POWER RECOVERY TURBINES FOR THE PROCESS INDUSTRY

by

S. Gopalakrishnan

Manager, Research and Development

Byron Jackson Pump Division

Borg Warner Industrial Products, Incorporated

Long Beach, California

S. Gopalakrishnan is Manager of Research and Development for the Byron Jackson Pump Division of Borg Warner Industrial Products, Incorporated, in Long Beach, California.

He obtained the Doctorate in Mechanical Engineering from the Massachussetts Institute of Technology in 1969.

Dr. Gopalakrishnan has been responsible for advanced research in the areas of fluid mechanics and rotordynamics of turbomachinery. His work at AVCO Lycoming (1969-73) culminated in the development of a numerical technique for transonic flow calculation in turbines and compressors. His work at the Borg Warner Research Center (1973-78) was related to pumps, compressors and torque converters. He has been with Byron Jackson since 1978, and is presently responsible for the development of new products for Byron Jackson and the coordination of research activities carried out by Byron Jackson affiliates worldwide.

Dr. Gopalakrishnan serves on the Advisory Committee for the International Pump Symposium.

ABSTRACT

Pumps running in reverse as turbines are often used for energy recovery applications in the hydrocarbon processing industry. Empirical methods for predicting turbine performance from known pump performance are reviewed. Since in the majority of applications, gas laden fluid streams have to be handled, the problems of gas evolution in the turbine are reviewed next. Based on the findings of a recent study of two-phase flow in turbines, a new high speed turbine has been built and tested. The design features and the performance of this turbine are reported.

INTRODUCTION

There are many instances in the hydrocarbon industry where the processing of a fluid stream requires its pressure to be reduced. This pressure reduction is usually accomplished through the use of a throttling valve. In this method, the energy of the fluid stream is lost. Currently, emphasis is being placed on more effective energy usage in the processing industry. As a consequence, areas in which energy is wasted are being closely monitored and methods for energy recovery are being investigated. It is being recognized increasingly that replacement of throttling valves with turbines can effectively recover a large percentage of the available energy with acceptable first cost.

Typically, reverse running pumps are often used as power recovery turbines. Even though in many applications this offering is satisfactory, typical reverse running pumps do not provide the anticipated performance. Such situations often arise when the operating flow conditions change, or when there are significant volumes of gas to be handled along with the liquid. For such cases, new turbine designs are needed.

The applications in the hydrocarbon processing industry where power recovery turbines are applicable are described herein. Next, methods for selection of reverse running pumps as turbines are described. Elementary payback analysis will be included to show the cost effectiveness of turbines. The results of a recent, extensive study of the phenomena associated with gas evolution will be briefly described. This will include a description of the analysis technique, with a comparison of the theoretical values to actual measurements on bubble growth rates in a turbine design. Such a turbine has recently been built and tested at Byron Jackson. The features of this machine and its performance will be shown. This turbine is presently awaiting installation at a refinery in Kentucky.

APPLICATION AREAS

There are three application areas where power recovery turbines are suitable. When hydrocarbon liquids are refined by hydrotreating, hydrocracking, etc., the liquid is treated with hydrogen at high pressures and temperatures. The resulting liquid at high pressure contains dissolved gases like methane, ethane, etc., which tend to evolve when the pressure is reduced. A schematic diagram showing how a turbine can be used to generate useful power in such a system is featured in Figure 1. In this figure the turbine is shown as being used to unload the

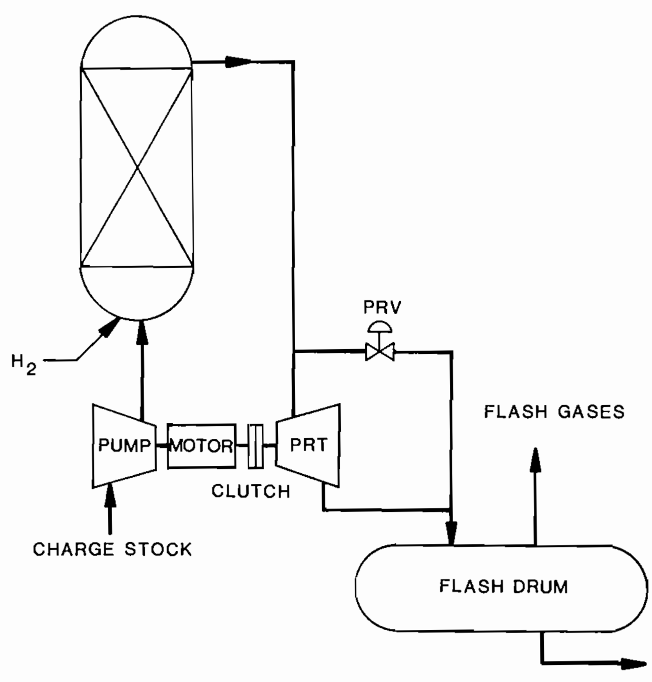

Figure 1. Schematic of Hydrotreating Process.

motor that drives the charge pump. In other instances, the turbine can be mechanically uncoupled from the pumping system and be used to directly drive a generator feeding the power grid. In the case shown in Figure 1, a one-way clutch is needed to prevent the turbine from overloading the motor when the flow conditions are inadequate. Indeed, it is now known that many operating problems have been due to the lack of this key feature. Also, when the turbine is used to directly drive a generator, precautions must be taken to assure mechanical reliability in the event of a power failure. Under such conditions, the turbine will overspeed very quickly and may also cause water-hammer in the piping system. These eventualities must be carefully controlled.

The second application area is in the scrubbing of natural gases, which may contain unacceptable quantities of impurities like carbon dioxide. The application is shown schematically in Figure 2. Here a liquid like monoethanolamine or diethanolamine is used to scrub the unwanted gases from the marketed gas. The lean amine entering the reactor vessel absorbs CO_2. The exiting rich amine solution can be used to drive a turbine. The turbine mechanical operation is similar to the previous application.

The third application area is in synthetic ammonia manufacturing. The system involving the turbine is very similar to the gas processing application mentioned previously.

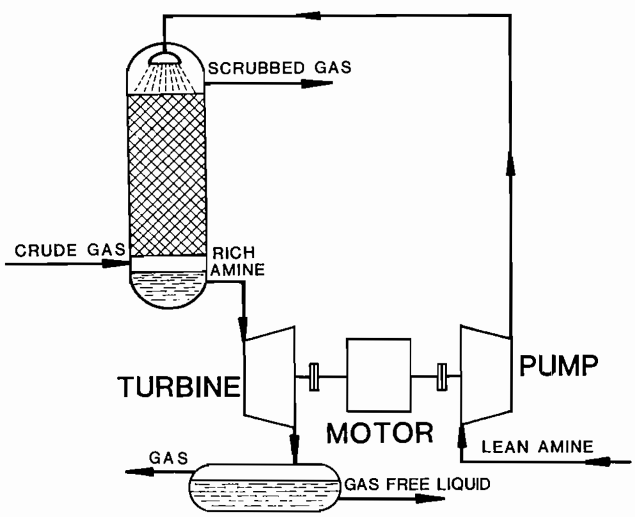

Figure 2. Schematic of Gas Scrubbing Process.

The head and flow available for the turbines in these applications are shown in Figure 3. For refining applications, when high pressures are involved, a double case pump is commonly used. For natural gas scrubbing, split case multi-stage pumps are commonly selected. For synthetic gas manufacturing applications, low pressure, high volume conditions naturally lead to selection of single-stage, double-suction, between-bearing pumps. Single stage overhung process pumps may also be suitable when moderate pressures and volumetric flows are involved.

The power recovery that is feasible is also shown in Figure 3. The constant kW contours are drawn for a combined turbine-generator efficiency of 0.7. It can be seen that a potential recovery of 150 kW to 1500 kW exists.

PAYBACK ANALYSIS

A simplified payback analysis can be performed to check the financial viability of a power recovery turbine installation. The computation listed in Table 1 is for a typical case involving a

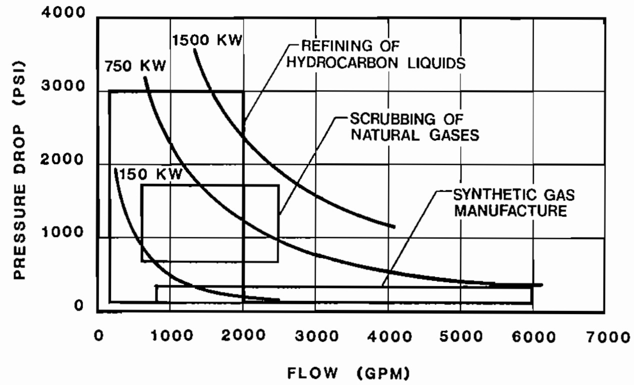

Figure 3. Application Range for Turbines in the Hydrocarbon Processing Industry.

generation of 527 kW. The values of the dollar savings are computed for 8000 hr annual operation and six cents per kW-hr electricity cost. The first cost to the user includes the cost of the turbine and installation costs, which may vary considerably from site to site. The $250,000 shown in Table 1 is considered to be typical. On this basis, the turbine can be seen to pay for itself in less than two years. Such a payback period would be considered acceptable by most users. It should be noted that this analysis does not include the effects of accelerated depreciation, energy tax credits, or maintenance-related expenditures. A more thorough analysis including these items can of course be made, but is beyond the scope of this study.

Table 1. Power Recovery Turbine Payback Analysis.

OPERATING CONDITIONS	
1. Total Liquid Flow	1,000 gpm
2. Total Pressure Drop	4,000 ft
3. Fluid Specific Gravity	0.8
4. Efficiency PRT	0.8
5. Power Recovered	485.
6. Annual KW Savings	232,000 (8,000 hr)
$\frac{(1)\times(2)\times(3)\times(4)}{5{,}310.5}$ KW	
$(5)\times 8000\times .06$	
EQUIPMENT COST	
7. Power Recovery Turbine, Clutch, Gear	$170,000
8. Cost of Installation	$250,000
9. Total Installed Cost (7) + (8)	$430,000
Payback Period 2 Years	

PERFORMANCE PREDICTION

Since in the majority of applications, the turbine is a pump running in reverse, attempts have been made to predict the turbine performance from the known pump performance. In these methods, the turbine flow and head at the best efficiency point (BEP) are predicted from the known head and flow at the BEP of the pump. If the pump specific speed is used as a correlating parameter, curves of the type shown in Figures 4 and 5 can be derived from known test data. The ratio of the pump flow to the corresponding turbine flow at their respective BEP, plotted as a function of the pump specific speed is depicted in Figure 4. Similarly, the corresponding head ratio is shown in

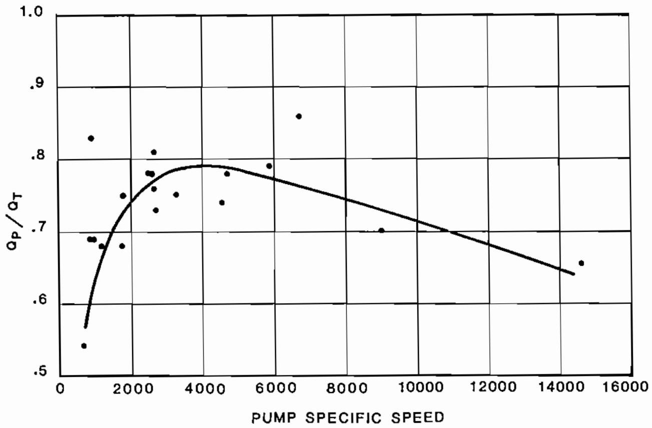

Figure 4. Pump to Turbine Flow Ratio at BEP as a Function of Pump Specific Speed.

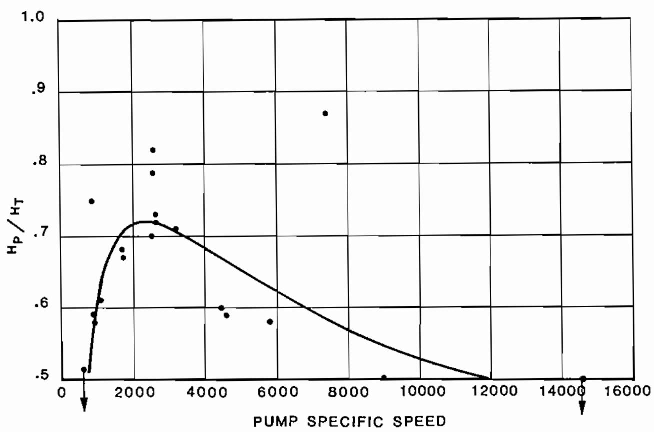

Figure 5. Pump to Turbine Head Ratio at BEP as a Function of Pump Specific Speed.

Figure 5. The dots show data from actual tests of a number of pumps. A curve is drawn through test points to indicate a hypothesized trend. It may be observed that this performance trend of first increasing to a maximum and then decreasing, is similar to the pump efficiency-specific speed trend. Consequently, these ratios may be expected to vary with efficiency.

Stepanoff suggests that the head ratio will equal the efficiency and the flow ratio will equal the square root of the efficiency [1].

$$\frac{H_p}{H_T} = \eta_p$$

$$\frac{Q_p}{Q_T} = \sqrt{\eta_p}$$

where

H_p = pump head
Q_p = pump flow
H_T = turbine head
Q_T = turbine flow
η_p = pump overall efficiency

All quantities are evaluated at their respective BEP. It is assumed that turbine and pump efficiencies will be nearly the same.

Thorne assumes that the power input to the pump and the power output from the corresponding turbine are likely to be the same, and the efficiencies are also about the same [2]. He then concludes that both head and flow ratios will equal the efficiency.

$$\frac{Q_p H_p}{\eta_p} = Q_T H_T \eta_T$$

$$\frac{Q_p}{Q_T} = \frac{H_p}{H_T} = \eta_p = \eta_T$$

where

η_T = turbine efficiency

The flow ratio plotted as a function of the pump efficiency for a number of pumps tested both as pumps and turbines is shown in Figure 6. The lines in this figure represent the flow ratio varying as the efficiency and as the square root of the efficiency.

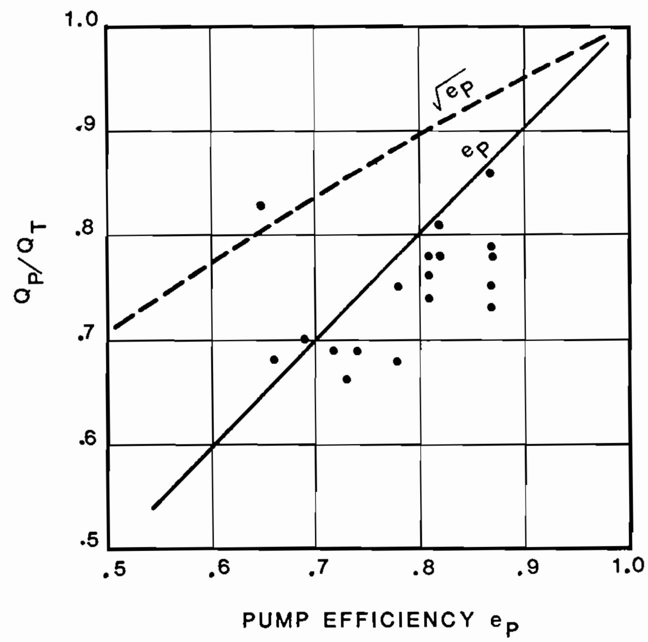

Figure 6. Flow Ratio as a Function of Pump Efficiency.

It can be seen that the square root variation is not valid for most of the test points. Variation with the efficiency is closer to test data. A similar plot for the head variation is shown in Figure 7. The variation with the square of the efficiency seems to be more in keeping with the test data. Other methods also exist for turbine performance prediction from known pump data [3, 4, 5].

As can be seen in Figures 4 through 7, the actual performance deviates rather substantially from any correlating equation. The assumption of pump and turbine efficiencies being equal is basically invalid. Efficiencies of a number of pumps plotted against their respective efficiencies when operated as turbines are shown in Figure 8. Clearly, there are significant discrepancies between the two efficiencies. Therefore, it will not be prudent to guarantee the performance of the turbine from known pump test data. Actual testing of the machine as a turbine is advisable.

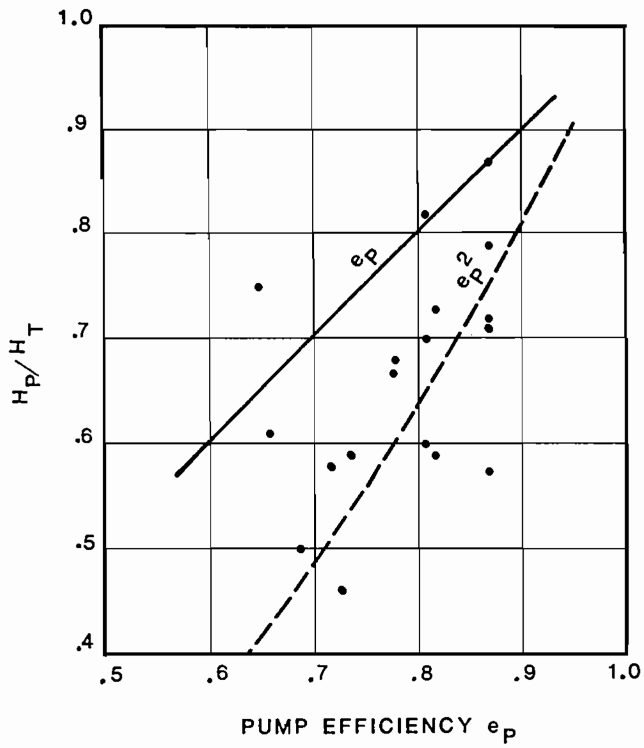

Figure 7. Head Ratio as a Function of Pump Efficiency.

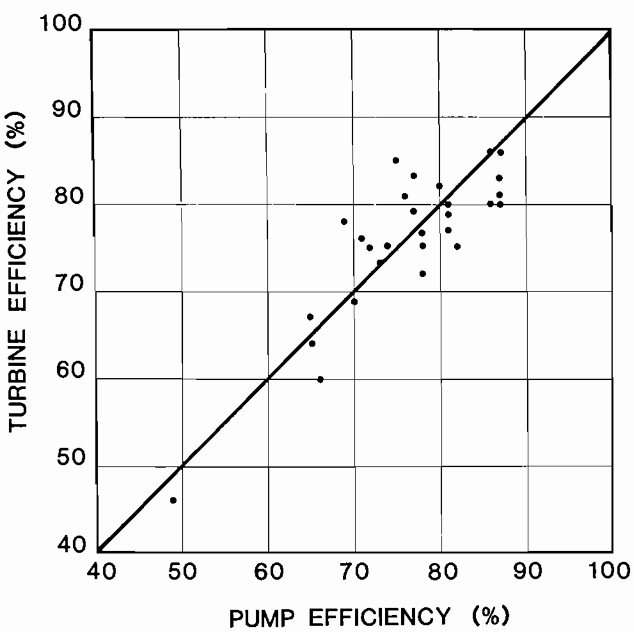

Figure 8. Turbine Efficiency Versus Pump Efficiency.

OPERATING PROBLEMS

In spite of the attractive payback features of turbines, they are not all that common in the industry. The main reason is that turbines often have not produced the energy savings initially anticipated. Reliability problems, due to the lack of a one-way clutch, have been mentioned earlier [6]. Due to difficulties in prediction, the turbine may not be generating the ouput that was expected. Further, when operation first begins, balancing the whole process is sometimes difficult and the presence of the turbine makes the startup process somewhat more involved. These problems are gradually being eliminated.

However, two problems still remain. Reverse running pumps generally have no adjustable features. Therefore, when operating conditions change, the equipment cannot track variations in flow conditions without significantly losing performance. For example, in Figure 9, which shows the performance of a fixed geometry turbine, reduction of available flow from 2000 gpm to 1000 gpm will reduce the power output from 870 kW to 60 kW! With adjustable geometry machines, as will be shown later, efficiency can be maintained.

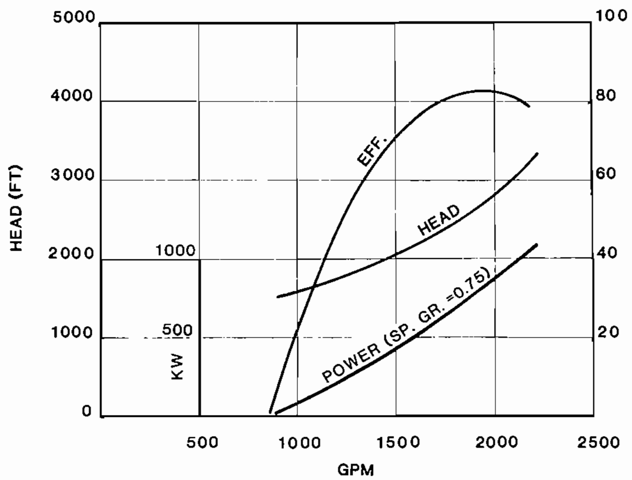

Figure 9. Typical Characteristics of Pumps Running in Reverse as Turbines.

The second key problem is related to gas evolution. As has been remarked in reference to Figures 1 and 2, nearly all hydrocarbon processing industry (HPI) applications involve gas-laden liquid streams. When the pressure is reduced in the turbine, large volumes of gas come out of solution. In principle, expansion of gas in the turbine actually increases the turbine ouput. The contribution due to gas expansion based on a

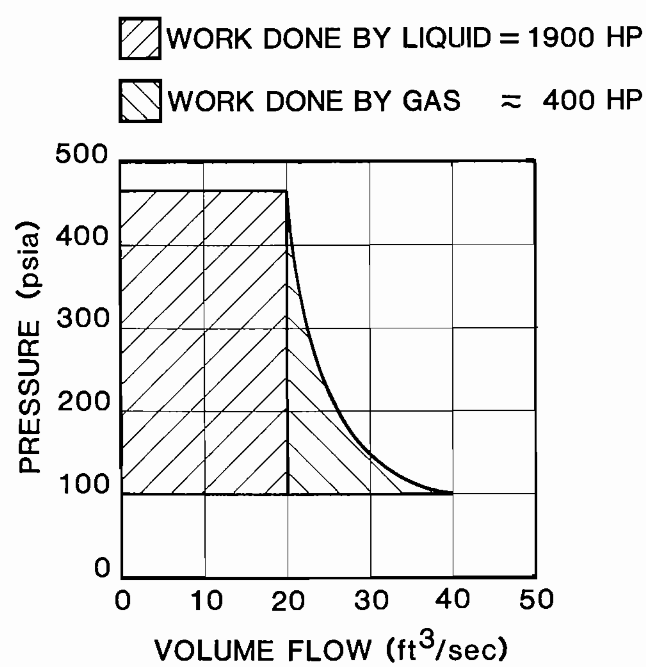

Figure 10. Theoretical Work Due to Gas Expansion.

theoretical calculation is reflected in Figure 10. The liquid specific volume is assumed to be constant at 0.02 ft^3/lb and the gas specific volume at inlet is taken to be 0.43 ft^3/lb. The gas expansion is assumed to follow the perfect gas law and the change in temperature during expansion is disregarded. The contribution by the initial gas volume, which is presumed to be contained in the molecular interstices of the liquid, is ignored. For a gas ratio by weight of one percent at the discharge, gas to total ratio by volume is 50 percent. Under these conditions, it may be expected on theoretical grounds that the gas expansion produces nearly 20 percent of the work done by the liquid.

In an actual fixed geometry turbine, very little of this available gas horsepower (hp) may ever be recovered. Under two-phase flow, the actual turbine characteristic will change, because of the changing volume of the flow. The turbine characteristics are more easily represented on a pressure versus mass flow basis, instead of the usual head versus volume flow basis. The computed characteristics for a single stage double suction pump running in reverse are shown in Figure 11. Two-phase flow characteristics have been computed by assuming that thermodynamic equilibrium exists everywhere. As can be seen from Figure 11, plotted for 0.6 percent of gas by weight, the mass flow that can be accommodated by the turbine for a given pressure drop decreases because of the increased volume. For the design pressure drop of 350 psi, the capacity of the turbine decreases from 1350 lb/sec to 1200 lb/sec when gas evolution takes place. The net HP generated, therefore, actually slightly decreases.

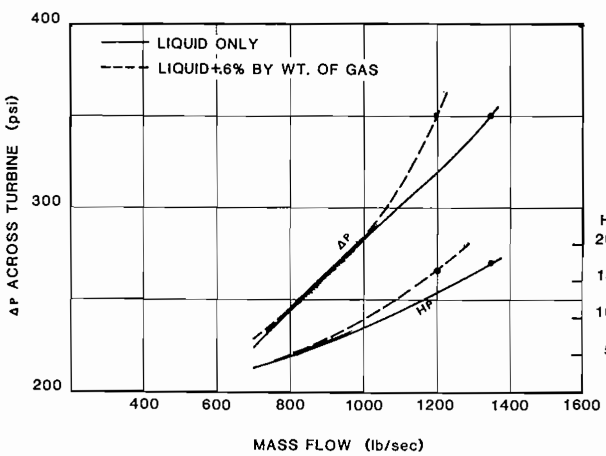

Figure 11. Predicted Characteristics in Single and Two-phase Flow of a Single Stage, Double Suction Pump Running in Reverse as a Turbine. (16 in diameter impeller, 2950 cpm, rich selexol solution with CO_2 evolution).

Apart from the preceeding issue of hydraulic performance, gas evolution can cause serious reliability problems. Multistage pumps rely upon the stiffening effect of close clearance, annular running fits to provide high critical speeds. These fits are the wear rings, balance sleeves, center stage piece, etc. Results show that without the stiffening effect, critical speeds are well below the operating speed, whereas, when these effects are included, critical speeds can become infinitely large [7]. In the turbine, wear ring gaps may get filled with the evolved gases when the gas volume to total volume ratio at the discharge becomes large (e.g., greater than 50 percent). Thus, the wear ring stiffening effect can disappear and bring the critical speed down to or below the operating speed. Increasing deflections can eventually cause rubbing at the fits.

In order to develop a design solution for the gas evolution problem, it is necessary to understand the mechanics of the gas evolution process.

TWO-PHASE FLOW STUDIES

An extensive theoretical and experimental study was conducted at the Borg Warner Research Center to estimate the void fraction at the outlet of hydraulic turbines, when the inlet conditions are specified. The details of this study are reported in [8]. For the present investigation, only the highlights of the study will be described.

When the liquid containing dissolved gases is rapidly depressurized in the turbine, the gas bubbles initially present grow in size due to the drop in pressure and the diffusion of the gas from the bulk fluid into the gas bubble. The rate at which pressure is decreasing is established *a priori* through single phase analysis of the turbine flow. After computation of the outlet void fraction, one could recalculate the pressure time history and solve for the void fraction again. Such an iterative calculation has not yet been done. As for the mechanism of bubble generation, information relating to the bubble frequency probabilities and the number of favorable sites for bubble generation, as reported in the literature of nucleate boiling, is used. The mathematical analysis results in an integral equation for the bubble radius at any given time as a function of its size at an earlier time in a pressure field, following the given pressure time history. The integral equation is solved iteratively.

To verify the theoretical model, a test appratus was constructed. This consisted of a cylindrical chamber which was initially filled with pure ethyl alcohol, saturated with CO_2 at the desired test pressure. A single stream of CO_2 bubbles was injected into the chamber, through an injection needle. A quick release valve was then opened and the chamber was depressurized in 50 to 120 milliseconds. A high speed camera operating at a speed of 3000 frames per second recorded the bubble stream. Projecting the movie on a large screen, the bubble diameters at any given time could be directly measured. The comparison of the results of the experiment at an initial pressure of 64 psi and initial

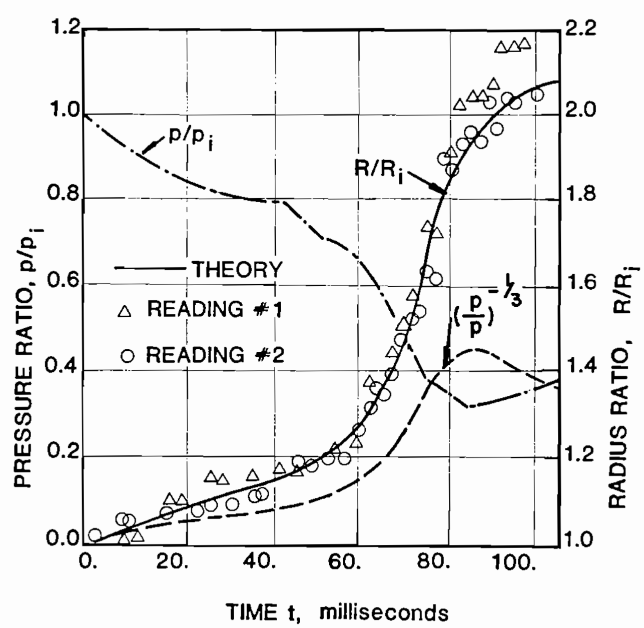

Figure 12. Comparison of Theoretical Experimental Bubble Growth Histories. Initial bubble radius 0.005 in, initial pressure 64 psi, ethyl alcohol CO_2 solution).

radius of 0.005 in, with the prediction s of the theoretical model is shown in Figure 12. The pressure variation during the test is shown as the curve marked p/pi. The curve marked $(p/p_i)^{-1/3}$ shows the expected bubble growth history, including the effect of mass transfer, which may be observed to be quite significant. The measured data are shown by symbols and are in excellent agreement with the predicted results.

The mathematical model was modified with some additional assumptions to calculate the bubble growth history in hydraulic turbines. The main assumptions are that the two-phase flow is homogeneous and one-dimensional, the pressure distribution is known *a priori*, and the gas volume rate increases because of

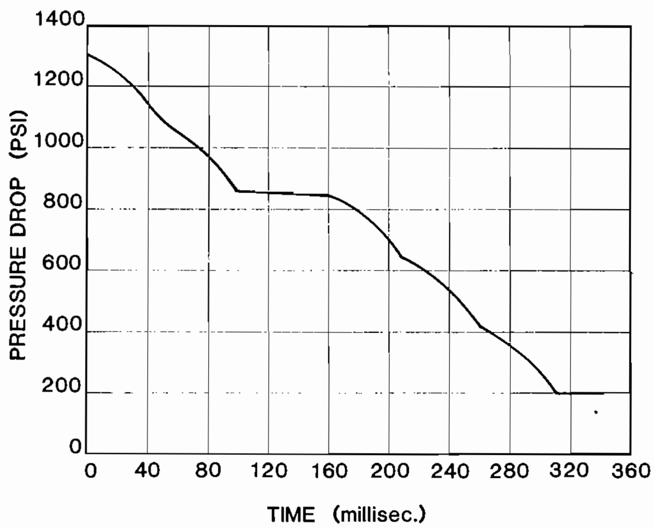

Figure 13. Pressure Variation in a five Stage Reverse Running Pump. (9 in diameter impeller, 3600 cpm, 800 gpm, 2600 ft head).

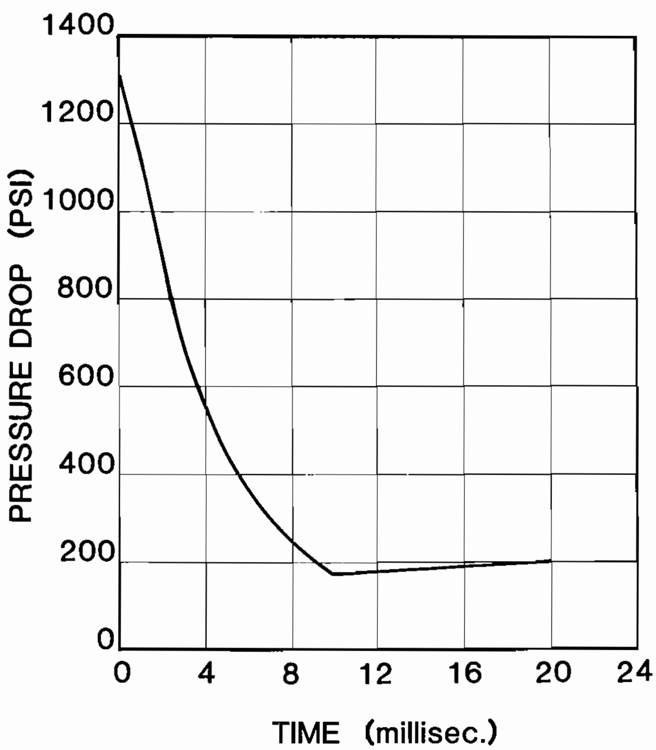

Figure 14. Pressure Variation in a Single Stage Turbine (8 in diameter runner, 6800 cpm, 800 gpm 2600 ft head).

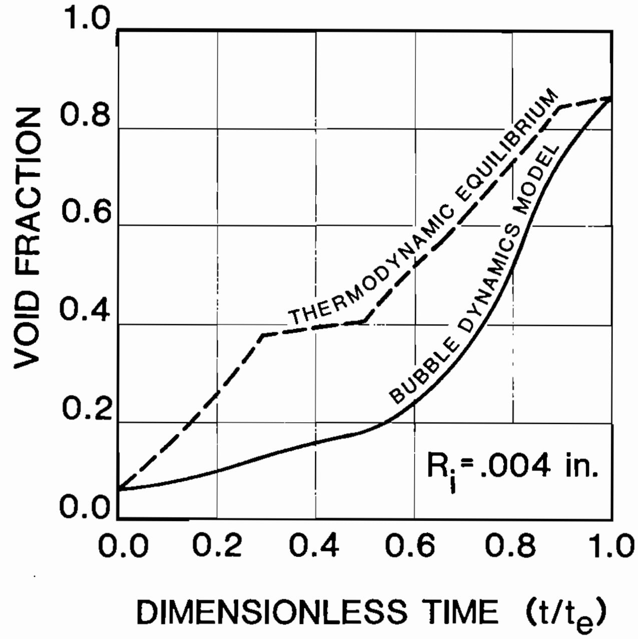

Figure 15. Comparison of Bubble Dynamics and Equilibrium Void Fractions for Five Stage Reverse Running Pump, (Monoethanolamine—CO_2 Solution, Inlet Void Fraction 0.07).

the growth of bubbles and because of the generation of new bubbles (and their growth) at all solid surfaces in contact with the fluid stream. A brief account of this calculation may be found in the Payvar study [8], but a more comprehensive report has yet to be published. The pressure time history in a five-stage reverse running pump with horizontally opposed impellers and a long crossover is shown in Figure 13. The history for a new single stage high speed turbine for the same pressure drop and flow is depicted in Figure 14. The features and performance of this new machine are described in the next section. It can be immediately observed that the residence time in the single stage machine is greatly reduced (from 340 milliseconds to 20 milliseconds). Assuming an inlet void fraction of 0.07 in both cases, the void fraction variations through the machines were calculated using the developed theory. The computed results are shown in Figures 15 and 16. It can be seen that for the reverse running pump, the final void fraction is very nearly the same as when thermodynamic equilibrium is assumed. This significant finding implies that for multistage machines the exit void fraction can be easily estimated by assuming that the specific volumes vary inversely as the pressures, since the temperature remains substantially constant in the typical cases of small gas to liquid mass ratios.

On the other hand, the exit void fraction for the single stage machine is significantly less than the value computed for thermodynamic equilibrium (Figure 16). Thus, less free gas would be present at the exit of the turbine than with the multistage machine, thereby circumventing some of the problems of reliability associated with gas evolution. The main reasons for this difference between the two design options are the greatly reduced residence time and solid surface area.

FEATURES OF NEW HIGH SPEED TURBINE

The basic consideration in the hydraulic design was to keep the residence time in the turbine as small as possible. A single stage design was adopted. In order to get a reasonable specific speed, the operating speed had to be as high as mechanically practicable. For a design condition of 1500 gpm and 4000 ft

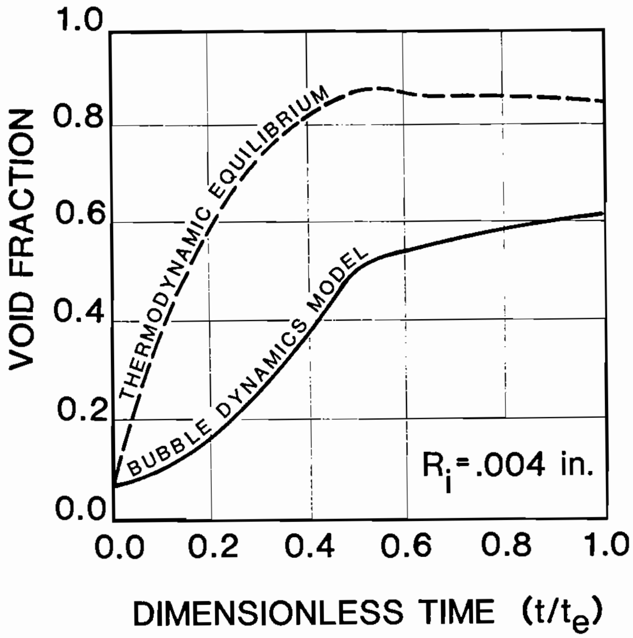

Figure 16. Comparison of Bubble Dynamics and Equilibrium Void Fractions for Single Stage Turbine (Conditions same as in Figure 15).

head, the operating speed was chosen to be 8500 cpm. This resulted in a specific speed of 650 (in gpm, cpm, ft units), which is admittedly low and results in a possible peak efficiency of no more than 70 percent. This was considered to be an acceptable compromise, because in a high gas situation, the alternatives of low speed, multistage machines do not work at all. The choice of 8500 cpm also included the consideration that the same machine should be capable of handling higher flows and heads. The highest operating speed for the turbine is 11500 cpm.

A cross section of the turbine is shown in Figure 17. The single stage eight inch diameter runner is mounted overhung on the shaft and supported by two fluid pivot bearings. A gearbox between the turbine and the driven machine (either the motor driving the charge pump or a generator) gives the desired speed ratio. An integral thrust disk operating against tapered land thrust bearings is used to take thrust in either direction. There are two mechanical seals, one high pressure bellows type seal to seal against the discharge pressure of the process stream, and the other, a so-called gas seal, to seal the bearing oil from leaking to the atmosphere. The bellows seal is capable of operating at pressures which are typical for the discharge side of the turbine. However, it cannot handle the usual inlet pressures.

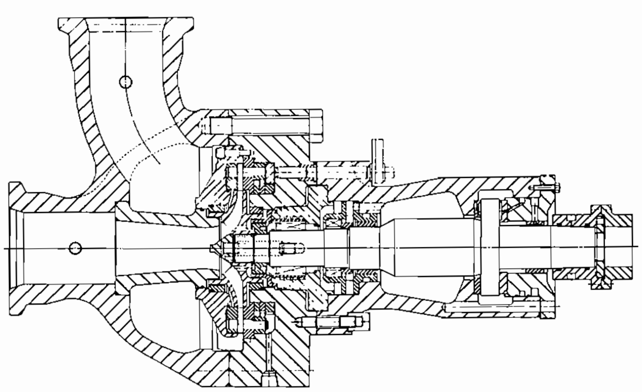

Figure 17. Single Stage Turbine Assembly.

Therefore, care must be taken to prevent opening of the inlet valve before the discharge valve during startup. For temperatures exceeding about 500°F, provision is made for cooling the seal via heat exchanger ports located in the seal housing. When solid contaminants are expected to be present in the process stream, it is advisable to provide for seal flushing using cyclone type separators. The fluid pivot bearings are oil lubricated and provide bearing properties of stiffness and damping through combined hydrostatic and hydrodynamic effects. Lubrication is delivered to the bearings of the turbine and the gearbox by two pumps—one is shaft driven and located on the gearbox, the other driven electrically and used during startup only. The gearbox used for the step down from 8500 cpm to 3600 cpm is of the epicyclic type.

The turbine runner containing 24 vanes is made of titanium. Alternate vanes are cut back at the discharge side to provide

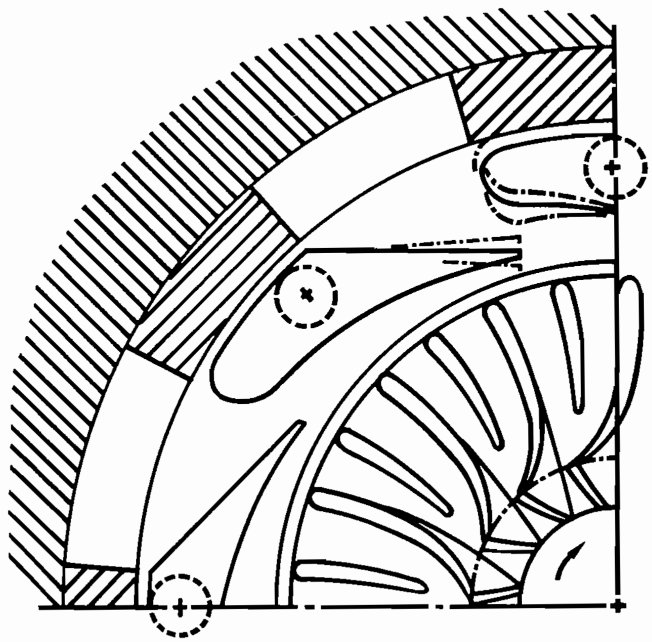

Figure 18. Schematic of Wicket Gates with Runner.

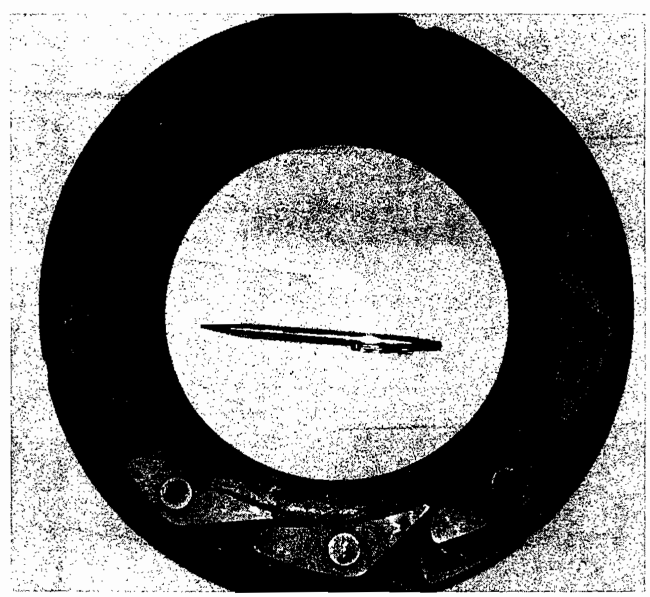

Figure 19. Photograph of Wicket Gate Arrangement—Front View.

adequate area for gas evolution. The inlet edges of the vanes are substantially radial, and the discharge angles are such as to provide nearly swirl-free exit flows into the draft tube. The calculated first critical speed, without including any stiffening effect from the fluid in the wear rings, is about 21000 cpm.

The flow capacity of the turbine can be adjusted by altering the position of the wicket gates. As shown in Figure 18, the throat opening at the wickets can be varied from ⅛ in to ⅝ in. The adjustment is made through a single rod which penetrates the turbine cover. A small gear attached to the rod rotates a large ring gear, to which is attached a set of cams, which activate followers belonging to each guide vane. Photographs of the wicket gate turning mechanism, viewed from opposite ends, are

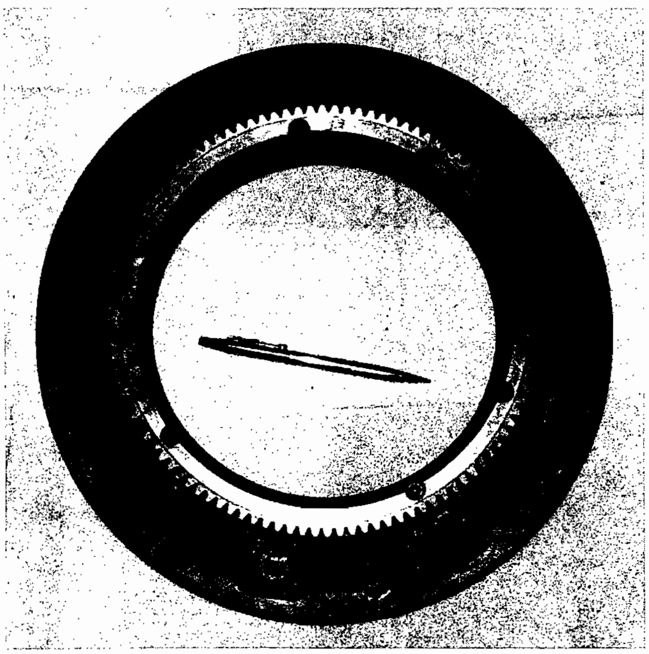

Figure 20. Photograph of Wicket Gate Arrangement—Back View.

shown in Figures 19 and 20. In the present embodiment, the pressure drop across the wicket is used to clamp the vanes tight against any motion. Even though the vanes cannot be turned while running, there is the benefit that vane tips will not flutter in spite of the high pressure drop. For lower pressure drops, where vane flutter is not a problem, the radial positioning of the "O"

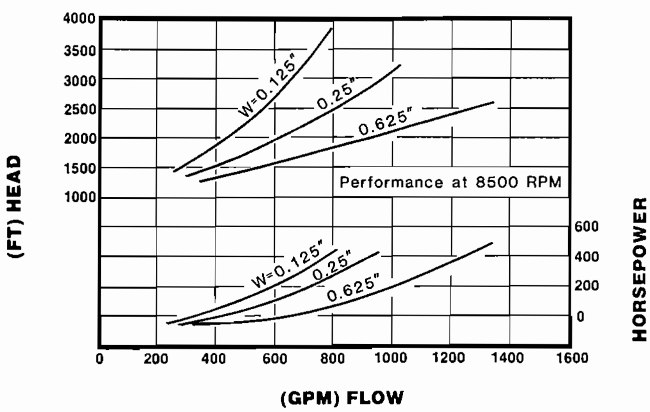

Figure 21. Test Data for Single Stage Turbine in Water for Various Wicket Gate Settings.

rings on either side of the vane can be changed to permit wicket adjustment during operation.

PERFORMANCE OF THE TURBINE

The turbine was tested at 8500 cpm and the test data are given in Figure 21, for three settings of the wicket gates. It can be seen that for a given head, a wide range of flows can be accommodated. For example, at 3000 ft head, flow capacity can be varied from about 700 gpm to 1600 gpm. The generated power correspondingly varies form 300 hp to 700 hp. The fact that the turbine maintains good efficiency over the complete range of openings can be seen from Figure 22. In this figure, the variations are plotted for two different values of the differential head.

The performance of the turbine can be altered by using an impeller-wicket gate combination having a larger axial width (usually referred to as BA) at the runner inlet. It is projected that the same casing can be used with a larger BA runner. In that case, the flow capacities are increased approximately in proportion to the increase in BA.

A comprehensive range chart can be made for this turbine as shown in Figure 23. For any given speed, there is an optimum head which gives a satisfactory head coefficient. For a given

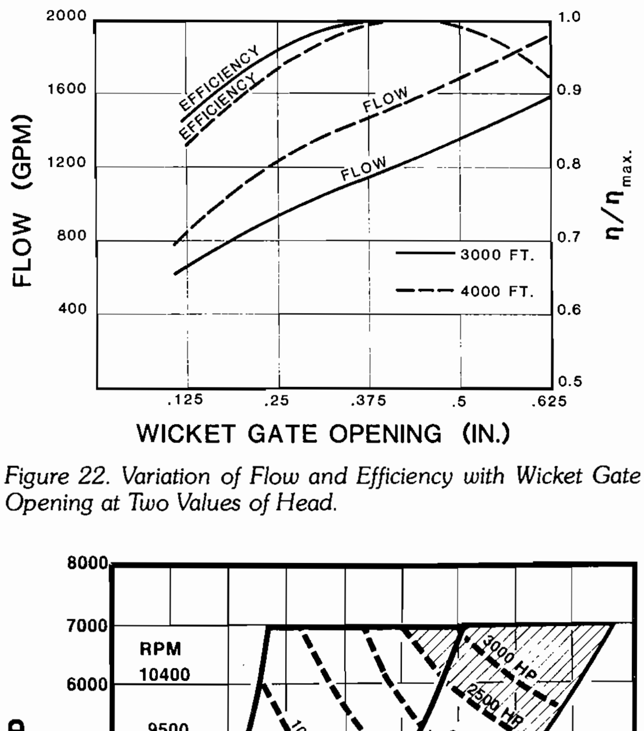

Figure 22. Variation of Flow and Efficiency with Wicket Gate Opening at Two Values of Head.

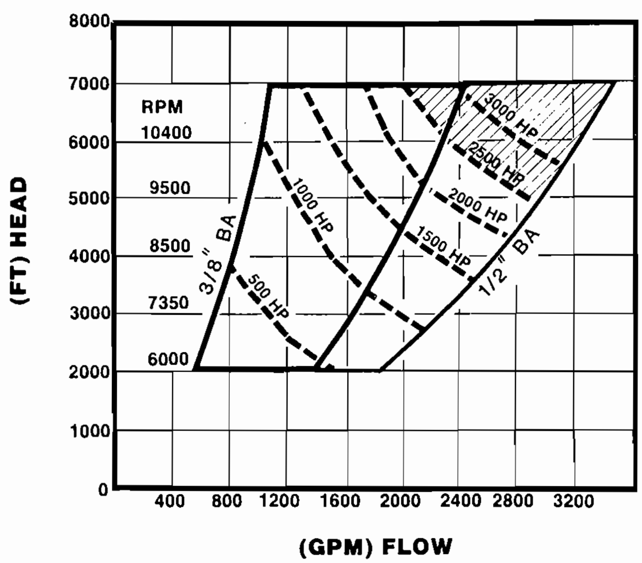

Figure 23. Single Stage Turbine Range Chart.

value of the coefficient, optimum efficiency is reached for an intermediate wicket opening. For other openings at the same speed and head, the efficiency will fall off slightly as has already been shown in Figure 20. Since the opening can be continuously changed, a continuous range of flows can be achieved for the given head. By changing the operating speed of the turbine, the optimum head can be changed. Since by using a suitable gearbox any speed can be chosen, every point in a large range can be covered by one turbine. Furthermore, the area can be moved to the right by increasing the BA from ⅜in to ½in, as shown in Figure 23. On the lower end, the range chart is cut off at 6000 cpm, because below these speeds, power recovery is too small to justify the turbine cost. The range change is restricted on the top end to about 11,500 cpm, because of mechanical seal limitations. BA values cannot be decreased below ⅜ in because the efficiency becomes too low. The top right hand portion is shown shaded, because in that area the power density is quite high, and the mechanical design must be carefully executed.

This turbine has been delivered to a refinery site for recovering energy from a 800 gpm fluid stream of amine solution containing hydrogen sulphide with a differential head of 3000 ft. At present, it is awaiting field start-up.

CONCLUSIONS

• In typical applications, an optimum choice of the turbine can provide payback periods of less than two years to the user.

• Empirical data on pumps running in reverse as turbines indicate that prediction of turbine performance from known pump performance can lead to significant errors. Actual testing of the machine is advisable to provide safe guarantees.

• Evolution of gas can produce increased power output. However, in fixed geometry machines, if allowance is not made for the increased flow rates, power output may actually decline.

• Gas evolution can cause reliability problems, due to the loss of support in close clearance running fits.

• A theoretical and experimental study has revealed that in a multi-component two-phase flow, evolution of gas is not instantaneous. Therefore, if the residence time is reduced, the gas evolution problems can be minimized.

• A new high speed turbine has been built with this idea. Its design features and performance are reported. It has operated satisfactorily in water, and is awaiting field testing under two-phase flow conditions.

REFERENCES

1. Stepanoff, A.J., *Centrifugal and Axial Flow Pumps*, New York: John Wiley and Sons, Incorporated (1957).
2. Thorne, E.W., "Centrifugal Pumps as Power Recovery Turbines," 6th Technical Conference of the British Pump Manufacturers' Association, Canterbury, England (March 1979).
3. Kittredge, C.P., "Centrifugal Pumps Used as Hydraulic Turbines," Transactions of ASME, Journal of Engineering for Power (January 1961).
4. Buse, F., "Using Centrifugal Pumps as Hydraulic Turbines," Chemical Engineering (January 1961).
5. McClaskey, B.M. and Lundquist, J.A., "Hydraulic Power Recovery Turbine," ASME Paper No. 76-Pet-65 (1976).
6. Taylor, I., "Some Installed Power Recovery Turbines Found Unusuable Due to Misconceptions of Process and Performance," *in* "Performance Characteristics of Hydraulic Turbines and Pumps," ASME Symposium, Boston, Massachussets (November 1983).
7. Gopalakrishnan, S., Fehlau, R., and Lorett, J., "Critical Speed in Centrifugal Pumps," ASME Paper No. 82-GT-277 (1982).
8. Payvar, P., "Experimental and Theoretical Study of Mass Transfer-Controlled Bubble Growth During Rapid Decompression of a Liquid," accepted for publication in International Journal of Heat and Mass Transfer.

ACKNOWLEDGEMENTS

The author is grateful to the management of Byron Jackson Pump Division, Borg Warner Industrial Products, Incorporated, for permission to publish this paper. The author would also like to recognize that the design of the single stage turbine was accomplished by J. A. Lorett.