EFFICIENCY IN MECHANICAL DRIVE STEAM TURBINES

by

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Mr. Brown was involved at C. A. Parsons & Co. Ltd. in the aerothermodynamic design of supercritical pressure turbogenerators and in commercial uses of atomic energy.

After moving to Canada in 1960, he worked as a manufacturing engineer at Canadian Westinghouse Company in setting up facilities to manufacture small steam and gas turbines. Ford Motor Company moved Mr. Brown and his family to the U.S.A. midway through an eight-year stint in various engineering assignments. He next managed advanced power plant development projects at Eaton Corporation Research Center.

After a term as manager of engineering for a supplier to the recreational vehicle industry, he assumed his present position as manager of Advanced Engineering and Development at the Steam Turbine Division of Turbodyne Corporation. One of his key assignments is conception and development of products for the process industries. Mr. Brown is an active member of ASME and SAE and has been active as a professional engineer in Ontario.

ABSTRACT

With cost of energy in the \$2.00 per 10^6 BTU range throughout the Western Industrialized World, the current value of one horsepower over a five-year period is in excess of \$600. This high energy valuation is of relatively recent origin.

Historically, initial cost has been the major purchase determinant in mechanical drive steam turbines for the process industries. In lifetime cost, the cost of energy (operating cost) was not a major factor and thus was largely neglected in evaluating the competitive products offered. In recent times the cost of energy has become the major determinant of lifetime cost and progressive users are evaluating equipment offerings using present value techniques.

INTRODUCTION

The concepts and designs presented in the paper have evolved over the years as a continuing program of efficiency enhancement for mechanical drive steam turbines for the process industries. Some of the information is already several years old. It is still current and valid.

The process industries use various types of steam turbines. For the purpose of this paper, three typical types are considered:

1. Single stage pump drive turbines
2. A large compressor driver such as might be used in an Ethylene plant
3. A high speed compressor driver as might be used in a Methanol or Ammonia plant.

Each of these is analyzed against a framework of commonly used line pressures in process plants. By the nature of the paper, the applications are generalized but the conclusions for specific jobs will show the same trends and be equally valid.

Single stage turbines are used for low horsepower applications in refineries using the plant steam mains. Common operating conditions are 600 psig 750°F inlet and 50 psig exhaust. Lower exhaust pressures are also common. Alternate inlet conditions might be 250 psig 450°F or 150 psig dry and saturated steam with the same exhaust pressure. From these conditions an enthalpy available range of 100 to 300 BTU/lb is defined.

Steam conditions for large turbines vary more since they are not tied to mains steam conditions. In my examples, I have chosen recent turbines as typical. For the Ethylene plant driver inlet conditions are 1439 psia 950°F with 280 psia extraction and 4 inches of mercury exhaust. Rotational speed is 4620 rpm. For the high speed compressor driver inlet conditions are 1450 psig 900°F with controlled extraction at 625 psig and 4 inches of mercury exhaust. Rotational speed is varied but the minimum speed considered is 10,500 rpm.

Economics

In order to have a frame of reference against which to compare the value of increased efficiency, I have assumed the following:

Cost of fuel	\$2 per million BTU (fossil fuels)
Cost of capital	10 percent
Economic life	5 years
Average thermal efficiency of use	25 percent
Utilization rate	90 percent i.e. downtime 10 percent

From this, the present value of one horsepower for five years is

$$\frac{2}{10^6} \times \frac{2545}{.25} \times 365 \times 24 \times .90 \times 3.9347 = \$624/\text{HP}$$

Obviously, other bases for comparison such as cost of steam per pound can be used but for turbines the \$/HP factor is most useful because it maintains the design flow and steam conditions, a typical situation, and values the increased power.

Thus on a single stage pump drive turbine in a refinery, say 300 HP a 20 percent efficiency improvement, or 60 HP, is valued at \$37,440 which is the same order of cost as the basic turbine.

On a large compressor drive of 30,000 HP an improvement of 5 percent or 1500 HP would be valued at $936,000, again the same order of cost as the basic turbine.

An assumption of longer life, say 10 years, would result in a value per horsepower of about $1,000; an assumption of higher cost of capital say 15 percent reduces the value to $560 per horsepower.

TECHNICAL PARAMETERS

I do not intend this to be an abstract technical paper. It is rather a pragmatic paper on how users can benefit from state-of-the-art application of good engineering principles. Much of the information has been well proven in other fields but is only now becoming recognized in the mechanical drive steam turbine field.

Velocity Ratio

First of all, some fundamentals. By far, the most important determinant of turbine performance is velocity ratio. Operation at optimum velocity ratio ensures maximum advantage can be taken of all important secondary parameters. The blade speed parameter in velocity ratio is constrained by considerations of mechanical blade strength to about 1500 ft/sec at the blade tip. Few mechanical drive steam turbines are at this limit. The fluid spouting velocity term in velocity ratio has no theoretical limit but in practice is constrained to below sonic velocity except in control stages. This partly is due to the requirements for operational flexibility required by users and partly by conservatism in the industry. From the point of view of optimum use of standard components, in this case turbine stages, the subsonic design regime has the wide flexibility to satisfy most operating characteristics. I am not proposing other than a very modest change to these design philosophies.

For single row impulse stages (Rateau) velocity ratio is at an optimum for efficiency at about .48: for two row velocity compounded impulse blading (Curtis) velocity ratio is optimum at .235. For reaction blading velocity ratios in excess of .50 result in higher efficiencies but the optimum depends substantially on a secondary parameter, degree of reaction. For 50 percent reaction optimum velocity ratio is about .65.

In each case the various manufacturers have minor variations in optimum velocity ratios as a result of the influence of secondary parameters which they consider, many of which are unique to their configurations.

There is little incentive in a standardized system in using high velocity ratios since this requires a larger turbine, either more stages or larger diameters. Low velocity ratios require fewer stages but means more work per stage and thus a lower cost but relatively less efficient turbine. This last regime is where mechanical drive steam turbines with emphasis on first cost have been designed.

Figure 1 in the Appendix shows typical curves of velocity ratio versus efficiency.

Degree of Reaction

Reaction blading is theoretically more efficient than impulse blading. 50 percent reaction blading has manufacturing advantages and is commonly used. For mechanical drive steam turbines 50 percent reaction blading is rarely the most efficient in practice due to the dysfunctional secondary affects of leakage and due to the unique geometry of mechanical drive turbines.

The effects of leakage are shown in Figure 2 together with typical efficiency curves. It should be noted that in general with short blades (high hub/tip ratios) the realization of 1 per-

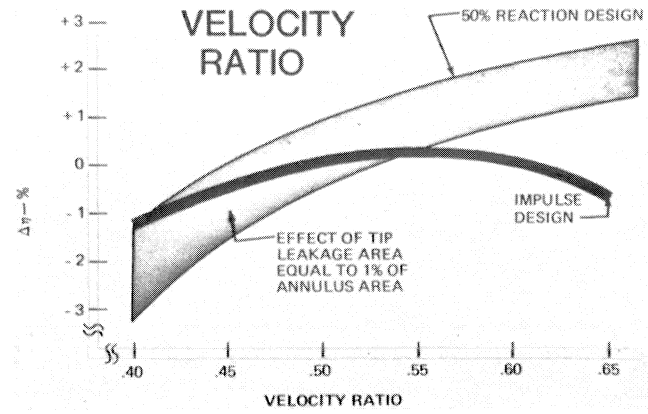


Figure 1. Velocity Ratio vs Efficiency.

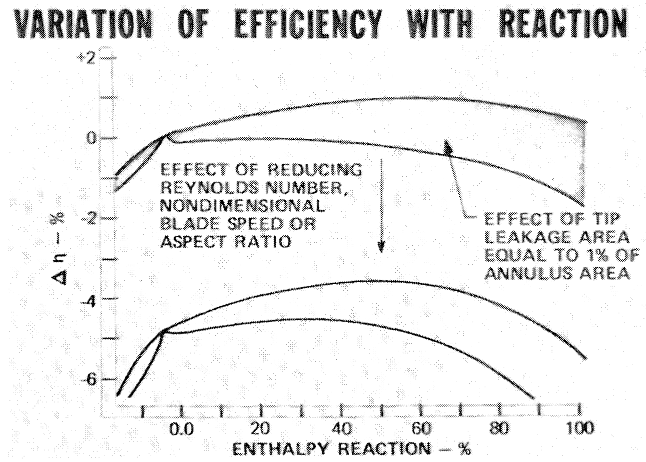


Figure 2. Variation of Efficiency with Reaction.

cent leakage flow is mechanically difficult even with the use of tip seals. Tip seal deterioration accounts for the observed deterioration of reaction blading with time; the seals abrade and erode to the point that they will not control leakage. A loss in efficiency of 1 percent per year is usual in reaction turbines. In order to keep leakage down, reaction turbines are designed with many stages reducing the pressure drop per stage and limiting the work per stage. All turbines in the mechanical drive field use impulse control stages.

Upstream from the bleed section on mechanical drive steam turbines blade heights are typically from .625 to 2 inches and impulse blading is more efficient. Where large blades are used in exhaust sections (low hub tip ratios) then leakage control becomes simpler and a degree of reaction is attractive although generally the degree of reaction for optimum performance would be other than 50 percent.

An additional factor which must be taken into account in the application of reaction blading is the provision for end thrust. If a balancing (dummy) piston is used then the flow associated with pressure balancing must be debited the turbine. In discussions of blade efficiency this factor is commonly neglected but can be an appreciable quantity when the ultimate efficiency is required. Using opposed double flow staging obviates this problem but may reduce blade height to an unacceptable level or increase shaft length.

Reynolds Number

Reynolds Number for typical head end stages in steam turbines is greater than 10^6. It is not manipulable by the turbine designer since it is dictated by the kinematic viscosity term.

In gas turbine practice Reynolds Number is important and can have a significant although secondary effect on performance. Only in low pressure steam turbine staging does Reynolds Number have similar values and influence. Even there, values in excess of 2 x 10^5 are common. Experience suggests that increasing Reynolds Number results in improving efficiency and less sensitivity to secondary flow losses.

At low Reynolds Numbers, turbine performance becomes sensitive to other turbine parameters.

Aspect Ratio

Blade aspect ratio and its close relative hub-tip ratio are manipulable by the turbine designer and can have an appreciable affect on the performance of the class of machinery used for mechanical drives. In the Ainley-Mathieson loss system hub/tip ratio is the prime determinant. The Dunham and Came modification gives experimentally reproducible predictions using Aspect Ratio.

Head end blading of an extraction turbine will have low aspect ratio blades with a large hub-tip ratio. Aspect ratios of one or less are common as the designer shortens the blade in order to control flow area and get close to full admission. Minimum blade length of .6 inch with chords of order 1.0 inch are typical. Exhaust end blading may have aspect ratios of 4.0 with hub-tip ratios of .60 to .70 not uncommon. Typical exhaust blade root chords would be in the 2 to 5-inch range varying with operating speed requirements. The critical stress areas in rotating blades are in the exhaust stages. Shortening the chord is an obvious way to improve aspect ratio but is limited by stress considerations especially steam induced bending stresses.

It should be strongly noted that extraction turbines for mechanical drives, do not have critical exhaust size problems. Often 80 percent of the steam flow is extracted at an intermediate pressure. Thus exhausts are sized for only 20 percent of inlet flow. In utility generator drives, 75 percent* flow to exhaust is more common and serious limitations arise in exhaust size.

Leakage

Leakage is an important consideration and deserves careful consideration by designers and users. Internal leakage (interstage) reduces useful work in a stage and may be the cause of aerodynamic losses through mixing and disturbance of boundary layers.

Blade Tip Leakage (Figure 3). In impulse turbines, blade tip leakage is small because the pressure gradient across the blade is small. High reaction blading has a significant pressure drop across the blade and elaborate seals are required to prevent major losses. These seals are the major modifier of theoretical higher efficiency of reaction blading. In short, high pressure blading the leakage loss exceeds the theoretical gain by a significant margin; in long blades, leakage may be controllable such that most of the theoretical gain can be realized. Blade tip seals tend to degrade with time due to wear and erosion and due to shaft excursions. The seal degradation results in performance deterioration with time not experienced with impulse blading. A degradation of 1 percent efficiency per year is commonly expected even with a good reaction turbine design.

*The balance is extracted for feed water heating.

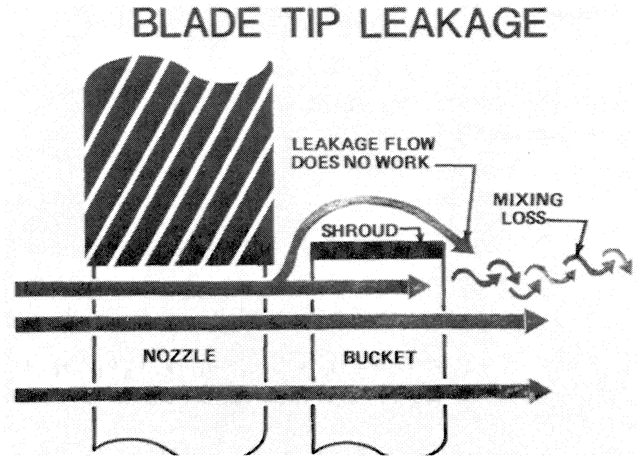


Figure 3. Blade Top Leakage.

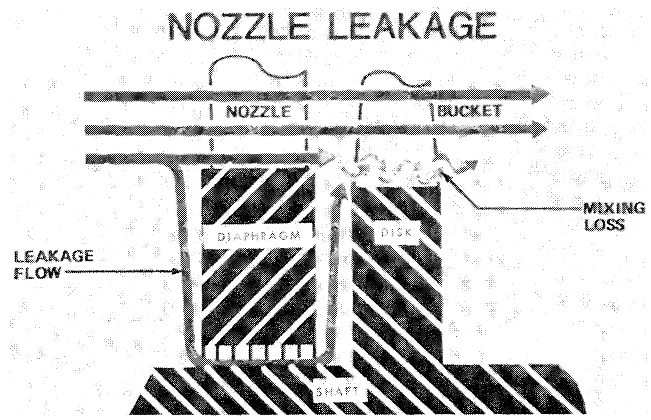


Figure 4. Nozzle Leakage.

Nozzle (stator) Leakage (Figure 4). Leakage through a diaphragm seal in impulse turbines is closely controlled, typically 1 percent or less of the design flow. Some degradation occurs with time and useful performance gains can be made by careful maintenance of these seals. Replacement of the seals will restore original performance.

In reaction blading (stators) the loss due to leakage depends upon the construction. The common drum rotor (rather than disc) and 50 percent reaction blading results in losses of order 3 to 6 percent in blade lengths of interest in mechanical drives. The controlling seals are often part of the blading and are thus not easily replaced.

Disc Windage and Pumping (Figure 5). Associated with leakage flows are windage and friction losses. With all else equal (steam pressure and temperature, etc.) smaller diameters and higher rotational speeds result in reduced losses (for the same tip speed).

Typical disc type rotors used with impulse blading have significant losses at high Reynolds Numbers i.e. the head end of the turbine. Drum rotors have less losses from these causes but this tends to be more than offset by the increased leakage loss. Indeed some care must be exercised by the turbine designer in head end staging that secondary losses do not wipe out higher aerodynamic performance.

DESIGN PHILOSOPHY

Mechanical drive steam turbines have been developed as a special breed in the U.S.A. Unique combinations of

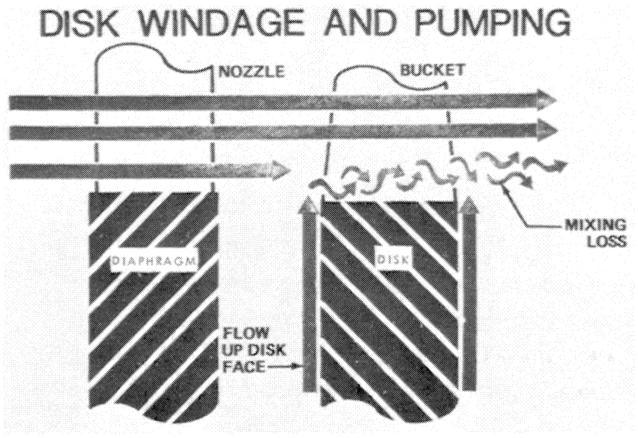


Figure 5. Disk Windage and Pumping.

standardized stages are matched to satisfy the operating requirements of a specific job. Because mechanical drive turbines are used essentially as topping turbines with much of the power generated at the high pressure end, then impulse blading becomes the first choice. When considerations of low quality of feed water, short blade height and, commonly, partial admission are also considered then the selection of impulse blading becomes the prime choice.

European designed turbines owe much more in their geneology to the power generation field, where closely matched reaction stages operating at constant speed with large exhaust flows are more typical. Blading designed for constant speed use, whether impulse or reaction, can be finely tuned through lacing wires and thus tends to be very slender (high aspect ratio) when compared to mechanical drive blading intended to be run over a wide range of operating speeds. The aspect ratio of exhaust blading on a steam turbine generator driver might be as high as 10:1 where a gas turbine equivalent might be 6:1 and a mechanical drive steam turbine blade 4:1. With wide ranges of operating speed it is a difficult if not an impossible mechanical design task to avoid all low order blade resonances in the operating range. Thus the blade must be sturdy enough to run under adverse operating conditions. For this reason double flow (and triple flow) single casing turbines have been developed for mechanical drives keeping the blading short and sturdy.

Single stage turbines (SST) are, of course, virtually all impulse design for reasons of control. In many end uses the energy available in the plant mains is greater than can be efficiently utilized in Rateau (single row) stages and Curtis (two-row velocity compounded) stages are used. Because of the inherently poor matching of the rows in a Curtis stage, maximum achievable efficiency tends to be substantially lower than with a Rateau stage. Each U.S. manufacturer of SST offers an off-the-shelf standard line of turbines. Availability and standardization

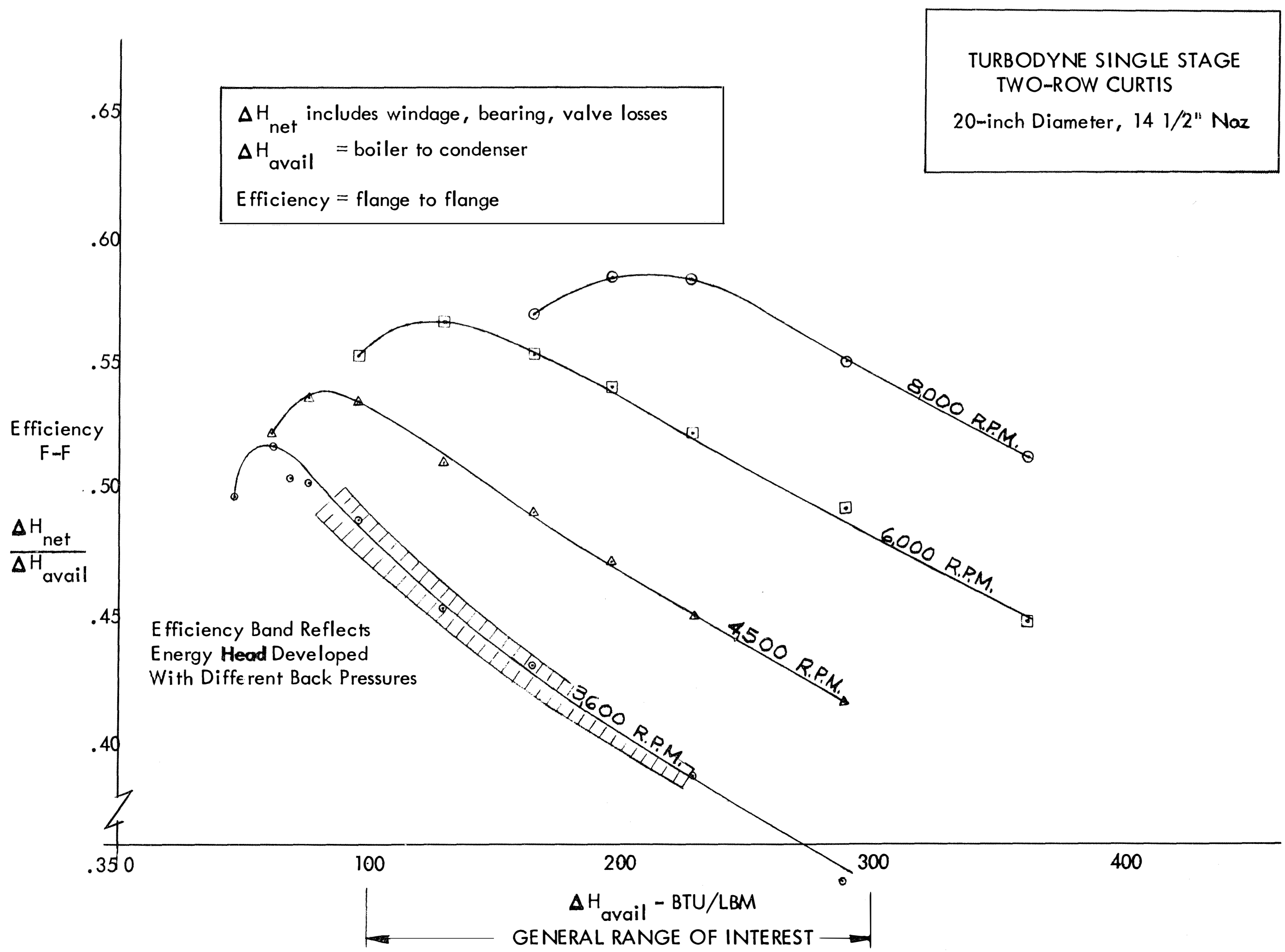


Figure 6. Efficiency vs Energy Head.

has historically been of more importance than efficient use of energy. If more efficient low speed (but higher horsepower) pump drives are required then standardized multistage turbines (MST) can be used. These designs reduce the energy drop per stage and use Rateau staging to achieve a substantial efficiency gain. Due to their added complexity and cost MST are generally not used for horsepowers less than 2000.

Analysis of Turbine Performance

I propose first to cover performance of SST, applying the various parameters previously noted. For my own convenience I have chosen to use turbine configurations standardized by Turbodyne STD. Key dimensions are pitch diameters of 14 inches, 20 inches and 28 inches.

SST PERFORMANCE — PRESENT AND POTENTIAL

Shown in Figure 6 is the distribution of net efficiency as a function of available energy head for a family of shaft speeds for a 20-inch diameter wheel. Available energy head is itself a function only of the steam conditions at the inlet and exhaust. Windage loss is an increasing function of speed and exhaust pressure. From this it can be postulated that each speed line of efficiency versus available head lies within a band of about ± 1 percent efficiency. The bottom of the band (least efficient) reflects an energy head developed with a relatively high back pressure, where the losses are maximum for that energy head. With this consideration in mind, the distribution of efficiency as a function of speed, diameter, and energy head can be extended to any set of steam conditions. It is seen that the family of speed curves reaches a maximum efficiency, with the low speeds reaching maximum at the low energy heads. Considering only the maximum points, it is seen that the maximum efficiency increases as speed and energy head increase. Figure 6 is particularly useful in selecting a shaft speed for the best efficiency at a specified set of steam conditions. If both shaft speed and steam conditions are selected the opportunity to optimize efficiency is lost. For the unfortunate selection of high energy head and low shaft speed, the efficiency is quite low.

Even maximum efficiencies for this class of turbines operating under typical conditions is at best modest. For construction of the curves optimum velocity ratio of .235 has been used. In practice, efficiency could be even lower due to use of other than optimal velocity ratios.

For constant rotational speed, larger diameter wheels with their higher blade speed and thus better velocity ratio for a given available enthalpy offer improving efficiency. This is shown in Figure 7 for 3600 rpm and Figure 8 for 6000 rpm.

Many refinery pumps are designed for low speed (compatible with motor drive at 3600 rpm) and the turbine performance penalty paid is quite clear. Not as clear is that low speed pumps may also sacrifice efficiency. Figure 9 is a reproduction of pump performance curves from "Centrifugal Pumps and Blowers" by Church. The use of inducer type pumps of optimum geometry at 6000 rpm is expected to result in additional efficiency improvements but is beyond the scope of this paper.

Turbines also offer potential for control of pump head and flow by speed control. With frequent operation at other than design point, substantial additional improvements are possible.

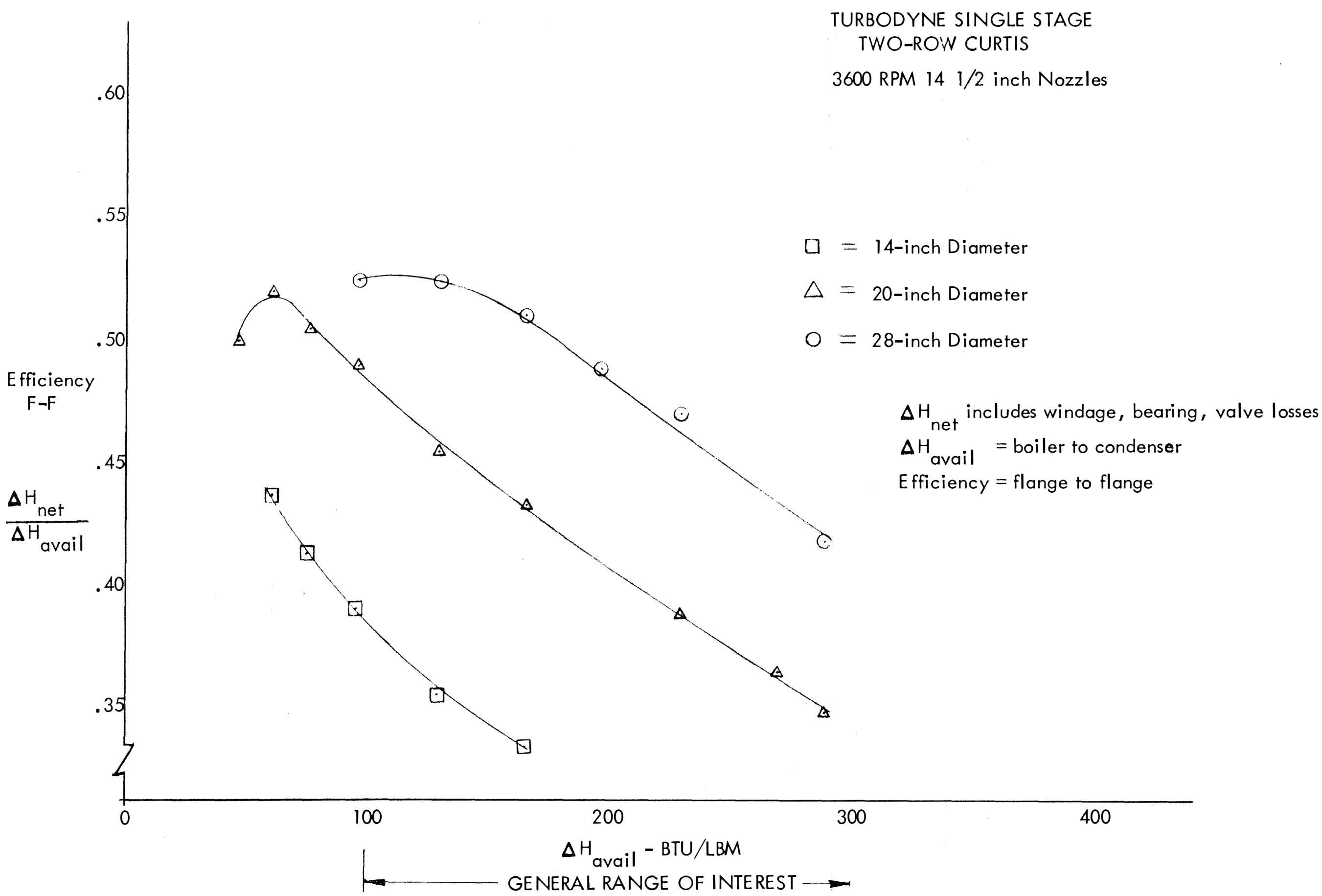


Figure 7. Efficiency vs Energy Head.

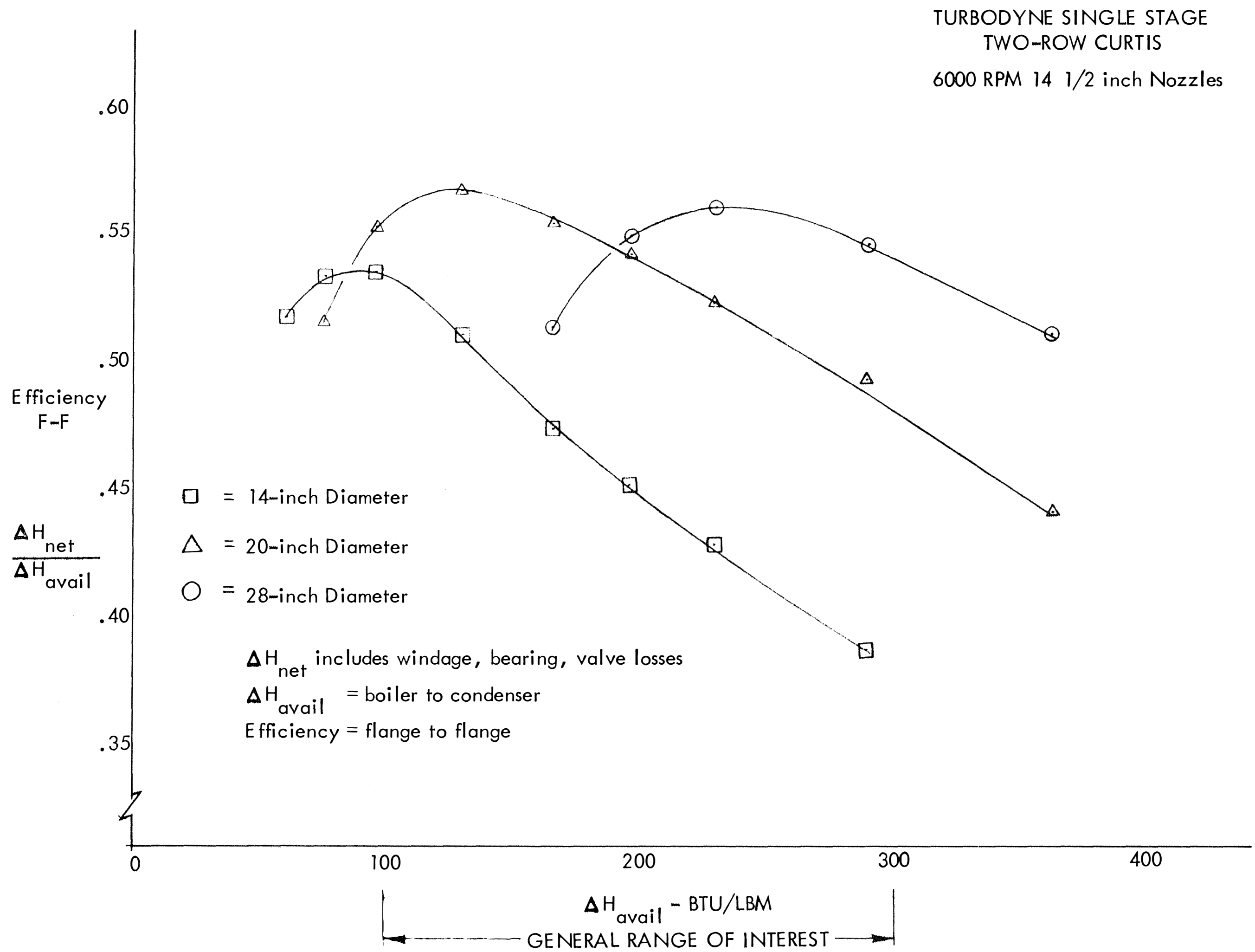
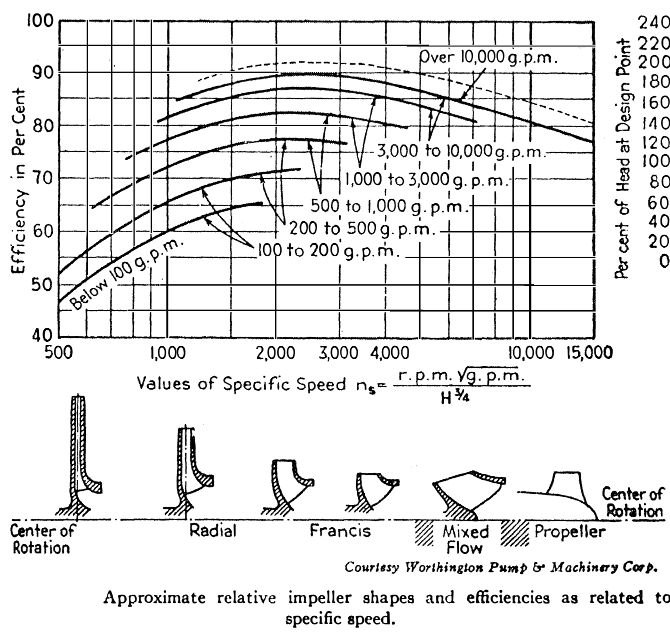
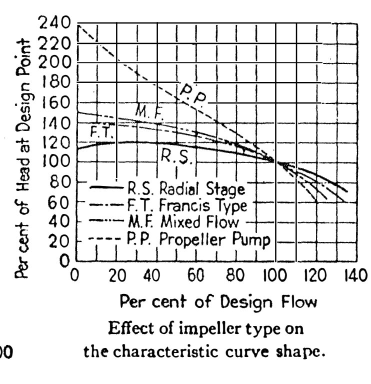


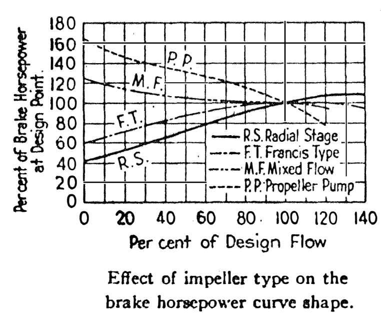
Figure 8. Efficiency vs Energy Head.



Approximate relative impeller shapes and efficiencies as related to specific speed.

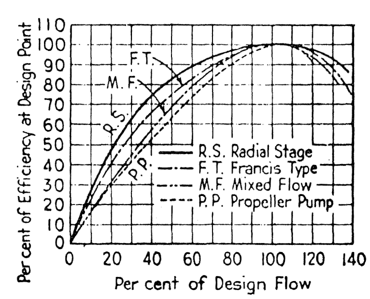


Effect of impeller type on the characteristic curve shape.



Effect of impeller type on the brake horsepower curve shape.

From "Centrifugal Pumps and Blowers" by Church



Effect of the impeller type on the efficiency curve shape.

Figure 9. Pump Geometry and Performance Data.

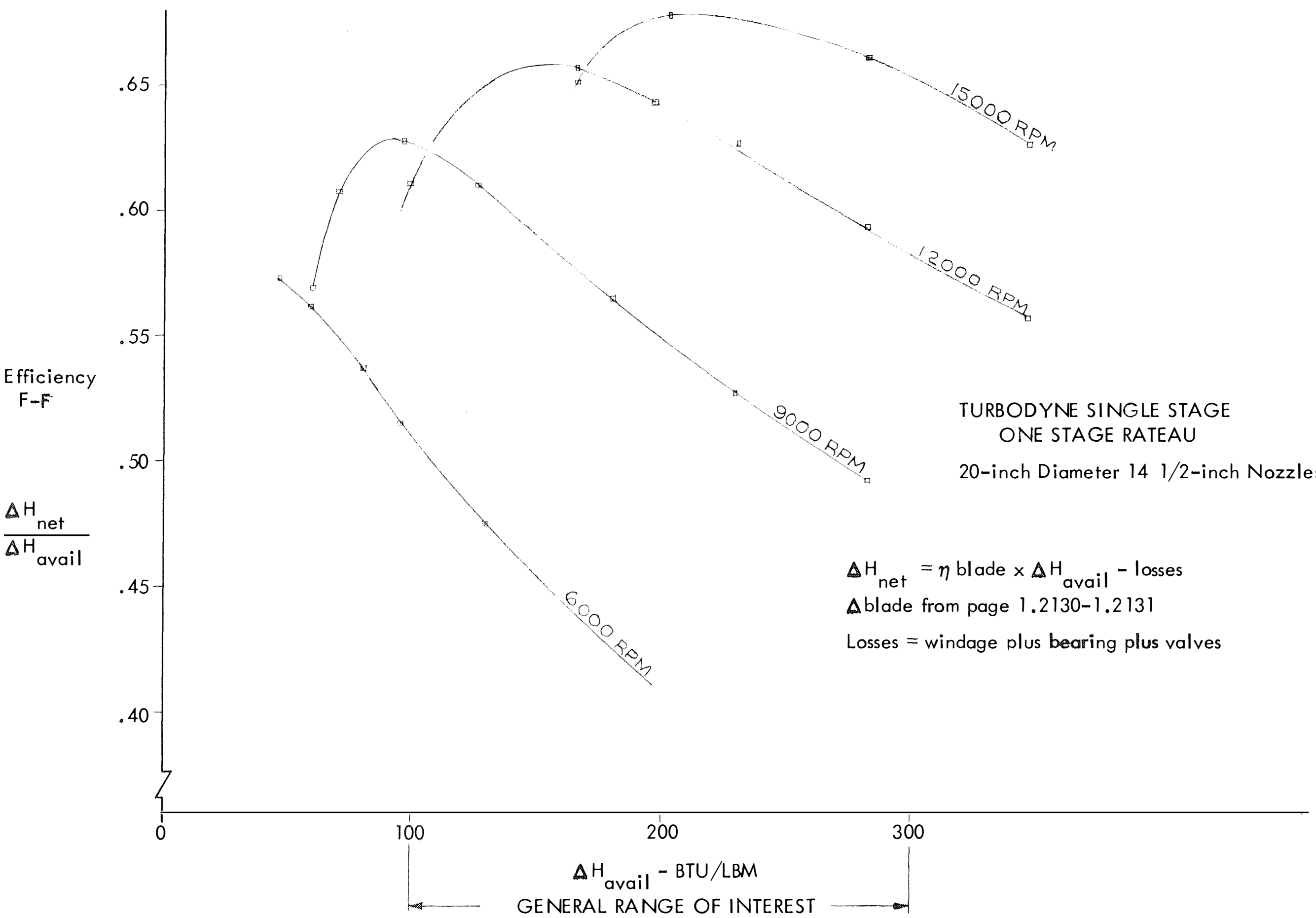


Figure 10. Efficiency vs Energy Head.

Thus rather than a constant speed turbine driving a pump with a pressure regulator, a turbine can be built incorporating a governor controlling speed and pump output pressure. This gain is a system gain and independent from turbine performance.

For speeds over 6000 rpm the addition of gearing would appear to offer significant performance advantage with small risk. Single mesh gears of good design would degrade efficiency by 2-3 percent which is very small in comparison to the thermodynamic gain. Again, the cost of gearing appears modest compared with multistaging costs required to achieve equal overall efficiency.

Figures 10 and 11 show the benefits expected from the use of Rateau staging rather than Curtis. Figure 10 is a single rotating row configuration and Figure 11 a two-row configuration. Much higher rotational speeds are required to optimize velocity ratio for Rateau staging (.49). Cost of Rateau staging is considerably more than for Curtis staging and blade root construction much more critical but still within the bounds of historical practice in the mechanical drive steam turbine industry.

A comparison of maximum efficiency envelopes is shown in Figure 12 for the three configurations investigated. The maximum rotor speed considered in this comparison is 12,000 rpm (1046 ft/sec blade speed for a 20-inch diameter wheel). At an energy level of 100 BTU/LBM, a typical 3600 rpm direct drive turbine may be optimized to give an efficiency improvement of 17 percent for a geared two-wheel Curtis, 37 percent for a geared one stage Rateau, and 42 percent for a geared two-stage Rateau. At an energy level of 200 BTU/LBM, the efficiency improvements are 46 percent, 61 percent and 85 percent, respectively. It is clear from these numbers that significant improvements in single stage turbine performance are possible by optimizing the wheel speed through the judicious selection of a gear. Clearly the freeing of the turbine from artificial rotational speed constraints results in very substantial gains in efficiency. This advantage must be evaluated rationally against the increased cost of the turbine and gear (and possibly pump) to determine net advantage. I estimate that a geared high speed, high efficiency turbine would cost about twice as much as its low speed counterpart.

Manipulation of other technical parameters resulted in no net gain in SST performance. Indeed a reaction design considered was 5-10 percent inferior in overall efficiency.

An interesting sidelight to this discussion is a comparison of SST with the first stage of a syngas compressor driver. The syngas stage is an essentially full admission (rather than 30 percent) single Rateau wheel and can develop up to 25,000 HP at 10,500 rpm with an overall efficiency of .70 from 1450 psig 950°F steam with 350 psig back pressure. Such turbines have been in successful operation for several years of running time.

EFFICIENCY LEVELS OF MULTISTAGE TURBINES

For both of the examples of multistage turbine performance, I have chosen turbines recently installed in their respective plants (since mid 1975). Thus the base from which I am working is representative of technology of 1973 vintage. The

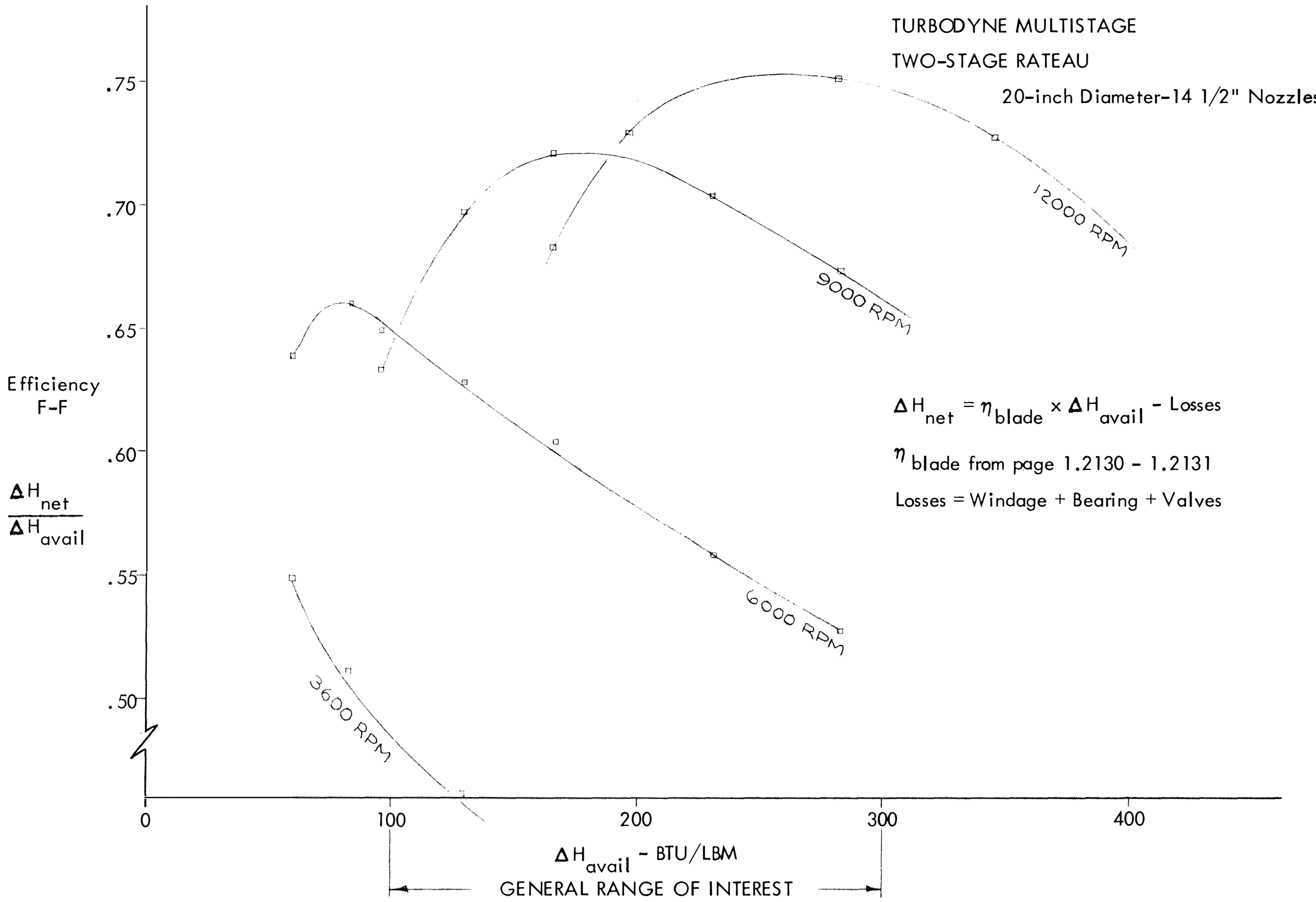


Figure 11. Efficiency vs. Energy Head.

design parameters are those in use prior to the "energy crunch" i.e. low first cost over efficiency.

Both turbines have common features and concepts appropriate to one are generally appropriate to the other although the design detail varies substantially. The low speed ethylene driver is less demanding in mechanical blade design than the syngas driver but poses major casing design problems in containing the first stage pressure (1050 psi versus 625 psi). Because of the lower blade speed less work per stage can be used at maximum efficiency.

General

Multistage turbines for process plants are often designed for bootstrap starting. Such a design requires a greatly oversized exhaust section for zero extraction i.e. full condensing operation at perhaps half power. At normal operation the extraction takes 70 to 80 percent of the stop valve flow. Sizing the exhaust for starting condition results in mismatched components under normal operation. Major efficiency penalties are taken for this convenience and I recommend that specifications be worded to ensure that the turbine design point is based on the steady state operating condition. The value of the efficiency gains is such that special equipment may be more attractive for start up — such as a clutched secondary turbine. The process engineers among you will know better than I the alternatives for start up.

For multistage turbines the reinjection of high pressure shaft seal leakage at an appropriate downstream stage results in a pickup of 50 to 100 horsepower. This may not sound inspiring until applying the valuation proposed in this paper. This very modest change is valued at $31,200 to $62,400 or over ten times the incremental investment to implement within five years! Surely it is worth specifying.

Selection of steam conditions can also reduce losses. By choosing inlet pressure and temperature at a suitable level to ensure exhaust with low wetness, then the losses (and mechanical problems) associated with such wetness can be obviated. Depending on the loss system, used, wetness results in losses of ½ to 1 percent in stage efficiency for each percent of wetness in the stage.

Compressor Drive Turbine (Ethylene Service)

Conditions of service

SVP	1439 PSIA
SVT	950°F
Superheat	360°F
RPM	4620
Extraction	280 PSIA controlled
Exhaust	1.9 PSIA
Stop Valve Flow	598,000 lb/hr.
Exhaust Flow	220,000 lb/hr.

The turbine as built generated 35000 HP in the head end and 24000 HP in the exhaust section from three and seven stages respectively.

By optimizing velocity ratio and including optimal degree of reaction in the exhaust blading the powers become 39000 and 26000 but from five and nine stages respectively. Overall efficiency weighted for horsepower increased 7%.

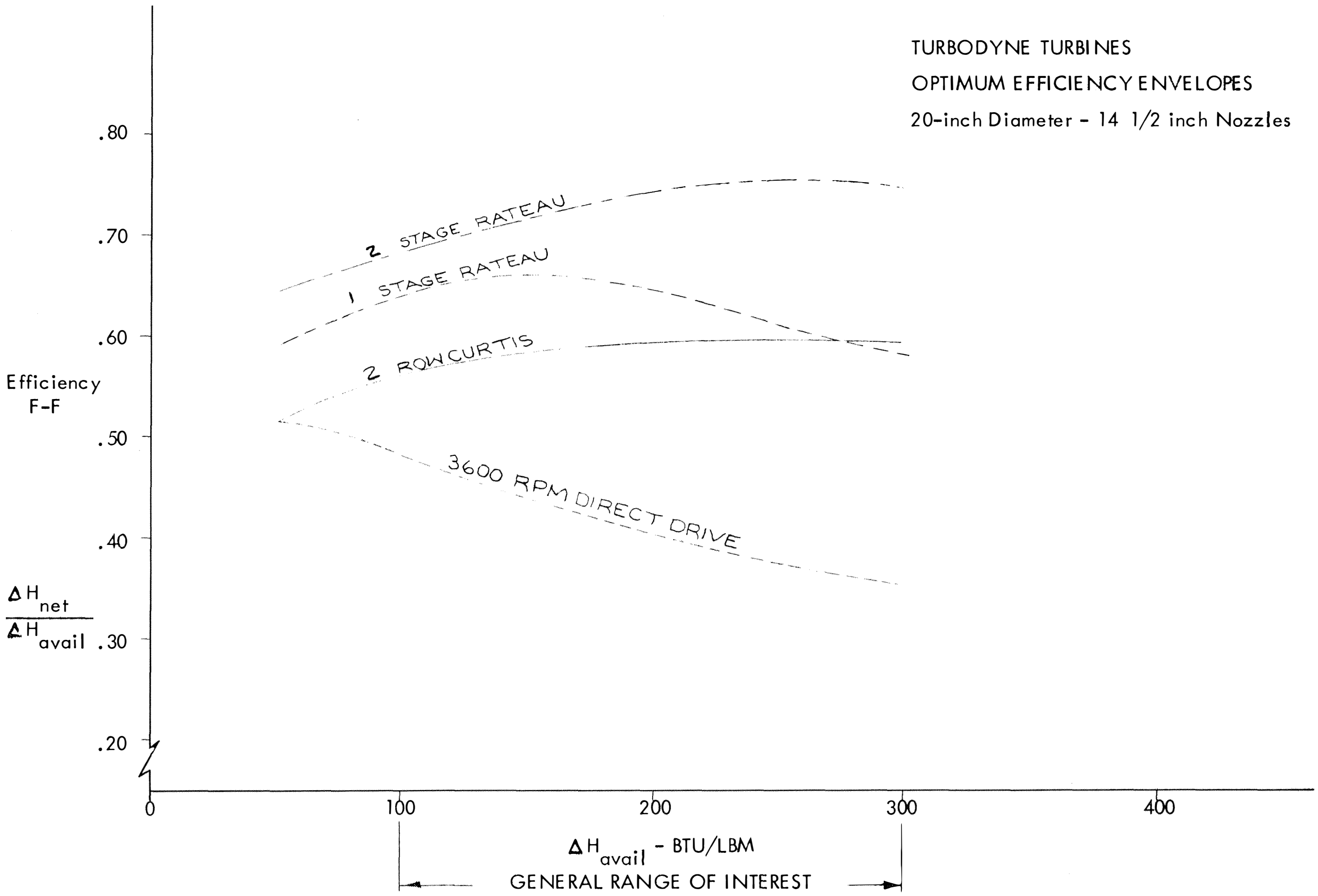


Figure 12. Efficiency vs Energy Head.

Thus according to the valuation of horsepower proposed in this paper the added value is $2.75 million. Added costs for the higher efficiency turbine in production, I estimated at about $250,000. The configurations are summarized in Table 1.

Of course, the increased number of stages can pose some problems. The longer, heavier shaft could have rotor dynamic performance problems and would require careful design analysis to ensure adequate stability. Present state-of-the-art in rotor dynamics appears able to resolve such problems as could occur.

There is no question in my mind that current standard hardware can be made smaller and more compact without performance detriment. Shaft seals and leakoffs are a prime example. Any compaction of seals, leakoffs, extractions, etc. can be used to offset the increased shaft lengths due to more stages.

Most of the aerodynamic improvements realized for this large turbine are the results of optimizing velocity ratio (4.6 percent). Optimizing the degree of reaction provides a small gain (1.0 percent). Reynolds Numbers were already high in the base turbine and would not change for the high efficiency version.

A head end diffuser is not included in the optimized version but could result in a 1.5 percent overall efficiency improvement. Reinjection of high pressure gland leakage at a downstream stage can result in a net gain of horsepower of order 50 HP.

TABLE 1

HEAD END	TURBINE AS BUILT	OPTIMIZED
Number of Stages	3 Rateau	5 Rateau
Mean Diameter	32(1) 34(2,3)	34 ALL
Horsepower	36000	39000 (+9 %)
EXHAUST END		
Number of Flows	2	2
Number of Stages	7 Rateau (with 15% reaction)	9 Rateau (with 20% reaction)
Mean Diameter	33(1) 34(2,3,4,5)	34(1-7)
	41(6,7)	41(8,9)
Horsepower	24000	26000 (+5 %)
TOTAL TURBINE		
Horsepower	59893 10 stages	64308 (+7.4%) 14 stages
Efficiency F-F	Base	+7 % improvement

Compressor Drive Turbine (Ammonia Service)

Are equivalent efficiency gains possible in a syngas drive turbine where high speed 10,000-12,000 rpm is already factored into the design?

In order to determine the potential for improvement, a study was conducted to compare efficiency of a recently designed turbine, as built, with an "optimized turbine" using extant technology and a "wholly optimized turbine" capable of being built within 2-5 years. The "optimized turbine" only includes low risk conventional technology but assembled in a unique way.

It should be noted in an ammonia type plant with several large turbines involved considerable flexibility in steam flow and steam conditions exists provided that the specification permits such flexibility. By switching steam flows from less efficient to more efficient turbines several points of CYCLE efficiency can be gained.

The conventional syngas driver was required to generate 25,000 HP at 10,500 rpm from 1450 psig 900°F steam with a major extraction at 600 psig 700°F. The extraction steam header is used to supply the air and ammonia compressor drivers with 4 inches of mercury absolute exhaust.

Figure 13 shows the configuration and Table 2 summarizes the pertinent information. I have included the secondary drivers for completeness since they were part of the overall study. However, I do not propose to analyze them in detail. The same efficiency improving features can be applied to them.

It can be seen that useful efficiency gains (about 3 percent) can be made using present technology but at the penalty of special construction in the high pressure nozzles and by incorporating special low loss seals and a low loss steam chest. The head end (one stage) develops 16,500 HP or 65 percent of the total turbine power. For reference the blade tip speed is 1110 ft/sec. The steam conditions of the particular turbine permit optimized velocity ratio (.49). Arc of admission is 95 percent despite the use of six valves to maximize efficiency at part load.

The Rateau stage is essentially impulse. Due to the large steam bending stresses the aspect ratio is less than 1.0 although

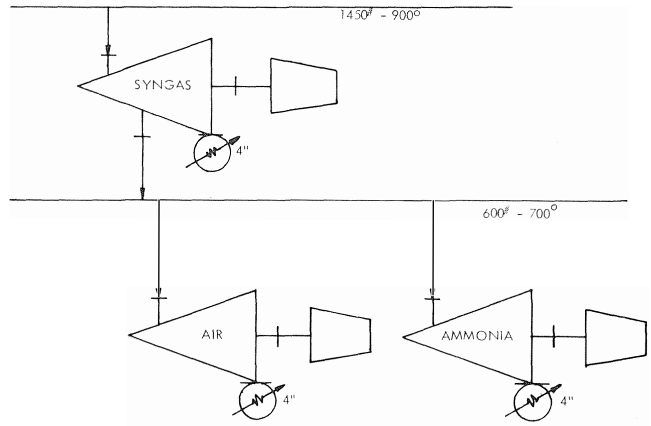


Figure 13. High Efficiency Ammonia Plant Turbines.

conventional root construction is used. Considerable additional loading can be accommodated by the use of special root construction.

Inquiries of compressor manufacturers suggest that the 10,000-12,000 rpm of present syngas drivers is a compromise and that substantial potential exists for improved train efficiency by increasing rotational speed. A number of versions were analyzed fully using complete aerodynamic and mechanical analytical tools. By varying wheel diameter and rpm various levels of performance were compared without exceeding stress levels for which running experience is available. At speeds in excess of 15,000 rpm the reduced flow area available through the rotating blades (for the same limiting tip speed) restricts power to less than that required. At 15,000 rpm the turbine can generate the specified power. Table 3 lists the optimum present and the proposed 15,000 rpm turbine. You will note that the head end gains a percent while the exhaust end loses a percent, not a bad trade since the head end carries twice the power. Most of the added losses in the exhaust end are due to

TABLE 2

HIGH EFFICIENCY AMMONIA PLANT TURBINES

TURBINE	CONFIGURATION	FLOW	RPM	HP	DOLLAR SAVINGS
SYNGAS HEAD END	Present	527,000			
	Improved Present	524,000	10,500	25,000	As Below
	Optimized Present	510,000			As Below
SYNGAS BACK END	Present	72,137			
	Improved Present	70,750	10,500	Included	144,000
	Optimized Present	64,600			785,000
AIR	Present	100,915			
	Improved Present	100,249	7,600	13,331	52,000
	Optimized Present	94,040			541,000
AMMONIA	Present	70,087			
	Improved Present	69,150	7,200	9,370	73,000
	Optimized Present	66,310			296,000

TABLE 3

HYPOTHETICAL AMMONIA PLANT TURBINES

PERFORMANCE COMPARISON

						LOSS DISTRIBUTION - BTU/LBM								
Turbine	Unit	RPM	HP	η_{F-F}	Flow	Control Valve	Exhaust	Stage	Leaving	Moisture	Windage	Leakage	Mechanical	Other
Syngas Head End	12-11-75 3-25-75	10500 15000	16500	Base +1.6%	510,000 509,820	Base Same	Base Same	Base -1.2	Base +.1	Base Same	Base -.2	Base Same	Base Same	Base Same
Syngas Back End	12-11-75 3-25-75	10500 15000	8900	Base -1.1%	64,460 63,520	Base Same	Base Same	Base +5.6	Base +6.3	Base +.3	Base -9.3	Base +.8	Base Same	Base Same
Air	12-11-75	7600	13000	-	94,000	-	-	-	-	-	-	-	-	-
Ammonia	12-11-75	7200	9400		66,300	-	-	-	-	-	-	-	-	-

greatly increased leaving loss somewhat offset by reduced windage losses. The last (exhaust) rotating blade poses a development problem. Its tip speed is 1500 ft/sec and a conventional root section is not possible. However, detailed analytical calculations show that the Turbodyne axial entry root configuration (fir tree root) is a viable design configuration. A larger blade would gain some 4 percent on efficiency for the exhaust section or about 1.3 percent weighted to the whole turbine. Such a blade could be made using titanium construction together with an axial entry root. While titanium blading has not been widely accepted in the process steam turbine industries the technology to make such a development is well known and should pose only normal design compromise problems.

Efficiency can be gained in the important head end stage by use of a diffuser section. A short annular diffuser, possibly with boundary layer suction for control, would be possible within the envelope of present casing designs.* Gains from a diffuser would approximate 2.0 percent on the head end or 1.3 percent weighted for the whole turbine.

Thus, in summary, we can see low risk developments which on a relatively efficient modern turbine can result in a significant overall improvement.

3.0% by design (mostly velocity ratio) optimization and special construction

1.2% by increased speed from 10500 to 15000 rpm

1.3% by head end diffusion

1.3% by advanced exhaust blading or

6.8% including the entire package of improvements.

Thus we can expect perhaps .86 efficiency from such an advanced concept turbine rather than the present .80.

Under the evaluation system proposed in this paper such an improvement would be valued at $950,000. Cost of implementation including tooling amortization is expected to somewhat less than $100,000 per turbine.

*Based on "Nelson, Yang, Hudson — Design and Performance of Axially Symmetric Contoured Wall Diffusers Employing Suction Boundary Layer Control" ASME 74GT152.

CONCLUSION

From the analysis it can be seen that very significant efficiency — performance gains can be gained across the range of mechanical drive steam turbines. Most of the gain can be achieved using old technology and thus poses low risk. The primary mechanism in achieving the improvement is optimization of velocity ratio with some secondary gains in optimizing degree of reaction. Other steps involve control of internal and shaft leakage and reinjection of the high pressure gland leakage.

Efficiency can be maximized by using impulse blading where blade height is less than three inches — as it is in most mechanical drive turbines — and Rateau staging should be preferred over Curtis provided that blade speed and work per stage can be optimized. For Rateau staging velocity ratio is optimum for efficiency at .48-.49: For Curtis at .235.

Important improvements in efficiency are possible through careful specification of design point as normal operating point. Starting or short time operating points may require special equipment provision.

For SST, optimization of velocity ratio requires modifying the blade speed parameter (numerator) since the available steam energy is dictated by steam mains pressures and temperatures. If this is done then gains of 20 to 50 percent are possible. Individual turbine contribution may not be large but in a large refinery aggregate turbine horsepower can be 25000 HP: and the savings (by this proposed valuation) between 3 and 6 million dollars over a five-year span. Depending upon unique plant economics this implies that replacement of all existing low speed turbines by high efficiency turbines will have a large pay off.

For multistage turbines in the mechanical drive field, optimization of velocity ratio concentrates on modifying the spouting velocity term (denominator) — this is proportional to energy per stage. By optimizing staging, efficiency gains of 5-6 percent are possible in high horsepower turbines. Value of such a gain may be greater than first cost of the turbine — over a five-year period. By careful detail design the added staging appears to be entirely possible within a single casing turbine. Larger exhaust flows, through blading using optimized degree of reaction, may require double flowing to equalize end thrusts and double (tandem) casing designs may be necessary.

Implications to Mechanical Drive Turbine Specifiers

All through this paper, I have been writing of the interaction of economics and turbine design: How the new energy costs have made different design parameters important. In order to take advantage of possible gains in turbine efficiency-performance specifiers must go back to basics, rediscover the art of aerothermodynamic design.

This suggests that more information on staging should be sought. Specifications should permit the flexibility the turbine designer requires to optimize performance. Performance desired, *not* hardware, should be specified.

Again, specifiers must be aware of their plant energy economics such that they can evaluate rationally cost-performance differences in turbines.

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