED PATH LOSSES (MINOR+FRICTIONAL)=      69.998090 m
  VERTICAL PIPE HEAD LOSS =      12.379720 m

  .
  .

 METRIC     SYSTEM OF UNITS [ DISCHARGE= l/s, HEAD= m, POWER= kW  ]
 ------------------------------------------------------------------
                    SYSTEM DEMAND CURVE
               DISCHARGE          HEAD
                 102.14          208.57
                 183.85          379.82
                 173.64          353.53
                 163.42          328.62
                 153.21          305.10
                 143.00          282.97
                 132.78          262.25
                 122.57          242.93
                 112.35          225.04
                 102.14          208.57
                  91.93          193.54
                  81.71          179.97
                  71.50          167.86
                  61.28          157.23
                  51.07          148.11
                  40.86          140.51
 ------------------------------------------------------------------
```

Fig. 6. **PUMPICK** sample output file showing network analysis and system demand curve.

```
1-WHEN PUMP ROTATIONAL SPEED = 1500.0 RPM

**************************REJECTED PUMPS**********************************
NO.PUMP COMPANY    MODEL        REASON FOR REJECTION
 1 MFR. NAME       10FHH        Q REQUIRED > Q MAX. OF THE PUMP
 2 MFR. NAME       8RH          Q REQUIRED > Q MAX. OF THE PUMP
 3 MFR. NAME       10RH         Q REQUIRED > Q MAX. OF THE PUMP

17 MFR. NAME       12H7000      Q REQUIRED > Q MAX. OF THE PUMP
18 MFR. NAME       8JKH         Q REQUIRED > Q MAX. OF THE PUMP
19 MFR. NAME       9FH          Q REQUIRED > Q MAX. OF THE PUMP

65 MFR. NAME       12KH         Q REQUIRED > Q MAX. OF THE PUMP
66 MFR. NAME       12TLC        NUMBER OF STAGES MORE THAN 30

SHAFT POWER LOSS =  2.840 kW

********************PUMPS ACCEPTED FOR THE APPLICATION******************
OPTION   COMPANY      MODEL    STAGES DISCHARGE   HEAD  T.POWER      EFF %
                                          l/s        m      kW
  1     MFR. NAME     H12HC      21     103.52   210.84  254.20      84.23
  2     MFR. NAME     12RMC      17     102.26   208.74  262.48      79.78
  3     MFR. NAME     12THC      17     103.46   210.70  262.67      81.41
  4     MFR. NAME     12DEH      16     104.63   212.55  263.64      82.75

 11     MFR. NAME     13MQH      14     102.57   209.27  274.88      76.61
 12     MFR. NAME     12FHH      19     105.69   214.33  277.20      80.17
 13     MFR. NAME     12HQSH     16     103.69   211.09  277.88      77.27

 23     MFR. NAME     SV12C      23     102.97   209.91  307.65      68.92
 24     MFR. NAME     14EC       11     109.35   220.22  310.10      76.18
 25     MFR. NAME     12RH       26     102.75   209.60  317.22      66.60

NOTE: THE SHAFT POWER LOSS IS ALREADY ADDED TO THE POWER
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Fig. 7. **PUMPICK** sample output file showing a list of feasible and non-feasible pumps.

The shaft, which transmits rotation from the driver at the top to the turbine pump, goes through the pipe and is isolated from water via a tube. The gap between the tube and the shaft is filled with oil to minimize wearing effects. The diameters for the casing, vertical pipe, tube and the shaft for the application under consideration are as shown in Fig. 2. These diameters are used by the computer program to estimate the shaft power loss and the head loss along the vertical pipe.

The discharge through each lateral is estimated by the user and is based on the lateral length and the nature of the crops being considered. This kind of information is available in tables. For example, wheat needs 7.57 liters per minute per 1000 m^2. The minimum lateral inlet head (H_i) is also available in tables. This value represents the minimum inlet head that needs to be maintained to achieve a satisfactory lateral operation. For instance, a 275-meter lateral would require a minimum inlet head (H_i) of 24.38 meters. The lateral inlet head is converted to a minor loss, and the minor loss coefficient is calculated using this inlet head value and the required discharge through the pipe under consideration. This minor loss coefficient is added to the minor loss coefficients in the network input data.

With the required discharge through each lateral estimated as explained above, the initial network flow distribution that satisfies the continuity equation is manually established by the user. Figure 2 shows the initial flow distribution for the example under consideration. The flow distribution shown is based on the required flow rate through each lateral. Figure 5 shows the input data for this example. Figure 6 shows the output of the final flow distribution at the required discharge. It is clear that the required discharge through each lateral is successfully imposed by **PUMPICK** in the manner described earlier. The hydraulic gradeline at the network nodes is calculated by the computer program using a reference hydraulic gradeline (100 meters in this example) at the network-vertical pipe junction. The system demand curve is also shown in Fig. 6. This curve is overlapped with the pump head-discharge characteristic curve to obtain the design values for head, discharge and input power.

Figure 7 shows a partial list of the output resulting from analysis of about 100 pumps for the application under consideration. A partial list of the non-infeasible (rejected) pumps is shown with reasons for rejection. A partial list of the feasible pumps is also shown with the model number, company name, number of stages, design discharge, design head, design input power and the efficiency. The best feasible pump for the application requires a minimum input power of 254.2 kW, while the last pump can do the job for an input power of 317.2 kW. It is worth noting that both pumps are provided with the minimum possible number of stages to supply just the required head and discharge.

EFFECT OF ROTATIONAL SPEED ON PUMP SELECTION

The computer program **PUMPICK** is used for analyzing the effect of rotational speed on the selection of the best pump. Since this type of study has not previously been published, such an analysis is presented herein. For this purpose, 10 real-world cases were considered, and details on these systems are shown in Table 1 and Fig. 8.

About 100 different pump models from different companies, whose characteristics curves are stored in the database, are tried for each case and under three conditions for the pump rotational speeds:

—Condition 1: A single pump rotational speed of 1800 RPM is considered;
—Condition 2: 9 different pump rotational speeds are considered, and they are (1500, 1600, , 2200 and 2300 RPM); and
—Condition 3: 41 different pump rotational speeds are considered, and they are (1500, 1520, 1540, . . . , 2260, 2280 and 3000 RPM).

The **PUMPICK** program is run in the same fashion explained earlier for pump selection for each of the three conditions mentioned above and for each of the 10 real-world cases. Figure 9 summarizes the results of analyses for these cases. The six curves shown represent the minimum and maximum pump input power among the feasible pumps for the three conditions mentioned above. For example, for Case 2, the maximum input power is 270 kW for both Condition 2 and Condition 3 (i.e., for 9 and 41 speeds), and the minimum input power is 95 kW for all the three conditions. The difference between the upper and the lower curves in Fig. 9 shows the extent of the excessive power one may end up paying for simply because of concentrating on one pump brand or model. A typical example, as in Case 1, the difference in input power between the best acceptable (feasible) pump

Table 1. Irrigation network data for the 10 real-world cases

			Laterals, Q_{ri} [a] (H_i) [b]			Pipes, L [c] (D) [d]				
Case	Layout	Q_{rp}	1	2	3	1	2	3	4	5
1	Fig. 8(a)	101.1	101.1	—	—	800.0	—	—	—	—
			(56.4)			(203.2)				
2	Fig. 8(a)	70.2	70.2	—	—	500.0	—	—	—	—
			(35.1)			(203.3)				
3	Fig. 8(b)	193.2	120.3	72.9	—	3.0	650.0	650.0	—	—
			(35.1)	(24.4)		(203.2)	(203.2)	(203.2)		
4	Fig. 8(c)	34.4	22.9	11.5	—	106.0	220.0	—	—	—
			(24.4)	(21.3)		(203.2)	(203.2)			
5	Fig. 8(c)	74.3	37.2	37.2	—	3.0	500.0	251.8	—	—
			(24.4)	(24.4)		(203.2)	(152.4)	(203.2)		
6	Fig. 8(b)	89.9	27.5	62.4	—	300.0	64.0	500.0		
			(24.4)	(24.4)		(203.2)	(152.4)	(152.4)		
7	Fig. 8(b)	77.3	53.5	23.8	—	500.0	150.0	—	—	—
			(24.4)	(21.3)		(203.2)	(203.2)			
8	Fig. 8(b)	83.5	35.4	48.1	—	4.0	380.0	1225.0	—	—
			(24.4)	(203.2)		(203.2)	(203.2)	(203.2)		
9	Fig. 8(d)	71.4	22.8	35.7	12.8	420.0	30.0	80.0	400.0	250.0
			(24.4)	(24.4)	(21.3)	(203.2)	(152.4)	(203.2)	(152.4)	(152.4)
10	Fig. 2	102.1	Shown	Shown	Shown	Shown	Shown	Shown	Shown	Shown

[a]—discharge in liters per second,
[b]—lateral minimum required inlet head in meters,
[c]—pipe length in meters,
[d]—pipe diameter in mm.

and the worst acceptable (feasible) pump is about 190 kW, although both are feasible and have the minimum possible number of stages to supply the required head and discharge.

To further investigate the effect of the pump rotational speed on pump selection, Figs. 10 and 11 show pump speed versus pump input power for two 305-mm impeller pump models. In both curves, the computer program was run for each of the three conditions mentioned previously. The pump of Fig. 10 would require minimum input power of 291.5, 280.5, 292.2 and 279.3 kW for the pump speeds 1500, 1520, 1540 and 1560 RPM, respectively.

One may notice that the curve of Condition 1 for both figures experiences a gradual (but non-linear) increase in the minimum input power then a sudden drop, and this pattern is repeated along the curve. The reason for this can be explained by referring to Fig. 10 at 1960 RPM. The input power for the pump at this speed is 326.6 kW, the power increases gradually from this value until it reaches 381.1 kW at 2060 RPM, and then, it drops to 337.8 kW at 2080 RPM. The reason for this drop is that the minimum required number of stages for the pump between 1960 and 2060 RPM is eight stages to deliver the required discharge and head. But at 2080 RPM, the increased speed caused only seven stages to be sufficient, thus, reducing the power required by the pump.

Figures 10 and 11 illustrate the benefits of considering a range of pump rotational speeds rather than a single rotational speed. When considering a single speed only,

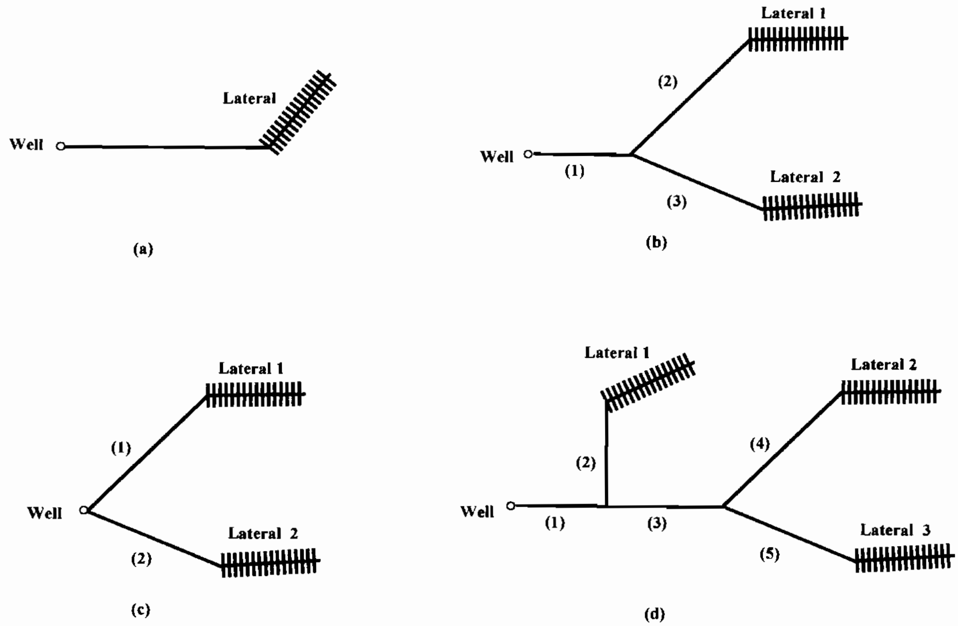

Fig. 8. Layouts of the 10 real-world cases studied.

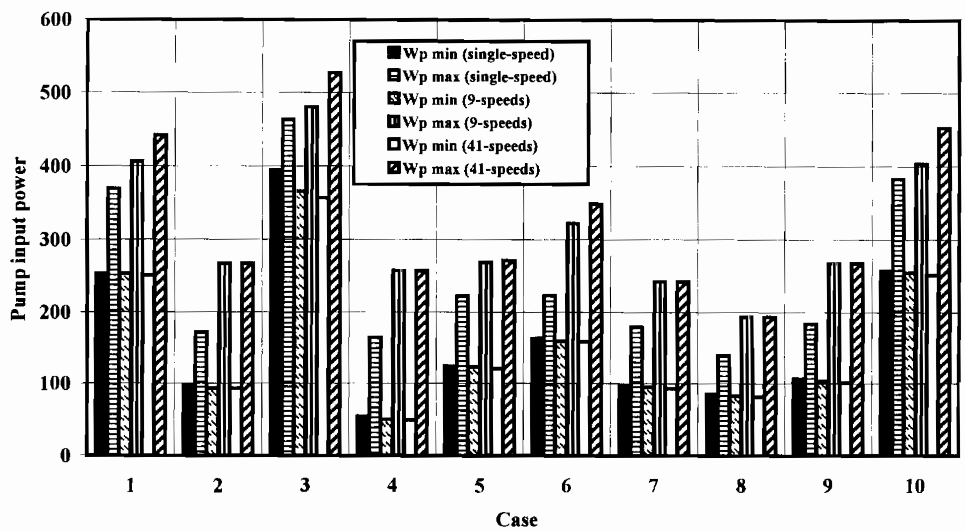

Fig. 9. Pump input power for different rotational speeds.

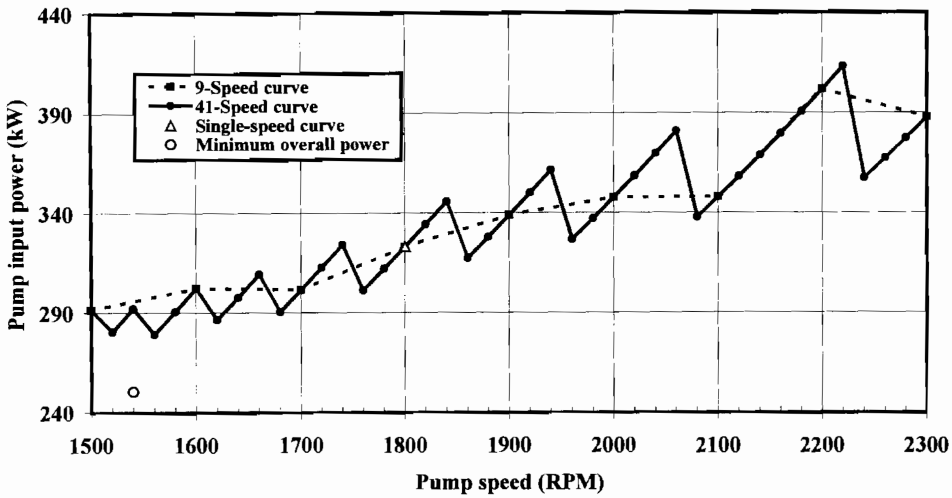

Fig. 10. Variation of pump input power with rotational speed (Pump A).

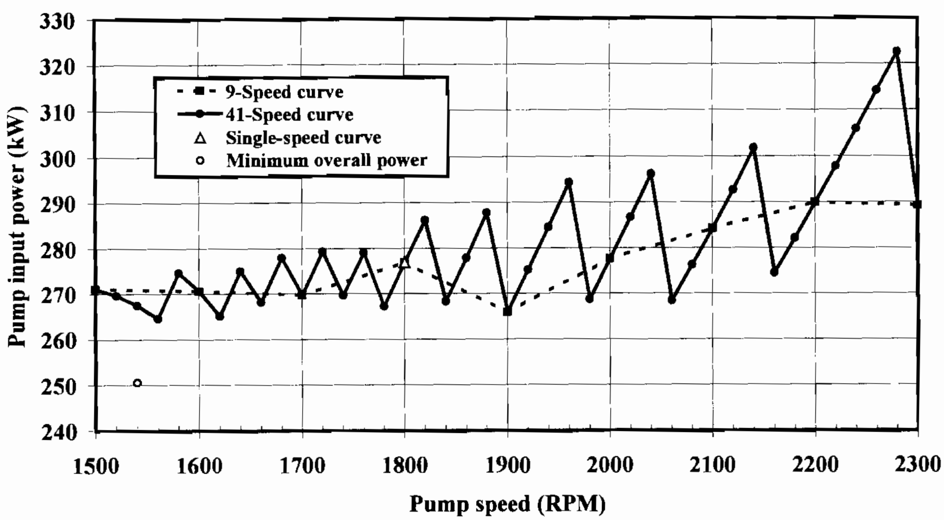

Fig. 11. Variation of pump input power with rotational speed (Pump B).

Fig. 11 shows that the pump input power at 1800 RPM is 276.9 kW, which is not the minimum input power value for this pump. In fact, there are about 19 different rotational speeds that could result in a lower required input power. Furthermore, when comparing the curve of Condition 2 to that of Condition 3 from Fig. 10 or Fig. 11, it is clear that if Condition 2 is used, several points (speeds) of lesser input power will be missed. This shows beyond a doubt the importance of considering as many different rotational speeds as possible in order to reach the optimum operation point for each pump. Figures 10 and 11 reveal a general trend by which the required pump input power increases with the pump rotational speed.

The two curves show optimum operation near 1500 RPM, which is interesting since pumps and pump drivers are usually designed to give optimum performance in the vicinity of 1800 RPM. However, further investigation of this matter is necessary before making any conclusions on this trend.

Figures 10 and 11 are for two pump models only. Thus, to achieve a better selection, all possible feasible pumps are considered for analysis. After including about 100 pump models as trial competitive pumps, the minimum possible pump input power among all competitive pumps and for Condition 3 is found to be 250.6 kW at 1540 RPM as shown in both Figs. 10 and 11. Therefore, the benefit of considering a wide variety of pumps is also evident when comparing this minimum power for the best possible pump to the power requirement for both the pump of Fig. 10 and the pump of Fig. 11. Thus, Figs. 10 and 11 show the effect of both the number of rotational speeds and the number of trial pumps considered on selecting the pump with the minimum input power.

CONCLUSIONS

A computer program named **PUMPICK** has been developed for variable-speed turbine pump selection for irrigation networks with sprinkler systems. The program considers pumps using a built-in database for about 100 pumps from major international pump manufacturers. The program lists the results of network analysis, the rejected (non-feasible) pumps with reason for rejection, and the accepted (feasible) pumps in ascending order with respect to their input power. An application example is presented, showing the nature of the input and the output data.

The computer program **PUMPICK** is also used to study the effect of the pump rotational speed on pump selection using 10 real-world cases of irrigation networks. It is found that the ranks of competitive pumps with respect to the input power at a given rotational speed may dramatically change at another rotational speed for the same application. Furthermore, it is found that as the number of different rotational speeds increases, the reliability of finding the pump with the minimum input power increases. The result of the analysis for these cases shows the extent of variation in performance between different competitive pumps and emphasizes the importance of considering all the competitive pumps in the selection process so as to obtain optimal operational expenditures.

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الاختيار الأمثل للمضخة التربينية متغيرة السرعة وتحليل أثر سرعة الدوران

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خلاصة

تقدم هذه الدراسة برنامج حاسب آلي يقوم بتحليل شبكات الري الزراعي بالرشاشات محورية ، ومن ثم يقوم باختيار المضخة التربينية العمودية المناسبة للشبكة والأكثر اقتصادية ن عدد كبير من الخيارات المتاحة من المضخات. تتكون مخرجات البرنامج من نتائج حليل شبكة الري و منحنى الطلب الهيدروليكي للشبكة بالإضافة إلى قائمة بالمضخات التي خض التحليل عن رفضها مع سبب الرفض وكذلك تشتمل مخرجات البرنامج قائمة لمضخات الصالحة هيدروليكياً للاستخدام في التطبيق مرتبة ترتيباً تصاعدياً حسب طاقة تغيلها . يتيح البرنامج اختيار المضخة على أساس سرعات دوران مختلفة يختارها مستخدم. يقدم في هذا السياق مثال على تطبيق البرنامج على واحدة من الأمثلة الواقعية. حتوي البرنامج على قاعدة بيانات تحوي منحنيات الخواص لحوالي 100 مضخة تربينية مودية مختلفة . يستطيع المستخدم تعديل قاعدة البيانات إضافة إليها أو حذفاً منها ، كما كن تعطيل فحص أي من المضخات الموجودة في قاعدة البيانات وإعادة إعمالها عند حاجة. تمت بواسطة هذا البرنامج دراسة أثر سرعة دوران المضخة على عملية الاختيار ، تبين أن سرعة الدوران تلعب دوراً مهماً لا يمكن إغفاله في عملية اختيار المضخة ؛ فيمكن ن توجد مضخة تتسنم الترتيب بأقل طاقة تشغيل عند سرعة معينة بينما تتذيل الترتيب عند رعة أخرى لنفس التطبيق . فضلاً عن ذلك ، فقد وجدت الدراسة أنه كلما زاد عدد سرعات دوران المستخدمة كلما كانت الفرصة أكبر لاختيار المضخة الأفضل.