



MICHELL-BANKI TURBINE

1. General Characteristics

The crossflow turbine or Michell-Banki turbine is primarily used for small hydroelectric utilities. Its principle advantages are its simple design and ease of construction and its main attraction lies in its low cost and potential to be used in small-scale operations.

However, this does not prevent the use of this type of turbine in large-scale installations. Although the turbine is best known in small-scale applications, this kind of machine also exists in schemes up to 6 MW.

The principle characteristics of this machine are:

- It is adaptable to a wide range of spin speeds
- The diameter of the turbine does not necessarily depend on the volume of the flow
- One can reach an acceptable level of productivity with small turbines
- It is possible to regulate the volume and power of the flow by way of an adjustable flier

2. Working Principles

The turbine consists of two principle elements: an injector and a rotor. The water is returned through a flushing mechanism at atmospheric pressure. The rotor is composed of two parallel discs, joined together by blades curved in circular sections.

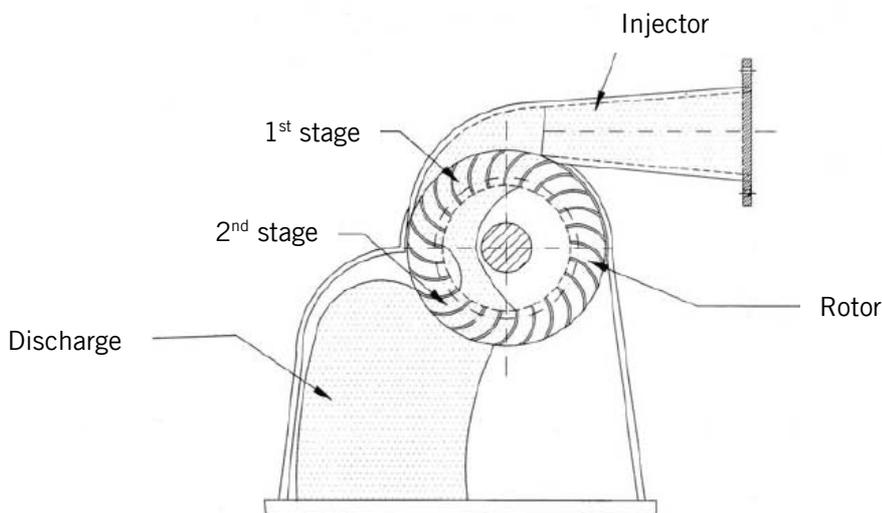


Figure 1: cross section of a Michell-Banki turbine

The nozzle has a cross-flow, rectangular section that is joined to the pipeline by a rectangular-circular transition. This nozzle directs water towards the rotor through a section which accesses a determined quantity of blades, and which guides the water so that it enters the rotor at a

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predetermined angle that ensures the greatest possible production of energy.

The energy from the water is transferred to the rotor in two stages, hence the name of the machine as the 'turbine of double effect'. The first stage on average transfers 70% of the total energy to the rotor whilst the second transfers the remaining 30%.

(Research conducted by different researchers places the hydraulic performance as being between 65 and 70%, whilst others state 61%, stressing that the second stage transfers only 17%. Mainly, literature indicate 70% up to 84%)

An attractive feature of this machine is its flat performance curve. This is achieved by designing the turbine for partial admission. For example: if the rotor is divided into three equal parts, the admission of water can be controlled by allowing it to enter 1/3, 2/3 or the entire wheel.

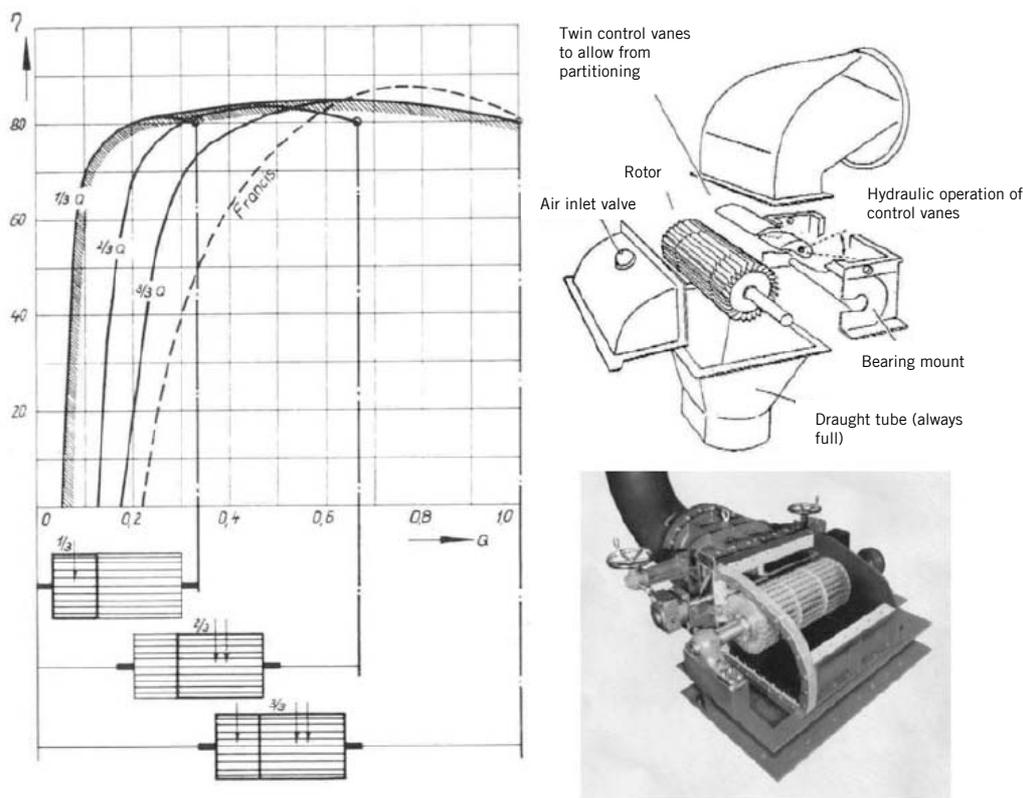


Figure 2: Partial admission cross-flow turbine

This type of admission is what enables the performance curve shown in Figure 2, where a comparison with the performance curve of the Francis model of turbine can be observed.

As has been previously mentioned, the cross-flow turbine is particularly appropriate for rivers with small flows.

For several months a year, these rivers carry very little water. Therefore, the turbine design is able to take into account the minimum flow of water by partial loading and in times of abundant water, the whole of the turbine will be used to deliver productive energy.

It can be seen from the characteristic performance curve of each turbine that the Michell-Baniki

3. The selection of the diameter and spin speed of the turbine

To follow on from what was stated earlier, the diameter of this machine is not dependent on the flow of water which allows a design with parameters that are independent of the flow of water. This also means the turbine can continue to function within a wide range of different spin speeds. As a result, the diameter of the turbine can be selected to begin with.

Many researchers indicate that diameters commonly used for small scale hydroelectricity are between 200 and 500mm. The standardized diagrams which are offered in the bibliography are frequently taken and used to select the diameter as the basis of the design.

Some of the factors that have to be taken into account are as follows:

- In reducing the diameter of the rotor, one reduces the hydraulic performance
- The biggest rotors have a lesser runaway speed
- A certain proportion between the diameter and the width of the turbine needs to be maintained in order to avoid excessively fast section changes between the ducts and the nozzle, which would provoke strong reverberations in the moving fluid.
- The parameters of the design need to be selected in order to allow simple manufacture of the turbine

Once the diameter has been selected, the spin speed of the machine can then be determined:

$$n = 40.62.k_c.\frac{\sqrt{Hn}}{D}$$

4. Number of blades on the rotor

The number of blades on the rotor is determined on the basis of the diameter and the working conditions of the turbine, that is to say the **height** between the reservoir and the turbine and the water **flow**.

One should keep in mind that a reduced number of blades can produce pulsations in the generation of power, and more blades will produce an acceleration in the flow with the ensuing rise of mechanical losses.

According to the research of A. Ulka and H. Olgun, there is an optimum number of blades. Table 1 shows the results of different research reflected in the literature documented.

Table 1: Characteristic parametres of turbines studied in different investigations

Reference	D/d	D/B	Z	n
Yokohama, 1985, Japan	0.66	4.25	26	80.60
Vigm, 1986, CCCP	0.63	3.00	24	78.00
Ganz, 1984 Hungary	0.66	1.00	30	75.00
Alabama 1983 USA	0.66	0.25	20	75.00
Resita, 1983, Rumania	0.66	1.28	24	75.00
KTU, 1987 Trazbon, Turkey	0.54	0.81	24	71.30
Oregon, 1949, USA	0.66	1.09	20	68.00
VDI, 1981, Ethiopia	0.67	3.26	36	66.00
Los Andes, 1973, Colombia	0.62	1.87	27	60.60
ODTU, 1985, Ankara, Turkey	0.83	1.44	30	55.50

In this table it can be seen that the optimum number of blades is between 24 and 30.

5. Calculation of the width of the rotor

An important factor to keep in mind is the calculation of the width of the rotor. This is calculated with the selected diameter and the functioning parameters H and Q in mind.

As we know the area of admission is given by:

$$A_a = \frac{Q}{C_{ml}} \quad (1)$$

Followed by

$$A_a = B \cdot p_z \cdot Z_a \quad (2)$$

The step P, is given by:

$$p_z = \frac{\pi \cdot D}{z} \quad (3)$$

If we define

$$x_z = \frac{Z_a}{z} \quad (4)$$

Like the ratio between the number of blades for admission and the total number of

blades, which we will have after selecting the number of blades of the rotor, we could determine the number of the admission blades, hence the value of x_z

This number of blades already has a close link with the angle of admission of the machine.

The coefficient x_z varies between 0.05 and 0.35. As a limit one adopts:

$$x_z = \frac{Z_a}{z} < 0.35 \quad (5)$$

Equalizing (2) and (3) and resolving B we have:

$$B = \frac{Q \cdot z}{\pi \cdot D \cdot c_1 \cdot \sin \alpha_1 \cdot z_a} \quad (6)$$

In the end, what remains is the formula to calculate the width of the rotor:

$$B = 0.259 \cdot \frac{Q}{k_c \cdot D \cdot \sqrt{H_n} \cdot x_z} \quad (7)$$

6. The design of the nozzle

The second component of this machine is the nozzle, which together with the rotor determines the efficiency of the turbine.

It is responsible for directing the flow of water towards the rotor. This transport should possess a good acceleration and an even distribution of speeds in the output section like a low level of load loss, so as to achieve the best possible transformation of potential energy into kinetic energy.

The nozzle can have distinct dimensions fundamentally differentiated by the angle of admission and the regulating valve which it has, if indeed it has one.

One should keep in mind the placement of this regulating valve, which regardless of the element in the interior of the nozzle can provoke disturbances in the exit flow.

The nozzle possesses a rectangular cross section composed of two straight lateral surfaces which permits the discharge of the flow across all of the rotor (figure 3), and an upper bound face that guides the flow.

This surface possesses an optimum, constant angle on each point of the curve. The absolute speed will be at a tangent to this curve at every single point. The surface below is straight and can have an angle of 5 degrees maximum.

Further information

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technical brief