"LET THERE BE A PUMP"

by

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INTRODUCTION

This paper takes its title from an offhand remark I made while reflecting on my previous role as a process engineer. Back then, I had said, I would point to a spot on a flow sheet and say, "Let there be a pump, and let it pump so many gallons per minute."

I have since gained a fuller appreciation of what is involved in "letting there be a pump." I believe that I can furnish a unique perspective to the problem of pump application, in the hope that what I say today will help us do our jobs better.

I'm going to be reviewing some basic material, in a way I hope will be useful and interesting. First, I'll talk about pump curves, and where some of the good places and some of the bad places are on them. Second, I want to talk about the system curve, and why the system curve is so important. Third, I'll touch on a few of the problems and pitfalls in pump application engineering.

THE PUMP CURVE

Let's begin by looking at a pump curve. (See Figure 1.) The primary information in the curve is the TDH (Total Dynamic Head) as a function of flow rate. This information is generally shown as a family of curves for different impeller diameters, all at a constant speed. It should be noted that for pumps which are belt-driven, such as rubber lined slurry pumps, the curves depict untrimmed impellers at varying speeds. In addition to the TDH curves, other information includes efficiencies, water horsepower requirements, and NPSH characteristics. There may be some variation in the presentation of the data between manufacturers. For example, some manufacturers show NPSH as a single curve, or they may omit explicit efficiency data in favor of "Iso-Horsepower" lines. Durco includes a stuffing box pressure curve for each pump.

Omitted data on a pump curve is significant, if you know what not to look for. As a general rule, operating conditions which are not recommended will not be published in the curve book. Not all pumps are rated for operation at 3600 RPM, for example. Consider the 2K6X4-10H, and the 2K6X4-10. The latter pump is rated for operation at 3600 RPM, and its curve reflects that fact. There is no such curve for the 2K6X4-10H, however. The 2K4X3-13 should not be operated at 3600 RPM with more than an 11-inch impeller, according to its curve. The specific gravity of the fluid might make even that diameter impractical, because the limitation is imposed by the horsepower. Finally, I have been asked the smallest diameter to which a given pump impeller can be trimmed. The answer is the diameter of the smallest curve shown.

Up to this point, I've discussed the pump curve as it is published. Not all points on a published curve represent equally desirable conditions. The best operation will generally be in the region to the left of the best efficiency point, marked with an "A" in Figure 2.

Operation at the far left, or near shut-off, is possible only for short periods of time. Two things happen here. Most of the power goes to heat the fluid. Minimum flow formulas based on heat and energy balances can predict the flow which will produce a given temperature rise for an application. A more serious problem is mechanical in nature, due to shaft deflection. Impellers are designed to have balanced radial loads close to the best efficiency point. As one moves back along the curve, unbalanced radial loading increases, causing the shaft to deflect in a downward direction, away from the discharge nozzle. This load is transmitted to the inboard bearing. For these reasons, operation at less than 10% of flow at best efficiency is not recommended for most pumps, with higher limits for certain other pumps. For example, the 2K6X4-13A at 1750 RPM should not be operated at less than 50% of "best efficiency" flow.

A design point to the left of region "A", as opposed to short term operation, is a sign that the pump is too big for the job. The energy efficiency will be low, and the motor will have to be sized to be non-overloading at the end of the curve. A smaller pump and motor will cost less to buy, and less to operate.

Operation at the far end of the curve is also undesirable. Efficiencies will be low, and discharge velocities will be excessive. Recovery of the energy in the discharge will be difficult, and the pump will appear to be operating at a point below the published curve. Let me give an example. I was asked to examine an 8X6-14A pump in pond water service, which was operating well to the right of the best efficiency point. As in often the case in plant situations, precise flows and pressures were not available, but it was clear that the flow was over 100 gallons per minute too low, and the pressure at the spray header was inadequate. The pump was discharging to an 8" pipe through a short 6X8 reducer. Before any changes were made to the pump, the standard reducer was replaced by a long fabricated reducer. This simple piping modification caused the energy in the discharge to be recovered in an orderly fashion, rather than being dissipated as a submerged jet. Flow through the pump increased by approximately 150 gallons per minute, and no other changes were made to the system.

Problems with shaft deflection also occur to the right of the best efficiency point, but the actual deflection will be in the opposite direction. The operation will tend to be noisy, and there will be an increased tendency for the pump to cavitate. None of this is good for the bearings or seal, of course. All of these problems are worse with larger pumps and higher rotational speeds. I never fully appreciated the problem of operation at the end of the curve until I stood next to a 2K4X3-13 as it tried to pump 900 gallons per minute. The problem turned out to be simple. The pump was equipped with an open bypass line to avoid minimum flow problems, and an orifice had been left out during construction.

To conclude our tour of the curve, let's look at the immediate boundaries of Region "A". A design point to the immediate left of the region may be covered more efficiently by a smaller pump. The additional cost of a selection in this area might be justified if there is a good chance that the flow will increase in the future, and you want to avoid creating a plant bottleneck. A design point to the right of the best efficiency point might still be a good selection if the pump will actually be operating at a lower rate most of the time. A design point on the maximum impeller diameter leaves you with no place to go if it turns out after start-up that you need more pump. The risk of such an error will be minimized the more you know about the hydraulics of the system. I'll be talking about this some more when we look at system curves. Finally a fully trimmed impeller indicates that you might want to consider a smaller pump.

THE SYSTEM CURVE

The other half of the problem is the system curve. The operating point is determined by the intersection of the pump curve and the system curve. The pump manufacturer is kind enough to furnish the pump curve, but you have to come up with the system curve yourself. This means that you must first have a clear idea of the piping system at both ends of the pump. Then you must calculate what pressure drops the system will develop at flow rates within the operating range. Unless you have done this, rational selection of an economical pump is a practical impossibility. Of course, there is nothing wrong with painting with a broad stroke when you are in the budget stage. The purchasing process is another matter entirely. I have seen a request for quotation for nearly a dozen stainless steel pumps, and every one had a required TDH of 70.0 feet. The problem here is that most of the applications were simple transfer operations, and 8" pumps would probably been sufficient, rather than the 10" pumps which the specification called for.

Another alternative to calculation is rough estimation, with safety factors thrown in to compensate for uncertainty. I once saw a simple transfer pump which required a full diameter thirteen inch impeller as a result of pyramiding of safety factors.

The calculation of the points on the system curve requires first that the length and diameter of the pipe be specified, as well as the number and type of fittings, and the static head. The friction losses may be estimated by any of a number of techniques, ranging from the classic Fanning friction factor to the use of precalculated tables such as those in Cameron Hydraulic Data[1] or The Crane Handbook[2].

The Fanning friction factor is determined in hand calculations by the Moody chart[3], which in turn is a graphical representation of the Colebrook Equation. The problem with the Colebrook Equation is that it cannot be solved explicitly for f. Thus, it does not lend itself to direct solution using a hand calculator, or even a simple spreadsheet. The more recently published Chen Equation[4] expresses the friction factor f explicitly. It is too unwieldy for use with a hand calculator, but it is well suited for spreadsheet applications. Figure 3 shows a spreadsheet using this equation. The math in this spreadsheet has been checked against an iterative BASIC program based on the Colebrook equation, with excellent results.

This spreadsheet was developed in Quattro Pro, and has been saved in Quattro Pro, Quattro, Lotus 1-2-3, and Supercalc formats. Copies will be made available by the author upon request.

Why do I advocate developing a complete system curve instead of just looking at one point? First, the points when plotted against a copy of the pump curve, will give a better feel for the selection. Second, using a computer to generate a system curve will allow you to test the sensitivity of the system to your assumptions. A spreadsheet model will let you do this rapidly and interactively. It will, in my opinion, allow you to enhance the quality of your work. By quality of work, I mean the ability to produce a sounder design because you have been able to look at more alternatives in a short time. The ability to do a five-minute calculation in ten seconds is trivial and insignificant compared with the ability to do better work.

CAVITATION

Cavitation is one of the least-understood problems in pump hydraulics. It should not be confused with air ingestion, which causes similar problems, but has different causes, and different cures. When I speak of cavitation, I refer to the phenomenon which occurs when the NPSH requirement of a pump is not met. The fluid being pumped vaporizes near the eye of the impeller, and the bubbles of vapor collapse violently. A pump in cavitation will be noisy and may pound and vibrate. Cavitation produces a characteristic rattling noise, which sounds as though the pump is pumping gravel or marbles. This will damage the pump by shock loading the bearings, wrecking the mechanical seal, and removing pieces of metal from the impeller itself. Certain design situations have an inherent danger of cavitation, and you should be aware of these. They include pumping liquids close to the boiling point, pumping under reduced pressure, pumping liquids with high vapor pressure, and long or restricted suction lines. In addition, practically any pump in the best designed installation can be made to cavitate by throttling the suction line. **Never throttle the suction of any pump.**

AIR INGESTION

Air ingestion by pumps is one of the most common causes of poor pump performance, and one of the most difficult and expensive to correct. The gas phase enters the pump from outside, rather than being formed within the impeller. This, and the fact that the bubbles do not collapse violently inside the pump, distinguishes air ingestion from cavitation. The air usually enters via the pump suction, but in systems under reduced pressure, air can leak in at the seal or the discharge.

Theories abound as to the mechanism whereby entrained air interferes with pump performance. I have heard, for example, that a persistent mass of bubbles will form at the eye of the impeller, or against the blades of the impeller. It is clear that a gallon of air passing through a pump will take the place of a gallon of process fluid. That same gallon of air will also reduce the specific gravity within the pump, and will reduce the pump's discharge pressure as measured in absolute terms. Both of the preceding statements are true as far as they go, but they do not take into account the fact that centrifugal pumps can develop extremely low pressures. A small bubble passing through a region of low pressure, will expand to satisfy the perfect gas law. For a pump with a high Required NPSH, this expansion can be over five times the original volume at atmospheric pressure. Whatever the actual mechanism, air will ruin the performance of a pump. The metal damage associated with cavitation will not occur, but the vibration associated with out of balance operation will occur, and shortened bearing and seal life will be the result.

Two ways that air can enter a pump are vortexing and entrainment. If the end of the suction line is too close to the surface, a vortex will form, which will pull air directly into the suction. Air can also enter from a process vessel which contains entrained foam, or bubbles which have not had sufficient time to disengage from the fluid. In both cases, these problems can be expensive to correct. A scrubber, which is already large and expensive, may turn out not to be large enough to allow the air to disengage from the scrubber liquor. Simply adding a few feet of elevation to the liquid level above the pump suction may cost in the thousands of dollars an inch if the elevation of the vessel controls the elevation of the upper level of the plant.

SELF-PRIMERS AND FAILURE TO PRIME

The self-priming pump is a special case of a pump which is designed to cope with air. It starts with a load of water in the priming chamber, and recirculates this water as it exhausts the air from the suction line. The air has to go someplace, or the pump will never develop enough discharge pressure to establish flow in the discharge line. If there is a check valve in the discharge line, there may not be enough pressure to lift the check valve. The solution to this problem is to install a small vent line near the pump discharge, to allow the air to leave the system. Another place to look for problems with a self-priming pump is in the discharge line. If the line has many "ups" and "downs", the line may become air bound. Imagine a pump connected to a long discharge line, with three 20-foot loops for truck passages, pipe bridges, or the like. Under worst-case conditions, the pump will be pushing against 60 feet of static head, which may not have been planned for.

USE OF COMPUTER PUMP SIZING

A recent development in pump application engineering is the pump selector computer program. Some of these have been developed by pump manufacturers, and some are commercial offerings. The Duriron Company has developed Pumpsel, which I shall describe briefly. The user fills in a questionnaire on a data input screen, and uses a function key to select a list of feasible pumps. The solution screen lists the candidate pumps in order of efficiency. A typical questionnaire screen and solution screen are shown as Figures 4 and 5. Normally the user would use the print routine, as in Figure 6, rather than printing the screen directly.

The output from the program should always be checked against the curve book, and the final selection should rely on human judgment. The computer may omit a perfectly reasonable selection for a technical reason. If it does, it will not tell you what that reason was.

Whether the selection is made by hand or by machine, the quality of the result can be no better than the quality of the data. We should be mindful of the fact that the acronym GIGO applies to hand calculations as well as computers.

REFERENCES

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2. Heald, Charles, ed., Cameron Hydraulic Data, Ingersoll-Rand, Woodcliff Lake, NJ, 17th Edition, 1988.

3. Moody, Trans. Am. Soc. Mech. Engrs., 66, 671 (1944).

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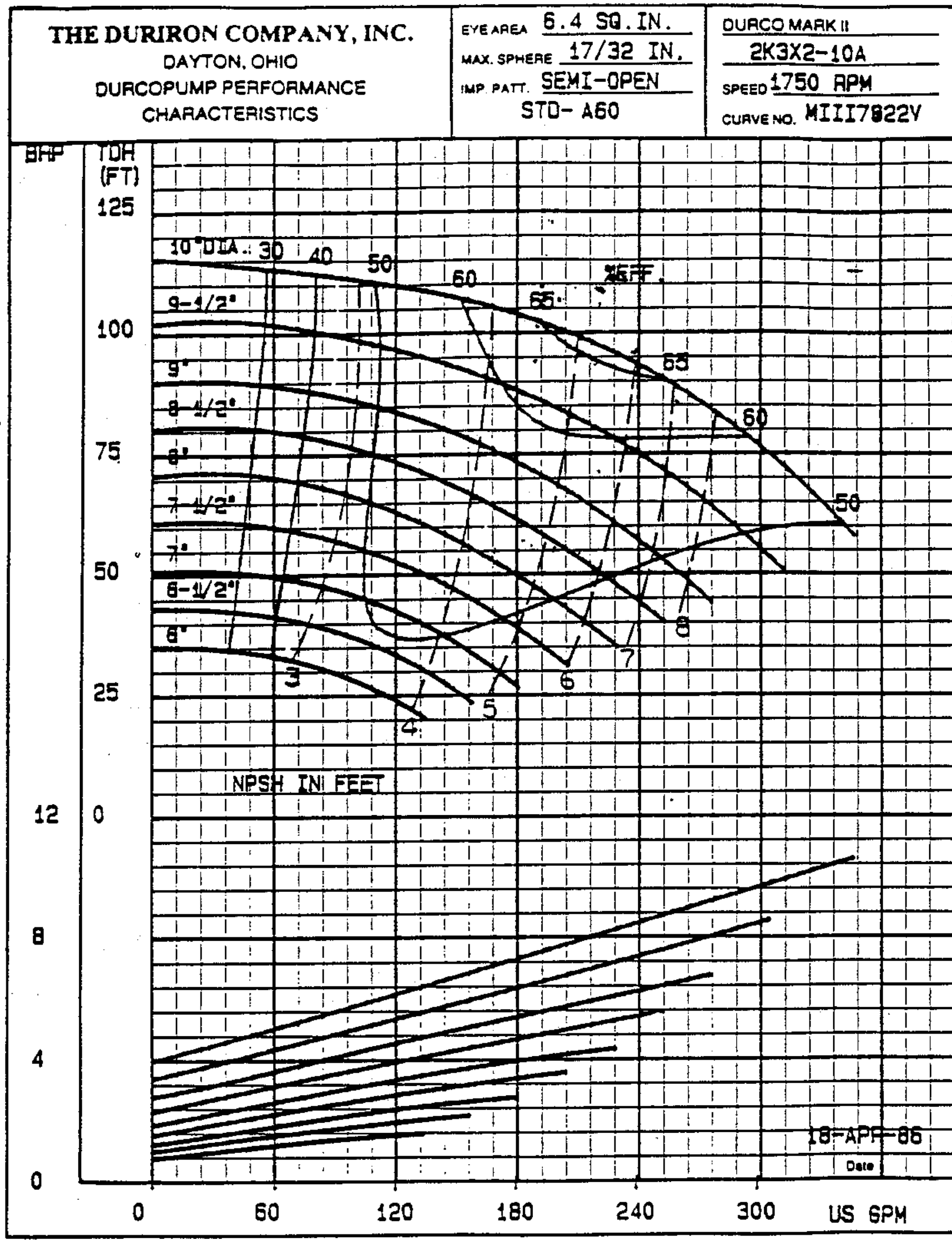
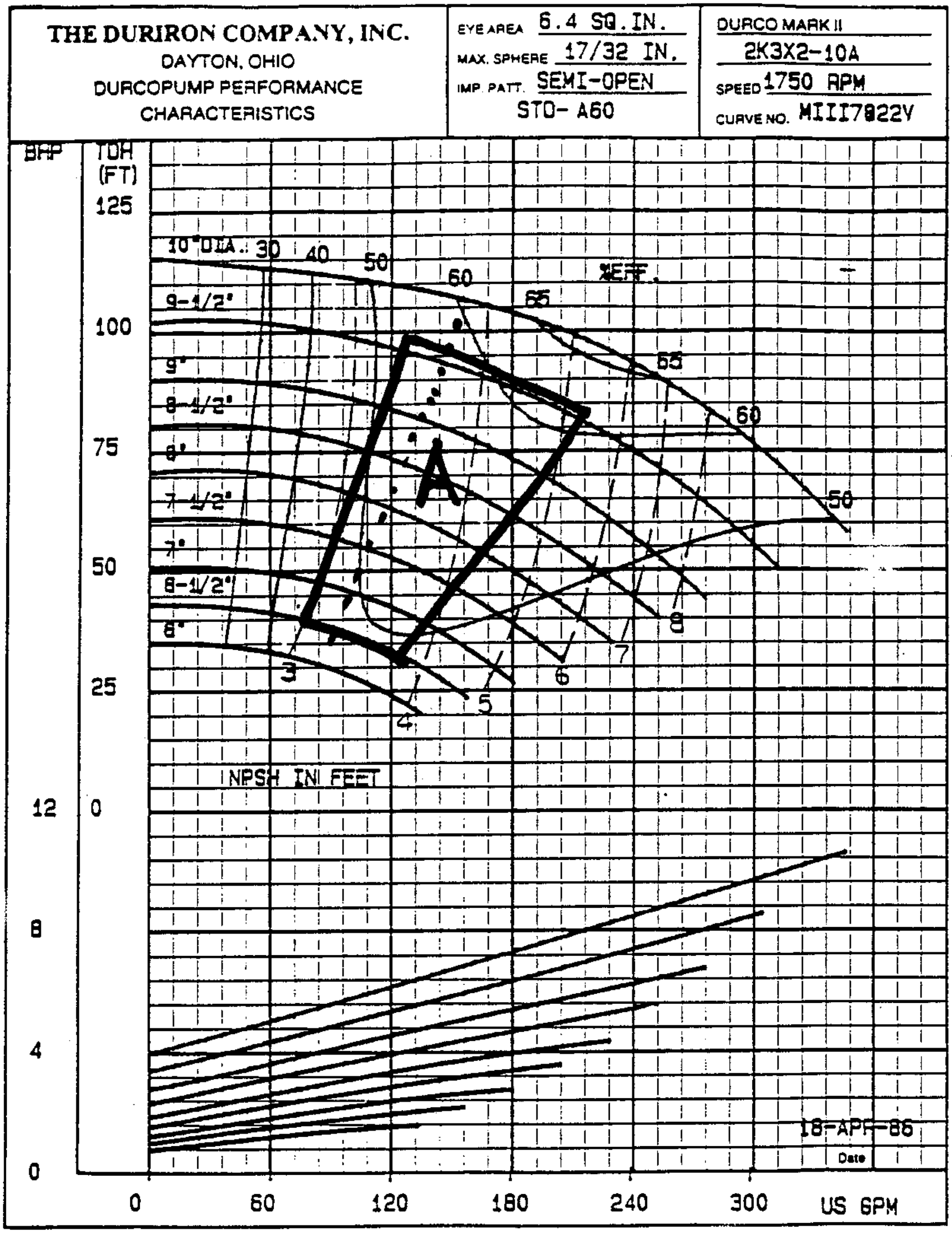

THE DURIRON COMPANY, INC.
DAYTON, OHIO
DURCOPUMP PERFORMANCE
CHARACTERISTICS
EYE AREA 6.4 SQ. IN.
MAX. SPHERE 17/32 IN.
IMP. PATT. SEMI-OPEN
STD- A60
DURCO MARK II
2K3X2-10A
SPEED 1750 RPM
CURVE NO. MIII7822V
BHP
TDH (FT)
125
100
75
50
25
0
12
8
4
0
10" DIA.
9-1/2"
9"
8-1/2"
8"
7-1/2"
7"
6-1/2"
6"
30
40
50
60
65
%EFF.
65
60
50
3
4
5
6
7
8
NPSH IN FEET
18-APR-86
Date
0
60
120
180
240
300
US GPM

Figure 1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BHP
TDH (FT)
125
100
75
50
25
0
12
8
4
0
10" DIA.
9-1/2"
9"
8-1/2"
8"
7-1/2"
7"
6-1/2"
6"
30
40
50
60
65
%EFF.
65
60
50
3
4
5
6
7
8
A
NPSH IN FEET
18-APR-86
Date
0
60
120
180
240
300
US GPM

Figure 2

12:18	04/08/91		
d,in.	3	Visc.,cp	1
D, ft.	0.25	SpG	1.05
e, ft.	0.00015	Step	20
L, ft.	423		
Sum of Ks	6		
Static head, Ft.	35		

GPM	Hf	V	Hv	TDH	PSI
20	0.58	0.91	0.01	35.65	16.21
40	2.03	1.82	0.05	37.34	16.97
60	4.28	2.72	0.11	39.97	18.17
80	7.32	3.63	0.20	43.55	19.79
100	11.13	4.54	0.32	48.05	21.84
120	15.71	5.45	0.46	53.46	24.30
140	21.05	6.35	0.63	59.80	27.18
160	27.15	7.26	0.82	67.05	30.48
180	34.01	8.17	1.03	75.21	34.19
200	41.63	9.08	1.28	84.29	38.31
220	50.00	9.99	1.55	94.28	42.85

12:19	04/08/91		
d,in.	3	Visc.,cp	1
D, ft.	0.25	SpG	1.05
e, ft.	0.00015	Step	5
L, ft.	423		
Sum of Ks	6		
Static head, Ft.	35		

150	Hf	V	Hv	TDH	PSI
155	25.55	7.04	0.77	65.15	29.61
160	27.15	7.26	0.82	67.05	30.48
165	28.79	7.49	0.87	69.01	31.37
170	30.48	7.72	0.92	71.02	32.28
175	32.22	7.94	0.98	73.09	33.22
180	34.01	8.17	1.03	75.21	34.19
185	35.84	8.40	1.09	77.40	35.18
190	37.72	8.62	1.15	79.64	36.20
195	39.65	8.85	1.21	81.94	37.24
200	41.63	9.08	1.28	84.29	38.31
205	43.65	9.30	1.34	86.70	39.41

Figure 3

SYSTEM CURVE SPREADSHEET
(Two runs)

```
                           Pump Selection

 Purchasing Company:  NRFPT, INC.
 Company Address:     WIMAUMA, FLORIDA

 Pump Line:           MARK III (Rev. Vane)

 Pump Service:        PROCESS LINE 85

 CYCLE:               60                   Cycle 60
                                        1 - Any
 RPM:                 1750              2 - 3500
                                        3 - 1750
 Capacity:  (GPM)        185            4 - 1150
                                        5 - 880
 TDH:   (FT)              78
                                        Enter a number:
 Viscosity: (CP)           1     NPSH

 Specific Gravity:      1.05     BEP Desired:          Any

F1 - Select                        F8 - Set Up F9 - Help F10 - Quit
```

Figure 4

PUMPSEL INPUT SCREEN

Durco Pump Selection

SELECTED PUMPS ARE:

	PUMP	RPM	IMPELLER	HEAD (FT)	HP	END HP	NPSH(FT) REQUIRED	%EFF
1.	2K3X2-13	1750	9 3/8	78	5.9	7.3	3.4	65
2.	2K3X2-10A	1750	9 1/4	79	6.3	8.1	4.7	62
3.	2K4X3-10	1750	9	79	6.7	10.1	1.0	58
4.	2K4X3-13HH	1750	9 1/8	79	6.9	10.3	3.1	56
5.	2K3X1-1/2-13	1750	9 1/2	80	7.1	8.0	3.3	56
	OPTIONAL PUMP(S)							
6.	1K3X2-6	3500	5	79	5.1	5.6	7.5	76
7.	2K4X3-13HH	1150	12 7/8	80	7.3	13.5	1.8	54

Note: For guaranteed NPSHR values, a tolerance of 0.5 Ft. should be added to PumpSel values due to manufacturing variations. See The PumpSel User's Guide for more details.

F1 - Print F2 - Plot F3 - Return F9 - Help F10 - Quit

Figure 5

PUMPSEL OUTPUT SCREEN

THE DURIRON CO. INC.
DAYTON, OHIO
PUMPSEL - VERSION 4.2

PUMP SELECTION FOR:

DATE: 4/7/1991

NRFPT, INC.
WIMAUMA, FLORIDA

PUMP LINE:

MARK III (Rev. Vane)
60 CYCLE

PUMP SERVICE:

PROCESS LINE 85

OPERATING CONDITIONS:

GPM = 185
TDH(FT) = 78
VISCOSITY (CENTIPOISE) = 1
SPECIFIC GRAVITY = 1.05

AT 1750 RPM

PUMP	RPM	IMPELLER	HEAD (FT)	HP	END HP	NPSH(FT) REQUIRED	%EFF
2K3X2-13	1750	9 3/8	78	5.9	7.3	3.4	65
2K3X2-10A	1750	9 1/4	79	6.3	8.1	4.7	62
2K4X3-10	1750	9	79	6.7	10.1	1.0	58
2K4X3-13HH	1750	9 1/8	79	6.9	10.3	3.1	56
2K4X3-13	1750	9 1/8	80	7.0	12.1	1.5	56

OPTIONAL PUMP(S)

1K3X2-6	3500	5	79	5.1	5.6	7.5	76
2K4X3-13HH	1150	12 7/8	80	7.3	13.5	1.8	54

Note: Due to manufacturing variations, for guaranteed NPSHR values, a tolerance of 0.5 Ft. should be added to PumpSel values. See PumpSel User's Guide for more details.

Figure 6

PUMPSEL PRINTED OUTPUT