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DESIGN AND EVALUATION
of
SMALL WATER TURBINES

by

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SUMMARY

An evaluation was made of the design and hydromechanical performance characteristics for three basic turbine types: axial flow (Jonval), inward radial flow (Francis) and crossflow (Banki). A single commercially available turbine representative of each type and within the appropriate power range (<5hp) was obtained for evaluation. Specific turbine selections were based on price, availability and suitability for operation at heads of 50 feet or less and flows under 2 cubic feet per second. In general, the peak operating efficiencies of each unit tended to be lower than anticipated, falling in the range of 40% to 50%. With sufficient flow, however, significant useful power outputs up to 3 hp were obtained. While the radial flow turbine (a centrifugal pump operated as a turbine) had the lowest initial unit cost, the axial and crossflow designs exhibited more stable operation, particularly under transient loadings. The crossflow turbine had the added advantage that it was essentially self-cleaning. With further developmental effort and appropriate design modifications it should be possible to bring each of these microhydro designs to their full performance potential.

MICROHYDROTURBINE DESIGN and EVALUATION

The use of small scale hydraulic turbines for electric power generation is certainly not a new concept. Until the early or middle 1900's with the construction of large, more economical centralized hydroelectric facilities, a significant portion of this country's electric power was provided by relatively small local or municipal plants. Even today in third-world countries hydroelectric units as small as 5KW (6.7HP) are in common use, particularly at remote and rural sites. It has only been in the past ten years that a serious interest has been seen in the U.S. for the redevelopment of microhydropower, sparked by the search for alternate renewable energy sources and the move toward total self-sufficiency among certain sectors of the population. A number of established domestic and foreign manufacturers with extensive R & D facilities currently market a variety of off-the-shelf and custom built small hydroelectric packages. These units tend to fall in the power range well above 10 KW - 50 KW where economic factors affecting marketability are more favorable. On the other extreme, "model" turbines with only fractional horsepower outputs are available for educational and demonstration purposes. A distinct gap exists from 1 to 5 HP ($3/4$ to $3-3/4$ KW), with relatively few vendors offering equipment in this range, a number of whom are not large enough to maintain extensive in-house design and testing departments. At the same time it is just this power range that would be most compatible with potential hydro sites offering heads to 50 ft. and flows to 2.0 CFS, as yet undeveloped, but reasonably common through the southeastern United States.

The primary objective of this project was to evaluate the design and performance characteristics of microhydroturbines suitable for single-user generating applications, for example rural farms and residences, remote park locations or small commercial operations where at least selective or partial electric demand could be met by harnessing nearby springs or streams. Initial intentions were to examine turbine types designed for operation in the range of flows from 1 to 5 CFS and heads from 10 to 100 ft. with generating capacities in the range from 2 to 10 KW. During the course of the project, however, in consultation with individuals from this area of the country who expressed an interest in the implementation of a microhydro generating system, it was found that usable on-site heads and flows in excess of 50 ft. and 2 CFS were uncommon. Consequently, the study was confined to heads below 50 ft. and flows under 2 CFS, which ended up being within the limitations of the test facility constructed for the project (refer to section describing "Test Facility and Instrumentation"). As a result of the evaluations, recommendations were to be made pertaining to the criteria for the use of specific water turbine types for available head/flow combinations. A secondary objective was to reach conclusions concerning the cost-effectiveness of using the various turbine types based on cost of the units, and quantitative and qualitative operating characteristics.

SELECTION OF TURBINES

Conventional turbine types fall into three general classifications: reaction (axial and radial flow), impulse and crossflow. Impulse designs such as the Pelton wheel are best suited for high head (over 50 ft.) and low flow situations, and were dropped from consideration. The selection of specific examples of the remaining types was based on a number of factors including unit cost, availability from the manufacturer and availability within the desired power range. Although radial flow devices such as the Francis turbine typically require high heads (over 100 ft.) for

efficient operation, the decision was made to examine the use of a centrifugal pump operated in reverse as a radial flow turbine for the following reasons: It has been demonstrated that large well-designed centrifugal pumps can be operated efficiently in the reverse mode as hydraulic turbines, and small suitably sized centrifugal pumps in need of modest refurbishing are often readily available as "junk" at scrap cost. The centrifugal pump-as-turbine thus represents the lowest initial cost option. One domestic vendor was identified as a source of small Kaplan fixed-propeller axial flow turbines. It was hoped to obtain a unit of this type for evaluation, but prohibitive cost and an indicated 2 year backlog of outstanding orders forced the decision to obtain a somewhat less desirable Jonval type axial flow turbine as an alternative. Numerous domestic manufacturers are currently marketing Banki crossflow turbines in a wide range of sizes. The particular crossflow unit selected came from a vendor who was willing to supply a unique variable-gate nozzle as a specified option. A detailed description and specifications are presented in the sections that follow for each of the three turbines selected.

In order to evaluate multiple examples of each turbine type, an effort was made to contact various established domestic and foreign manufacturers and to purchase or obtain on-loan suitably sized microhydroturbines from their lines for testing. These efforts were largely unproductive due to prohibitive purchase prices, excessive delivery schedules or, in some cases, disinterest. None-the-less it should be noted that the performance characteristics remain qualitatively similar for size variations in turbines of a given geometrical design.

TEST FACILITY and INSTRUMENTATION

A substantial portion of the effort on the project was spent with the design, construction, instrumentation and calibration of the test facility, which has seen use since completion of the grant work for related studies on microhydroturbines.

A complete description of the facility is presented in the section of this report following "Conclusions and Recommendations".

PERFORMANCE EVALUATION

The maximum theoretical power output of a hydraulic turbine is represented in terms of the hydraulic inputs to the device of head and flow:

$$P(\text{theoretical}) = \frac{\gamma Q H}{550}$$

where

P is power in HP

γ specific weight of water lb/ft³

H net (static) head across the turbine in ft.

550 $\frac{\text{ft-lb}}{\text{sec-hp}}$ conversion constant

The deviation of actual power output from this value is accounted for by "losses" associated with conversion of hydraulic power to mechanical shaft power. Fluid turbulence, viscous fluid friction and bearing/bushing friction are examples of phenomena that contribute to overall energy loss.

$$P(\text{actual}) = P(\text{theoretical}) - \text{losses}$$

The actual power output is measured at the output shaft as a function of shaft speed and torque:

$$P(\text{actual}) = \frac{TN}{63,025.4}$$

where

T is shaft torque in lb-in

N is shaft speed in rpm

63,025.4 $\frac{\text{lb-in-rpm}}{\text{HP}}$ conversion constant

Overall hydromechanical efficiency, η , one of the basic performance parameters for power turbines, is the ratio of actual to theoretical power output and is determined by the combination of fluid head and flow, and shaft speed and torque defining each operating point for the device.

Hydraulic turbines must be coupled to an alternator or

generator in order to transform shaft output to useful electric power. These devices operate either at constant or variable speeds, depending upon electrical load. As a result, output performance data, power and efficiency for the test turbines will be presented as a function of shaft speed for various values of the input variables, head and flow.

The principles of operation and factors leading to nonideal performance in hydraulic turbines are well documented in classical turbomachinery texts [1 - 3].* The complex interrelationships between the potential sources of "loss" in turbomachinery make the accurate prediction of performance for a specific unit extremely difficult based on a purely theoretical analysis. A more practical bases for evaluating the performance of turbines examined in the present study is a comparison with similar, geometrically larger turbines, for which data is already available. It has long been recognized that if geometrically similar turbines, for example scale model and full scale prototype, are tested under dynamically similar conditions the smaller unit will operate at a lower overall efficiency. This relationship is expressed in terms of the Moody equation:

$$\frac{1 - \eta_m}{1 - \eta_p} = \left(\frac{D_p}{D_m} \right)^{0.2}$$

where η is overall hydromechanical efficiency

D is turbine rotor diameter

m and P designate the smaller and larger turbines respectively.

Quantitatively the idea of "dynamically similar" operating conditions for turbines can be expressed in a number of ways, but the basic similarity parameter is "specific speed":

$$\text{Specific Speed} = \frac{N_m P_m^{1/2}}{H_m^{5/4}} = \frac{N_p P_p^{1/2}}{H_p^{5/4}}$$

* Number in brackets represent references listed in the bibliography at the conclusion of this report.

which should remain about constant for two units of different size but similar geometrical configuration. Each type of turbine tends to operate most efficiently within a set range of specific speeds. For example, axial flow turbines operate most efficiently at specific speeds above 90, while for radial flow units the range is from 10 to 90 [1].

In the sections that follow the quantitative and qualitative performance of each of the three turbine types will be presented and discussed individually, using data available for similar units as a basis for comparison.

RADIAL FLOW TURBINE

Small radial flow reaction turbines are currently available as demonstration or "model" units from several domestic and foreign manufacturers: James Leffel and Company, Springfield, Ohio and Gilbert Gilkes and Gordon, LTD, Kendal, England. These Francis turbines have power output in the range of from 1/2 to 2-1/4 HP, with peak efficiencies between 60% and 78%, for low operating heads of under 50 ft. While technically feasible for use in actual microhydro systems, the cost of such turbines is generally prohibitive. One alternative is to use readily available and relatively inexpensive radial flow (centrifugal) pumps as hydraulic turbines, by operating them in reverse. In the reverse or "turbine" mode, water enters the pump discharge section under pressure (head), then passes through the pump impeller where this head is converted to shaft output, and is finally discharged through the pump suction line at low pressure.

It has long been recognized that an efficient, well designed pump will also operate well in reverse as a hydraulic turbine [4 - 7]. The basic physical laws governing the operation of both pumps and turbines are similar and in some cases identical. The peak efficiency of a given centrifugal pump may be as high as 80 to 90% at its design head and flow, sometimes designated "best efficiency point" or BEP. Operated in reverse as a turbine the peak efficiency at the turbine BEP may be as high as 93 percent or more of the pump BEP efficiency [4], although the BEP head and flow in the turbine mode are higher. The conversion is given approximately by the relationships [4, 5]:

$$H(\text{turbine}) = \frac{H(\text{pump})}{\eta(\text{pump})} \quad Q(\text{turbine}) = \frac{Q(\text{pump})}{\eta(\text{pump})}$$

These results have been verified for the operation of relatively

large (>400 HP) low head, high capacity pumps as turbines. These large units tend to have pump impellers that closely approximate the geometry of true Francis turbine runners, i.e. the rotor outside diameter is about the same as the eye diameter. Small medium capacity, medium head centrifugal pump impellers, on the other hand, are more purely radial flow designs, with the rotor outside diameter (discharge), significantly larger than the than the eye diameter (inlet). Present results suggest that this basic geometrical difference may have an effect on the validity of the relationships given above, when applied to small centrifugal pumps. They will be utilized, however, to examine the performance of the centrifugal pump chosen for evaluation in this project.

DESCRIPTION

A 2 HP Worthington 2-1/2 DN62 Monobloc pump, casing designation UH1194, was acquired for evaluation. The pump impeller is a fully-enclosed, backward curved ($\beta = 160^\circ$) six blade design; with a 6-1/4 inch outside diameter, 3/4 inch discharge passage width, 2-3/4 inch eye diameter and inducer section on inlet. The pump casing has 3 inch diameter suction, 2-1/2 inch diameter discharge with no guide sets at rotor discharge.

PERFORMANCE

This particular pump model is no longer in production, and performance curves could not be obtained from the manufacturer. The 2 HP, 4G7-2½ series pump manufactured by Burkes Pumps, Decatur, Illinois is very similar to the Worthington pump evaluated. Performance curves for the Burkes pump [8] indicate a pump BEP of 0.312 CFS and 31 ft of head, with 64% efficiency, at 1750 RPM. Converting this to turbine BEP yields 0.49 CFS and 48 ft head, with 60% efficiency, for an estimated power output of 1.57 HP. This will serve as the basis for comparison of results from the Worthington pump.

An attempt was made to test the Worthington pump as a turbine at specific constant-head settings by stepping the turbine load

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through various speeds from stall to freerun, and adjusting the system resistance to maintain a constant head at the same time. This method proved unworkable due to the relatively unstable head/flow/speed characteristics of this particular radial flow device. Consequently the system resistance was set and the turbine was stepped through a range of speeds from stall to free run. At the same time turbine head and flow were monitored, in addition to shaft torque and speed. An extensive family of head, flow, power and efficiency curves versus turbine speed were generated for the various system resistance settings. By interpolating between curves, sets of constant-head performance curves were constructed, and the peak performance characteristics were determined at specific heads.

The 40 ft and 45 ft constant-head performance curves for the radial flow turbine are presented in Figures 1. and 2. Qualitatively these curves are very similar to those for Francis turbines and large centrifugal pumps operated as turbines. The peak power output occurs at a speed above half the freerun speed. Flow is relatively invariant for speeds up to that for peak power, with a rapid dropoff on out to freerun. Efficiency increases almost linearly with speed up to half of freerun, and peaks at a speed slightly above that for peak power, due to the dropoff of flow in this range. When operating at or slightly above the peak point in the power curve it was found that abrupt reductions in turbine loading cause unstable operation, with the turbine overspeeding toward freerun and then stalling out (stopping). This undesirable operating characteristic might occur in actual service, when there is a sudden drop in electrical demand, unless the turbine is fitted with a mechanical or electromechanical governor to stabilize peak performance (BEP) data for the turbine are presented in Figure 1. for net heads from 37 to 46 ft., where power output is in the full range above 1 HP. For each head the corresponding flow, power, efficiency, peak speed and freerun speed are indicated.

While peak power increases with net head, efficiency remains constant at about 42% suggesting that operation at higher heads above 50 ft will yield greater output, but not improved performance. The turbine BEP estimates of 48 ft head and 1.57 HP shaft output are in line with the results as seen in Table 1, but peak efficiency is well below the estimate of 60% since admission at this head is higher than predicted.

EVALUATION

The overall performance of the centrifugal pump operated as a turbine was adequate, and in reasonable agreement with estimates based on pump characteristics for a similar unit. Its relative instability under conditions of transient loading indicate the need for some sort of speed/load stabilizer. The turbine was operated with vertical intake and horizontal discharge through a relatively long straight pipe section. Were the unit operated with vertical discharge through a short bell diffuser (draft tube), improved pressure recovery at discharge would likely yield better performance.

FIGURE 1

FIGURE 2

TABLE 1
RADIAL FLOW TURBINE

PEAK PERFORMANCE CHARACTERISTICS

H_{PT} (FT)	Q_p (CFS)	P_p (HP)	η_p (%)	N_p (RPM)	N_{FR} (RPM)
46.3	0.716	1.58	42.0	1500	2263
46.0	0.717	1.56	41.6	1500	2256
45.5	0.712	1.53	41.6	1475	2243
45.0	0.713	1.49	41.0	1450	2231
43.3	0.714	1.48	42.1	1420	2188
40.0	0.690	1.31	41.8	1380	2102
36.8	0.665	1.17	42.3	1400	2014

AXIAL FLOW TURBINE

The Jonval turbine, developed in France during the mid 1800s, was the forerunner of more efficient reaction turbines similar to the Francis type designs that emerged during the remainder of that century. Flow is admitted to the Jonval turbine from a radial or axial direction, passes through a guide set and discharged to the rotor inlet with a purely axial throughflow velocity. Axial throughflow is maintained throughout the entire rotor so that the adverse effect of centrifugal forces on the flow becomes negligible, and the difficulty in governing seen for some radial flow turbines is removed (refer to previous section)[10]. Consequently, the axial flow Jonval turbine exhibits highly stable operation, even under transient load conditions. For an axial flow device the Jonval turbine operates at a relatively low specific speed and its peak efficiency tends to be lower than that of other axial flow turbines as a result.

Tests of a 30 inch diameter Chase Jonval turbine, similar in basic geometry to the unit evaluated in the present study, were conducted in 1834 [9] and results indicated peak efficiencies of 76 to 79% at specific speeds in the range of 45 to 51. At a specific speed of 20, however, the peak efficiency for this turbine was only about 44%. Using the Moody equation to scale down the efficiency from this 30 inch turbine to the 4 inch turbine evaluated herein yields estimated efficiencies of 64 to 69% at specific speeds in the range of 46 to 51, but only about 30% for a specific speed of 20. The utility of these figures will be seen in the discussion of performance results for the 4 inch turbine.

DESCRIPTION

The 4 inch Jonval turbine acquired for evaluation in this project is illustrated in Figure 3, with details of the rotor/stator

geometry given in Figure 4. The turbine casing is assembled from inexpensive plastic sanitation-and-drainage (S&D) elements, with the major component being a standard 4 inch S & D 45° wye. The other components of the turbine were sized to fit within the wye. The rotor and stator hubs are disks 3.1 inch diameter by 7/8 inch thick into which 1-5/8 inch radius slots are cut for mounting of the blades. The rotor and stator blades are fabricated from 3-1/4 inch diameter, 0.05 inch thick stainless tubing, and peen-fit into the hubs. By avoiding the use of expensive cast rotors and stators in this fashion, and selection of inexpensive casing components, the overall cost of this unit has been held down. The blading geometry, Figure 4, indicates that this is a 50% reaction turbine, a classification typical of many axial flow devices.

The "flow insert" shown in Figure 3. was not supplied with the unit, but added during testing in an effort to streamline the flow and turn it more uniformly into the guide set (stator). The turbine was tested with 45° inflow and horizontal discharge, as suggested by Figure 3, but can be operated with 45° or vertical discharge by adjustment of its mounting configuration.

PERFORMANCE

During initial tests it was found that turbine stability enabled constant flow conditions to be maintained fairly easily. As a result the tests were conducted by stepping the turbine through various speeds from stall to freerun, and adjusting the system resistance to maintain constant flow. The turbine was only rated by the manufacturer for an electrical output of 0.75KW (=1.0 HP) at a maximum head of 35 ft and 0.6 CFS flow. Consequently, only head/flow conditions extending into this power range were examined.

Plots of power output, net turbine head and efficiency versus speed are presented in Figures 5 through 7. respectively, for four flows from 0.7 to 1.0 CFS. By matching points along the "peak power" curves in Figures 5 and 6, the head and flow combinations

that yield peak output may be determined. As net head is increased peak power and corresponding flow also increase, with the peak in the power curve occurring at a speed slightly below half of freerun, Figure 5. From this figure it is implied that as head is increased power output will increase indefinitely. However, as net head is increased (by virtue of flow) the maximum turbine efficiency increases and then levels off around the 1.0 CFS flow condition, as seen in Figure 7. It is expected, therefore, that at some head above the 1.0 CFS flow condition the maximum efficiency will start to drop off toward zero, as the turbine moves into a lower specific speed range. The peak power curve in Figure 5 will tend to reach a maximum and drop toward zero as more flow is forced through the turbine.

The peak performance characteristics for flows from 0.6 to 1.0 CFS (net peak heads from 12.4 to 37.2 ft) are summarized in Table 2., along with corresponding constant-flow freerun speeds. As head increases so does the turbine specific speed, until a value of 13 is reached at flows of 0.9 and 1.0 CFS. This is a very low specific speed for an axial flow device, and the observed peak efficiencies of around 34% are not unrealistic in light of discussion presented earlier in this section. Table 2 also presents peak characteristics at 1.0 CFS flow for a modified design. As supplied, the leg of the wye through which the turbine power shaft extends, Figure 3, was "open", representing a void space within which a circulating eddy flow pattern could be set up as water passed through the device. With the addition of the flow insert it was hoped that the flow entering the guide set would become more streamlined, and head loss associated with eddy flow could be reduced. The effort was partially successful in that at 1.0 CFS the net dissipated head was reduced from 37.2 feet to 26.6 ft. Peak efficiency and power output decreased, however. This effect was attributed to the abrupt turning of flow into the "top" of the stator, Figure 3, forced by the presence of the flow insert. It is possible that there was little

if any resulting net throughflow at the top of the stator, possibly even flow reversal (pumping).

EVALUATION

The Jonval turbine evaluated showed relatively stable operating characteristics. Power output was in line with manufacturers' estimates, based on operating head, but at significantly higher-than-estimated flows. The single greatest disadvantage to this design was the tendency of small pieces of foreign matter to build up and accumulate on the inlet side of the stator, rather than pass on through. Cleaning required regular disassembly of the unit. As a final suggestion it is felt that performance could be improved by redesigning the rotor discharge section, in order to achieve more efficient diffusion of the flow. In the present configuration there is an abrupt loss-generating expansion of flow as it leaves the rotor.

AXIAL FLOW TURBINE

FIGURE 3

BLADE PITCH (MEAN, 15 BLADES) S , 0.733 INCH
 BLADE CHORD C , 1.055 INCH
 BLADE THICKNESS T , 0.051 INCH
 BLADE HEIGHT 0.40 INCH
 HUB DIAMETER 3.10 INCH
 ROTOR/STATOR TIP DIAMETER 3.90 INCH
 ROTOR/STATOR WIDTH W , 0.875 INCH
 BLADE RADIUS OF CURVATURE 1.625 INCH
 RELATIVE INLET ANGLE $\alpha_1 \alpha_3$, $12\frac{1}{2}^\circ$
 RELATIVE DISCHARGE ANGLE $\alpha_2 \alpha_4$, $34\frac{1}{2}^\circ$
 STAGGER ANGLE β , 34°

AXIAL FLOW TURBINE GEOMETRY

FIGURE 4

FIGURE 5

FIGURE 6

FIGURE 7

Q (CFS)	H_{PT} (FT)	P_p (HP)	η_p (%)	N_p (RPM)	N_{FR} (RPM)
1.00	37.2	1.42	33.7	1400	3120
1.00*	26.6	0.88	29.2	1200	2590
0.90	29.5	1.01	33.6	1250	2765
0.80	22.5	0.65	31.9	1075	2409
0.70	16.8	0.40	30.0	900	2054
0.60	12.4	0.225	26.7	700	1690

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CROSSFLOW TURBINE

The Banki crossflow turbine is a unique radial flow device in which the water makes two passes through the rotor blading, the first radially-inward where about 75% of the energy transfer takes place and the second radially-outward where the remaining energy transfer is accomplished. With reference to Figure 8., operation of the turbine may be described as follows: From a supply line water enters a converging nozzle where static head is converted to dynamic head. A relatively thin, wide high velocity rectangular jet leaves the nozzle, and enters the blades on the outer periphery of the wheel at a shallow tangential angle. Flow proceeds radially-inward through the blade passages, passes across the "empty" hub space defined by the blades' inner diameter, strikes the inner periphery of blades on the other side of the wheel, passes radially-outward and is discharged in a broad, low energy jet from the outer periphery of the wheel. Variations on the basic crossflow design include fixed and variable gate nozzles, and "open" or closed (full-cased) housing with draft tube.

European designs, most notably those of the German manufacturer Ossberger-Turbinenfabrik, have reached a high level of technical sophistication, with the incorporation of complex variable-geometry nozzles for load matching and full admission/discharge capabilities with draft tubes for optimum net head utilization. Such designs are highly efficient, at better than 80%, as they achieve some energy transfer through reaction in addition to their primary impulse mode of operation. The cost of these units is relatively high, however, and they are generally offered for applications exceeding 10 HP.

The simpler open crossflow designs usually consist of a rectangular fixed-geometry nozzle that discharges to an open rotor.

This is to say that flow through the entire rotor is essentially at ambient conditions. Consequently, energy transfer is almost entirely by impulse action rather than reaction. Turbine gating is usually accomplished with a valve in the supply line or a simple slide gate in the nozzle, to reduce the discharge jet width. Although the crossflow design is an impulse turbine it is particularly well suited for operation at heads below 50 ft, unlike other tangential flow impulse turbines such as the Pelton wheel. Nozzle dimensions, width and height at discharge, are determined by the desired design head and flow, with width primarily a function of design flow. While the rotor outside diameter is essentially independent of design flow, once this diameter is specified the remaining geometry for the runner (blade curvature, runner I.D., incidence and discharge angles for example) are determined from the discharge velocity and angle from the nozzle, and design speed. The design details commonly followed in the specification of these turbines are presented in [11].

Mockmore and Merryfield [11] designed, built and tested a small 13 inch diameter open crossflow turbine, similar to the unit evaluated in the present study, intended for use at 16 ft. of head and 3 CFS. The BEP for the turbine under 16 ft of head occurred at 2.22 CFS and 270 RPM, with a power output of 2.75 HP and corresponding efficiency of 68%. The specific speed was about 14, but efficiency was found to be satisfactory over a range from 6 to 16, which represents the suggested range of application for crossflow turbines.

DESCRIPTION

The crossflow turbine obtained for evaluation was an 11.2 inch diameter, 18 blade design with a rectangular nozzle and a unique circular, variable area gate. Details and dimensions are shown in Figure 8. for the nozzle and blading geometry. The turbine nozzle is fed from a flanged 6 inch diameter inlet. Nozzle discharge dimensions are 4.0 inch height (measured vertically) by 4.75 inch width, while the runner width is 6 inches.

As supplied by the manufacturer the turbine was to have a design head and flow of 35 ft. and 1.5 CFS, parameters that are largely independent of the rotor configuration for impulse turbines. The relationship between head and flow for a nozzle is given by:

$$Q = AC(2gH)^{1/2}$$

where

H is total head at inlet, ft.

g is approximately 32.17 ft/sec²

C is a nozzle discharge coefficient = 0.98

A is minimum nozzle area, ft²

Q is nozzle flow, CFS

The required nozzle discharge area for this turbine is then approximately 0.0323 ft². With a nozzle width of 4.75 inches the corresponding full gate nozzle height should be about 1.0 inch, considerably less than that supplied. As a result it was only possible to run the turbine at part-gate openings without flow exceeding the capabilities of the water supply system (1.9 CFS).

PERFORMANCE

Within the head and flow limitations of the water supply system the crossflow turbine was evaluated at each of three heads for gate settings of 20%, 30% and 40% and one head for 50% gate. Power curves and a tabular summary of peak performance characteristics are presented in Figures 9 through 12. Referring to Figure 8, full gate corresponds to a 42° gate opening, 50% gate corresponds to 21° gate opening and so on. Tests were conducted by setting the turbine gate and the system resistance, then operating the unit at speeds from stall to freerun via load adjustment.

Less than 3% variation of head and flow was seen over the entire speed range, with a tendency toward higher flows and lower net heads as speeds increased. This characteristic is typical of impulse turbines, as the head/flow interrelationship is primarily a function of nozzle geometry and not runner speed. Total net

head is proportional to the square of flow for a given nozzle configuration as illustrated in Figure 13 for this particular crossflow turbine. Due to the relative invariance of head or flow with speed, efficiency curves follow the performance curves very closely, with maximum efficiency occurring at essentially the same speed as peak power. Two head values are presented in Figures 9 - 12; H_{PPR} is peak static component of pressure at nozzle inlet and H_{PT} is the peak total pressure at the same location. H_{PT} is used to compute peak efficiency.

At each gate setting examined, both peak output and efficiency were seen to increase with increasing net head, indicating that for gates from 20% to 50% the maximum efficiency and power points occur at heads above the range tested. Peak power conditions tended to fall at, or above half the freerun speed, which is common for crossflow turbines. Turbine specific speeds varied from 6 to 12 at peak conditions, within the range of application for this design.

Although a peak efficiency of 50% was observed at 43 ft. of head and 20% gate, this and the other values fall well below the potential 70% to 80% for crossflow units. Referring to Figure 8. two possible explanations can be given:

- 1) At partial gate openings flow reaches a maximum velocity and kinetic energy at or slightly beyond the gate lip, where the cross sectional area is a minimum. Beyond this point there is a potential for the flow to diverge, with subsequent loss of kinetic energy, before it enters the turbine runner. There is less hydraulic energy available to the rotor for conversion to shaft output.
- 2) Also due to partial-gate geometry, as flow passes up over the gate it may be deflected from the top of the nozzle down into the rotor at a nonoptimum angle, resulting in reduced performance.

The turbine gate has recently been modified to a fixed-nozzle configuration, with a simple 18° inclined flat plate extending

to the outer periphery of the rotor. Initial results indicate a significant improvement in performance.

EVALUATION

Performance of the crossflow turbine was adequate, considering the difficulties associated with being able to run only at part gate settings. Reduction in the dimensions of the variable gate or substitution of a smaller fixed gate would likely increase output and efficiency significantly. The basic design is rugged and reliable, and can be easily sized to meet specific low head applications. Two additional advantages of the crossflow design are its stability under transient load conditions and the fact that it is self-cleaning.

FIGURE 8

	TURBINE SPEED - RPM						
	H_{PT} (FT)	H_{PPR} (FT)	Q_P (CFS)	P_P (HP)	N_P (RPM)	N_{FR} (RPM)	η_r (%)
I	43.2	42.6	1.19	2.61	550	925	49.9
II	36.5	36.0	1.10	2.07	500	845	45.5
III	29.3	28.9	0.97	1.21	460	756	37.6

FIGURE 9

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ED - RPM

P_P (HP)	N_P (RPM)	N_{FR} (RPM)	η_r (%)
3.00	460	885	42.3
1.86	390	765	41.6
0.65	260	631	33.2

	H_{PT} (FT)	H_{PPR} (FT)	Q_P (CFS)	P_P (HP)	N_P (RPM)	N_{FR} (RPM)	η_r (%)
I	33.8	32.3	1.95	3.06	450	845	42.2
II	21.4	20.4	1.61	1.60	350	685	39.9
III	10.7	10.1	1.24	0.44	250	507	29.3

FIGURE 11

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P_R (Γ)	Q_P (CFS)	P_P (HP)	N_P (RPM)	N_{FR} (RPM)	η_P (%)
4	1.75	1.39	330	605	42.6

FIGURE 12

FIGURE 13

POTENTIAL OF MICROHYDRO SYSTEMS

The results of this project clearly demonstrate that micro-hydro turbines covering the 1 to 5 HP range do, in fact, work and will produce useful power output given a suitable source of water with appropriate available head (elevation) and flow. The specific units evaluated in the present study were selected with initial cost as a primary consideration. Consequently, the design details reflect the use of less expensive (but not unreliable) fabrication techniques, eliminating for example the use of more costly cast turbine stators and rotors in the axial flow turbine. The relatively low peak hydromechanical efficiencies observed were due, in part, to a lesser degree of design sophistication. More sophisticated units with better hydro-mechanical performance are available from some domestic and foreign vendors, at a cost. The premise that "you get what you pay for" applies in the area of microturbine selection. If a more than ample supply of water is available and the electric demand to be met is not excessive, then a less efficient but lower cost unit merits every consideration. From a philosophical standpoint it would seem better to derive some use from a suitable existing hydrosite, one that can be developed, than to do nothing with it, particularly if the installation can be justified on an economic basis.

One essential aspect of a complete microhydro generating system that was outside the scope of the present study is the selection of electrical equipment for the conversion of shaft power to electrical power, switching and load control. Most manufacturers of microhydro equipment currently offer complete packaged systems,

including the turbine and matched electrical components, to meet a range of load requirements for synchronous or nonsynchronous, AC or DC power generation. In line with the conclusions drawn from this project, most manufacturers have chosen the crossflow type turbine for their low head (<50 ft.) packages, due to simple design, rugged reliability, stable operation, ease of maintenance and moderately good efficiency. A partial list of vendors for microhydro systems in the 1 to 5 HP range would include, but is not limited to the following:

CANYON INDUSTRIES
5346 Mosquito Lake Road
Deming, Washington 98244

HYDRO-WATT SYSTEMS
146 Siglun Road
Coos Bay, Oregon 97420

INDEPENDENT POWER DEVELOPERS
Route 3 Box 285
Sandpoint, Idaho 83864

Upon request these companies will generally supply information on how to determine if a given hydro site has development potential and how to measure available head and flow on-site, in addition to a description of their product lines. A number of government publications, References [12] and [13], are also useful and informative reading for individuals considering the installation of a microhydro system.

Small hydroturbine-driven electric generating systems have the potential to make effective use of a largely untapped renewable energy resource. A 1 to 5 HP turbine coupled to the required electrical generating equipment would be expected to deliver somewhere in the range of 1/2 to 3 KW of power on a continuous basis, due to losses associated with conversion of shaft output to electrical output. While 3 KW is adequate to meet the complete average daily electric demand of a small residence, peak daily loads are much higher. Unless relatively expensive electrical storage capabilities (such as a DC battery bank) are

built into the hydro system to buffer peak demand conditions, the application of 1 to 5 HP microhydro systems should be confined to predictable nonfluctuating electrical loads. Additionally, the expense of equipment needed to tie into and synchronize with existing power grids seems to suggest that this size system be used with separate stand-alone electrical loads. As with most nonconventional alternate energy devices and systems, pumped solar water heaters for example, a significant degree of "user involvement" is necessary with a microhydro system to cover such activities as routine system maintenance and troubleshooting, start-up and shut-down, and load management. All of these factors taken together indicate the type of applications and individuals best able to be effectively served through the use of a microhydro system, for example:

- 1) Rural farms, to meet partial residential demand, for lighting of barns and other outbuildings, to run well pumps or other low load appliances, to power shop facilities or other farm operations.
- 2) State or federal parks, to power park lighting, rest stations or visitor centers, particularly in remote locations.

Finally, once installed and operational, a microhydro system can serve as reliable emergency power source in the event of electrical outages.

CONCLUSIONS AND RECOMMENDATIONS

The use of small 1 to 5 HP hydraulic turbines for power generation is technologically feasible and a viable approach to the development of alternate renewable energy resources. The past several years have seen improvements in the quality and performance of equipment offered by vendors in this power range.

COST

Although the cost of the basic hydroturbine is only a fraction of the cost for a complete, installed generating system, it is not insignificant. The abundance of available head and flow at a prospective hydro site, along with the need or lack to make most efficient use of the water supply to meet desired electrical demand will determine in large part the amount to be spent on the basic turbine. Where particularly high performance is required the cost of the unit is generally greater. By examining the cost of each of the turbine types evaluated and their relative performance under similar operating conditions some conclusions can be drawn. Considering a 40 ft. net head, with 30% gate on the crossflow turbine, the following results:

TYPE	PRICE	PEAK POWER	\$/HP
Axial flow	\$ 750	1.58 HP	475
Cross flow	2980	3.16 HP	943
Radial flow	50*	1.13 HP	38

* cost as surplus equipment

With the lowest \$/HP value the radial flow turbine would appear to be an appropriate choice. The need to design and install a

mechanical or electromechanical speed controller with the radial flow turbine, in order to handle transient load instability, would raise the cost considerably, however. Also, to achieve higher horsepower output from this unit would necessitate operation at higher heads, above 50 ft. These factors would tend to offset the attractiveness of the low initial cost.

The axial flow turbine with an intermediate \$/HP value would be suitable for heads below 50 ft and conditions of relatively low available flow, but the complete unit would need to be redesigned in order to generate higher output. It is questionable that operation of this unit at higher heads would produce a significant increase in power output, since this would move its operation into a lower, even less efficient specific speed range. Another disadvantage of this design is the tendency for trash to accumulate and build up on the inlet guide set (stator) with a resulting reduction in performance.

Although the crossflow turbine has the highest \$/HP rating, this value could be reduced to the range of that for the axial flow device with a simple design modification. The use of a fixed, rather than variable area nozzle configuration would both reduce the production cost for the unit, and improve its peak output through increased efficiency. Several desirable qualitative characteristics of the crossflow design should be weighed against cost: reliability, stability of operation and low maintenance.

PERFORMANCE

While the overall performance of all turbines evaluated was less than desired, based on peak hydromechanical efficiency, it was adequate to produce usable amounts of shaft output. The basic principle for operation for all hydroturbines is to convert the potential energy or "head" of a column of fluid to shaft output. This conversion may be accomplished more or less directly as in a reaction (Francis) turbine, or the fluid head may first be transformed to kinetic energy as in the nozzle of a crossflow

(Banki) turbine before conversion to shaft output through the turbine runner, i.e. impulse operation. Regardless, the net fluid head, H , available to the turbine is the primary input variable for the turbine. How well the head is converted to shaft output is a measure of the performance of a given device. (Available flow is primarily a determinant of size and power range for a given design). Again, comparing data for the three test turbines at 40 ft head, with 30% gate on the crossflow unit, the following results:

TYPE	PEAK POWER	HP/FT	FLOW	η
Axial flow	1.58 HP	0.04	1.18 CFS	34%
Crossflow	3.16 HP	0.08	1.68 CFS	42%
Radial flow	1.31 HP	0.03	0.69 CFS	42%

The crossflow design has the highest HP per FT of head rating, but achieves this through the utilization of higher flow than the other turbine types. The peak efficiency of the crossflow unit at 40 ft. head and 30% gate is not significantly higher. It should be noted that a peak efficiency of 50% was obtained for this turbine at a head of 43 ft. and 20% gate, considerably higher than the best efficiency for the other two turbine types, within the range of heads examined. As part of the experimental studies for an M.S. Thesis currently underway at Tennessee Technological University, the crossflow turbine evaluated in this project has been modified to a fixed-nozzle configuration. Preliminary results indicate a significant improvement in performance over the variable nozzle design, with peak efficiency approaching 70% or better under head/flow conditions similar to the present study.

For the axial flow Jonval turbine, the addition of a diffuser cone to the discharge side of the rotor hub would likely provide a degree of pressure recovery for the flow leaving the turbine. While geometrically simple and likely to yield some increase in overall unit efficiency, this modification might prove costly, as it would require a change in the fabrication sequence for the rotor.

COMMENDATIONS

Aside from the conclusions and recommendations concerning design evaluation and design modification just presented, there are a number of specific recommendations to be made for the potential user of a microhydro system (in the 1 to 5 HP range, with heads below 50 ft). Once the usable available head and flow for the site have been determined and the electrical load conditions have been specified in detail (KW demand, AC or DC power, etc):

1. For flows under 1.0 CFS and heads from 40 to 50 ft: a small axial or radial flow device would be the appropriate choice, despite relatively low efficiency.
 2. For flows in excess of 1.0 CFS and heads from 15 to 50 ft: a fixed-nozzle crossflow turbine would be the best choice. It is reliable, stable, rugged and self-cleaning.
 3. Purchase a complete system package from a reputable manufacturer. While a basic crossflow turbine, for example, can be built in a conventional machine shop (lathe, mill, welders, etc.), a good technical background is necessary to make the proper selection of matched electrical components for a system.
- Insofar as possible the system, and particularly the turbine, should be custom built or matched to the users specific available head and flow conditions. One "stock" turbine will generally not perform well over a wide range of heads and flows.
- A system in this small a size range is best suited for use with predictable, stable stand-alone electrical loads. The benefits of tieing into a central power grid (power sellback option) are not justified with respect to the added cost of the system components.

TEST FACILITY AND INSTRUMENTATION

The first step in the project was the design and construction of a test facility suitable for evaluation of the turbines within the desired head and flow range. In order to hold down the cost of the facility the decision was made to use an open gravity-fed water supply system with recirculating pumps rather than a closed pressurized system. The major tasks in completing the system were 1) assembly of the hydraulic components; recirculating pumps, head tanks and turbine supply and discharge lines, 2) design and construction of an absorption unit for dissipation of turbine shaft power and 3) instrumentation of the system for head, flow, torque and speed measurements. The test facility and associated instrumentation are shown in Figures 14 - 16.

WATER SUPPLY and RECIRCULATING SYSTEM

A gravity-fed supply system required the use of an open supply reservoir to serve as a "headwater" and an open discharge reservoir to serve as a "tailwater". With reference to Figure 14, these purposes were served by two 600 gallon capacity water holding tanks with a free surface elevation of 164.7 feet and a 4 x 9 x 22 ft. concrete pit with a free surface elevation of 112.5 feet. This arrangement provided a maximum turbine elevation head of 52.2 ft. for the closed turbines (axial and radial flow). The maximum available elevation head for the open crossflow turbine was approximately 48.8 ft. Delivered head to each turbine was of course less due to flow losses, and could be controlled through a range of values by placement of orifice plates in the 6 inch turbine supply line or adjustment of a 4 inch gate valve located in the supply line upstream of the turbine. Discharge water from the turbines was recirculated to the 600 gallon holding tanks by two centrifugal pumps, 7½ HP and 15 HP, Figure 16, that could be operated singly or in parallel depending upon the desired flow rate range. Operated together the pumps were capable of supplying

the turbine with 853 GPM (1.9 CFS) on a continuous flows in excess of 2 CFS for short periods of time draindown of the holding tanks. Pump output that v by the turbines passed over a weir in one of the 60 holding tanks and returned directly to the discharge via a 4 inch overflow line.

The test facility was constructed at a university stadium with the water holding tanks located at the stadium bleachers and the discharge pit, pumps, turbine instrumentation housed in a basement level machine room the stadium. Due to constraints on the location of the facility and availability of suitable operational centrifugal pumps (one of which was donated for the project by local industry), the maximum possible head and flow for testing was less than originally intended for study. During the course of the project, however, in consultation with individuals from the area of the country who expressed an interest in the installation of a microhydro generating system, it was found that on-site heads and flows in excess of 50 feet and 2 CFS were uncommon.

ABSORPTION UNIT

An artificial loading device or "absorption unit" was required as part of the test facility in order to dissipate power developed by the turbines, in much the same fashion as a generator and electrical load in an actual hydroelectric plant. Depending upon nominal horsepower rating (size) and type of turbine operated within a specific range of speed and torque combinations. A single conventional electric, viscous or friction type loading device (dynamometer) was not suitable for use on all the turbines, because such absorption units typically provide loadings over a limited range and can not be run continuously. Consequently, a special hydraulic absorption unit was designed and built as part of the project that could operate continuously and cover the required range of torques and speeds for the turbines that were evaluated.

The completed unit is illustrated in Figure 15. A photograph

displacement gear pump is coupled to the turbine power shaft through a torque/speed sensor. As the gear pump is driven by the turbine, oil is forced through an adjustable flow control valve producing an artificial pressure drop or load across the pump. The oil then passes through an oil-to-water heat exchanger where viscous heat generated by the pumping is dissipated. By using either a 3.9 cubic inch or 0.7 cubic inch displacement gear pump in combination with a variety of oils ranging from SAE 10 to SAE 50, the specific torque/speed loading range appropriate to the individual turbines and head/flow conditions could be met.

INSTRUMENTATION

In order to evaluate turbine performance four basic measurements were required: turbine shaft speed and torque delivered to the absorption unit and head and flow available to the turbine. Speed and torque were used in the computation of actual turbine power output, while turbine differential head and flow were necessary, in addition, to compute hydromechanical efficiency based on maximum possible power. The instrumentation layout is illustrated in Figures 14 - 16 and the page following.

Torque and speed measurements were made with an S. Himmelstein and Co. MCRT 24-02T(1-3) torque/speed sensor rated to 15,000 RP, and 1,000 lb-in. coupled to a Lebow Associates 7555 Transducer and Digital Indicator (torque readout) and a General Radio 1191-B Digital Counter (shaft speed readout). The torque/speed sensor was interposed between the turbine power shaft and the gear pump drive shaft, Figure 15, to pick up shaft power developed by the turbine less coupling losses which were assumed minimal. By decoupling the sensor from the absorption unit freerun speeds could be measured, while a positive lock on the gear pump shaft enabled measurement of stall torques. For each data point (measurement) a 10 second running average was taken for the speeds and a similar visual average was estimated for the continuous display of torque. The calibration was periodically checked and updated for this and all other instrumentation during the course of the tests.

An in-line 6 inch Potter-Aero 6-50349 turbine type flow-meter, rated to 3600 GPM, was placed in the turbine supply line and used in conjunction with a Hewlett-Packard 523-C Electronic Counter to measure volumetric flow. The flowmeter was calibrated in-situ for flows up to 1.5 CFS by a modified collection technique. Owing to the high linearity of the resulting calibration curve ($r = 0.999$) extrapolation on out to 2 CFS seemed justified. Each flow measurement was taken as a 10 second running average.

Turbine differential pressures were measured using a Trimount H22-50-S-3, 50 inch well type mercury manometer. Flush wall-mounted pressure taps were placed on the turbine supply and discharge lines at appropriate positions upstream and downstream of the turbine. Flow straighteners and long straight pipe runs, which would have enhanced accurate differential pressure measurements, could not be used due to space constraints with the system. Resulting flow swirl at the measurement points would tend to put the measured pressures somewhat in error. It was estimated that this effect would be minimal, however, and on the conservative side with respect to efficiency computations. Measured differential pressures were corrected to account for differences in inlet and discharge pipe diameters. For the open crossflow turbine the discharge pressure was taken as ambient and a single inlet pressure, rather than differential pressure, was measured.

FIGURE 14 SCHEMATIC OF TEST FACILITY LAYOUT (not to scale)

TOP VIEW SIDE VIEW

FIGURE 15

ABSORPTION UNIT FOR CONTROLLING
TURBINE LOAD

Figure 16 Schematic of Test Facility
Equipment and Instrumentation.
(Legend on Following Page.)

LEGEND FOR FIGURE 16

- 1 Pump foot valves
- 2 Durco centrifugal pump: Type 4x3 10/100, 15 hp, rated 350 gpm at 1750 rpm and 78 ft. head (for water recirculation)
- 3 Federal 3x2-1/2 centrifugal pump: Type B34668, 7-1/2 hp, rated 270 gpm at 1750 rpm and 75 ft. head (for water recirculation)
- 4 Turbine discharge water level.
- 5 Turbine discharge line (draft tube), if so equipped.
- 6 Turbine
- 7 Turbine supply flow control and shutoff valve.
- 8 Turbine supply line (penstock)
- 9 Potter-Aero 6" turbine flowmeter, Model 6-50349, rated to 650 cps and 3600 gpm.
- 10 Trimount type H22-50-5-E mercury manometer, 50 inch scale (turbine differential pressure measurement)
- 11 General Radio, Model 1191-B Digital Counter (turbine shaft speed readout*)
- 12 Hewlett-Packard Model 523-C Electronic Counter (turbine flow readout)
- 13 Lebow Associates Model 7555 Transducer and Digital Indicator (turbine shaft torque readout*).

* Input signals from an S. Himmelstein and Company Model MCRT 24-02T(1-3) Torque/Speed Sensor, rated to 15,000 rpm and 1,000 lb-in.

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