MANUAL ON DESIGN, STANDARDIZATION AND MANUFACTURE OF EQUIPMENT FOR SMALL HYDRO POWER STATIONS

VOLUME I

DESIGN, STANDARDIZATION AND MANUFACTURE OF MICHELL - BANKI TURBINES

PLACE
LATIN AMERICAN ENERGY COOPERATION PROGRAM
INSTRUMENT TO STRENGTHEN OLADE

olade
LATIN AMERICAN ENERGY ORGANIZATION

MANUAL ON DESIGN, STANDARDIZATION AND MANUFACTURE OF EQUIPMENT FOR SMALL HYDRO POWER STATIONS

VOLUME I

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INDEX

INTRODUCTION 7

1. GENERAL DESCRIPTION AND DESIGN PARAMETERS 9

2. HYDRAULIC CALCULATIONS AND DESIGN 15

   2.1. Speed Diagrams 15
   2.2. Injector Geometry 18
   2.3. Runner Geometry 19
   2.4. Casing Geometry 28

3. DETAILED DESIGN AND MECHANICAL CALCULATIONS 30

   3.1. Design and Calculation of the Injector 30
   3.2. Design and Calculation of the Runner 34
   3.3. Design and Calculation of the Shaft 38
   3.4. Design of the Bearings Support 39
   3.5. Design of the Regulating Mechanism 41
   3.6. Design of the Casing 44

4. STANDARDIZATION AND SELECTION 45

   4.1. Standardization 45
   4.2. Selection and Relocation 47

5. MANUFACTURING RECOMMENDATIONS 52

   5.1. Manufacturing the Injector 52
   5.2. Manufacturing the Runner 53
   5.3. Manufacturing the Main Shaft 54
   5.4. Manufacturing the Bearings Supports 54
   5.5. Manufacturing the Regulating Mechanism 54
   5.6. Manufacturing the Casing 55

APPENDIX 1 56

APPENDIX 2 65

BIBLIOGRAPHY 116

FOREWORD

Considering that new and renewable sources of energy can play an important role in the energy panorama of Latin America , in the framework of the Latin American Energy Cooperation Program (PLACE), approved at the XII Meeting of Ministers of OLADE, guidelines were laid out for the promotion of their exploitation in the regional countries, particularly in those that are relatively less developed.

Hydroenergy presents itself as the most important and most easily-accessible alternative for the countries of the region; and consequently, it has deserved top priority in the PLACE, so that OLADE can carry out activities geared to accelerating development.

It is in this context that OLADE activities are developed in the field of small hydro power stations (SHPS), whose importance lies both with hydroenergy development as well as with the promotion of the use of diverse new and renewable sources of energy.

The Regional SHPS Program of OLADE includes various activities aimed at expediting small-scale hydroenergy developments in those aspects related to resource assessment; development planning; design and engineering for plant construction; definition of institutional schemes; orientation with respect to SHPS operation and maintenance; and particularly, support for the countries in obtaining technologies for the design and manufacture of equipment.

The availability of technologies suitable for regional realities, as well as the experience accumulated in this regard by several Latin American countries, will make it possible to develop technology for equipment manufacturing in the countries of the region. In this respect, as part of the PLACE activities, the Permanent Secretariat of OLADE is elaborating a nine-volume manual entitled "Manual on Design, Standardization and Manufacture of Equipment for Small Hydro Power Stations", the first volume of which, referring to Michell–Banki turbines, constitutes the material of the present document.

The preparation of this volume by OLADE responds to the need to provide and disseminate the technological bases required to produce cross-flow turbines, better known as Michell–Banki turbines; and it contains the necessary technical criteria for designing and manufacturing standardized series of this type of turbine.

The design methodology and the designs of the standardized turbine series included in this document are meant for free use by the countries and their institutions; and when the information contained herein is used in any other document, it is only required that express reference be made to the OLADE manual. Likewise, when the designs included herein are used in manufacturing turbines, the words "OLADE Technology" must be visibly displayed.

INTRODUCTION

The Michell–Banki turbine is a cross-flow, partial-intake turbine which is generally used in those small hydro power projects that tap a medium head and flow to satisfy the demand of an electric power sustem whose daily load diagram has a load factor of less than 0.5. Its range of application falls within the range of application of the Francis turbine, whose efficiency it surpasses when the turbine operates most of the time with a part load, as occurs in small hydro projects, when the turbine absorbs the variations in daily load demand. Another advantage of the Michell–Banki turbine over the Francis turbine is its low manufacturing cost.

The origin of the Michell–Banki turbine dates back to the beginning of this century, when in 1903 engineer A.G. Michell developed the cross-flow turbine with a double impulse, which was studied at the University of Budapest by D. Banki between 1917 and 1919. In 1933 German engineer Fritz Ossberger developed the turbine known as the cross-flow turbine, which was better designed that the one developed by Michell and studied by Banki. Later on, in 1938 this turbine's injector design introduced a rotating blade as the guiding device. In 1948, for the first time ever, this turbine was designed with the injector positioned horizontally. In the following years and to date, different studies have been done on this turbine, which has come to be known by several different names: the Michell-Banki turbine; the Michell turbine; the Banki turbine; the cross-flow turbine; and in the case of German manufacturers Ossberger, the Michell–Ossberger turbine. All of these use the same operational principles as those developed by Michell and studied by Banki, but they differ in the details of their designs.

In order to provide a reference document for calculating and designing Michell–Banki turbines, the present manual has been elaborated and geared to engineers and technicians interested in developing and adapting hydraulic turbine techology requiring the elaboration of designs up to the level of detail.

The manual is developed in five chapters and three appendices; it contains the overall design of a cross-flow turbine, which we refer to herein as the Michell–Banki. This turbine is characterized by an injector with a guide vane which regulates a stream of water towards the runner blades.

The first chapter of the manual describes the turbine; indicates its range of application as compared with those of other turbines; and shows its advantages with respect to the Francis turbine and the way to obtain its design parameters, i.e., maximum flow and number of revolutions.

The second chapter deals with aspects related to hydraulic calculations and design; and it analyzes speed diagrams in a dimensionless form, to allow flexibility for the engineer or technician, so that some design parameters can be varied during the process of private research. It also shows how to determine the shape of the injector, the runner and the casing and indicates the formulas for calculating dimensions with a general equation and a practical expression.

The third chapter describes the detailed design of the Michell–Banki turbine developed in the present manual and shows the mechanical calculations that must be done for each component of the machine. Reference is also made to the drawings in Appendix 2, which have been elaborated in order to describe each piece of the turbine.

The fourth chapter offers some criteria for determining standardized series of Michell–Banki turbines, and presents a particular case of standardization. It also makes some recommendations as to turbine selection, determination of number of units, technical specifications to be requested from manufacturers, and turbine relocation.

The fifth chapter notes alternatives and recommendations for the production of each part of the turbine and indicates the features of the materials that are recommended for use in each of them.

Among the appendices, the one providing a practical example of the calculation and design of a Michell–Banki turbine should be highlighted. For this appendix, data were taken from a 400-kW small hydro power station project.

It should be noted that other experience exists in the world in the design of Michell–Banki turbines, some of this in activities tied to research, development and technology adaptation, as in the case of the University of Santander in Colombia, the Federico Santa María Technical University in Chile, INECEL and the National Polytechnic School in Ecuador, the Nicaraguan Institute of Energy in Nicaragua, the National University of Engineering and ITINTEC in Peru, the National Energy Policy Commission in the Dominican Republic, SKAT in Switzerland, Oregon State College in the United States, Simon Bolivar University in Venezuela, and others. Among the factories that produce these turbines figure Ossberger–Turbinenfabrik in Germany, IMEG Técnica in Argentina, Nikki Corporation in Japan and Balaju–Yantra Ltd. in Nepal.

The present manual was prepared as part of the activities of the Regional Program of Small Hydro Power Stations of the Latin American Energy Organization (OLADE), under the coordination of engineer Enrique Indacochea R. de S. It was elaborated by engineer Carlos Alberto Hernández Bazó, resident expert of the aforementioned program.

1. GENERAL DESCRIPTION AND DESIGN PARAMETERS

The Michell–Banki turbine is a cross–flow, partial–intake impulse turbine with a double effect impulse. Its main elements are an injector or nozzle, which regulates and directs the water flow that enters the turbine, and a runner, which generates power to the turbine shaft when it receives a double impulse from the flow of water that circulates through it.

The Michell–Banki turbine presents some of the best prospects for use in small hydro power stations, due to its simplicity of design and fabrication; its high efficiency when operating with partial loads; and its reduced manufacturing and maintenance costs.

Its range of application is defined by the specific speeds N_q and N_s, obtained with the following formulas:

$$N_q = N \frac{Q^{1/2}}{H^{3/4}} \quad \text{and} \quad N_s = N \frac{P^{1/2}}{H^{5/4}}$$

where:
- P is the turbine brake power, in C.V.
- Q is the maximum flow rate through the turbine, in m^3/s
- H is the station's net head, in meters
- N is the turbine's speed of rotation, in RPM

The following chart shows the Michell–Banki turbine's range of application, as compared with those of other types of turbines. It should be pointed out that the range indicated for the Michell–Banki turbine has been defined on the basis of its mechanical design limitations for the higher level of efficiency and for the lower level of application. These ranges can vary according to particular experiences.

TABLE No. 1
RANGES OF APPLICATION FOR HYDRAULIC TURBINES

TYPE OF TURBINE	N_q	N_s
One–nozzle Pelton turbine	Up to 9	Up to 30
Two–nozzle Pelton turbine	4–13	14–42
Pelton turbine with 3 nozzles or more	5–22	17–73
Michell–Banki turbine	18–60	60–200
Slow Francis turbine	18–38	60–125
Regular Francis turbine	38–68	125–225
Fast Francis turbine	68–135	225–450
Axial–flow turbines	105–300	350–1000

The experience obtained with the Michell–Banki turbine shows that it can operate with maximum heads between 100 and 200 meters and maximum efficiencies between 80 and 85 o/o, and that it can generate a maximum power output between 750 and 1000 kW.

From the chart above, it can be seen that the Michell–Banki turbine's range of application overlaps the ranges of slow and regular Francis Turbines. This can also be observed in Figure 1, which illustrates the selection of different types of turbines.

The Michell–Banki turbine can be used in any hydro power station project where the daily load diagram has a load factor below 0.5 and where the design parameters, power output and head, fall within this turbine's area of application.

This can be observed in Figure 2, which graphs variations in the efficiency of Michell–Banki and Francis turbines, as a function of the percentage variations of part load, under which they operate during the day, in order to satisfy the demands of an electric power system with a 0.4 load factor (see Figure 3). It can be observed that, despite the fact that the Francis turbine has a higher maximum efficiency than the Michell–Banki, the latter has a greater average daily efficiency and betters the Francis turbine's efficiency when it operates with part loads below 42 o/o capacity (see Figure 4).

Figure 3 also indicates that, generally, when a power station is installed, the maximum daily power output is a percentage of the maximum daily power capacity projected for the station. This implies that the turbine operates 80 or 90 o/o of the time with partial loads of less than 42 o/o capacity.

To design a Michell–Banki turbine, it is necessary to determine the data for the utilizable net head and its maximum water flow rate. In some projects, this flow rate corresponds to the annual minimum that is available, according to the hydrological study; and in other projects, it can be deduced from the brake power that the turbine should deliver to the generator, so that it can, in turn, deliver a given amount of power to the electric power system.

The turbine brake power can be obtained with the following formula:

$$P_T = \frac{P_g}{\eta_g \eta_{tr}}$$

where:

- P_g is the maximum power output delivered by the generator to the electric power system
- η_g is the generator efficiency
- η_{tr} is the mechanical transmission efficiency between the turbine and the generator

With the turbine brake power, the design flow rate can be obtained as follows:

$$Q = \frac{P_T}{9.807\, H \eta_T}$$

where:

- Q is the maximum design flow rate through the turbine, in m^3/s
- P_T is the turbine brake power, in kW
- H is the utilizable net head, in meters
- η_T is the turbine efficiency when operating at full load

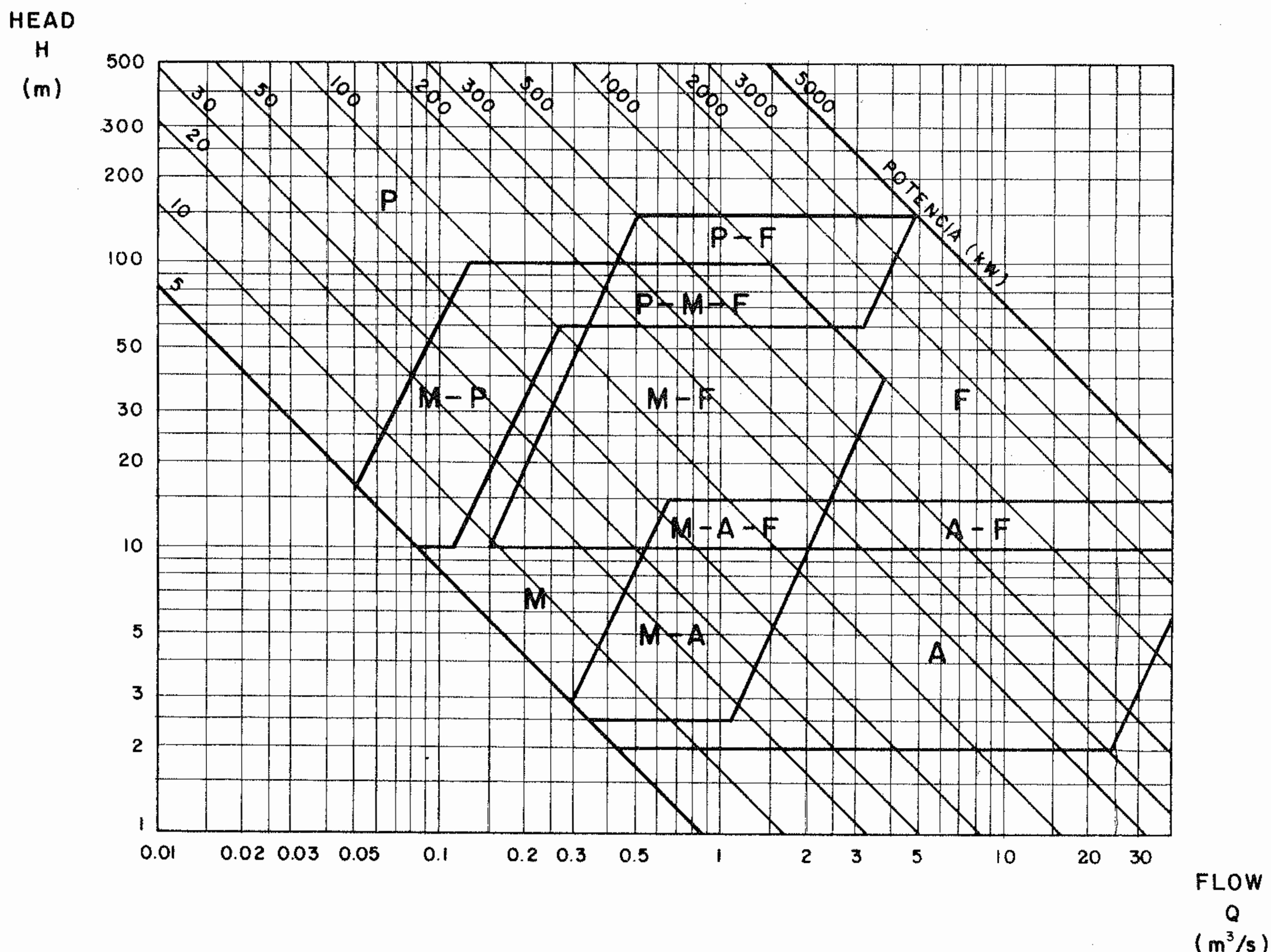
Another necessary parameter for the turbine design is given by the optimum number of revolutions with which the turbine should operate, as determined by the following formula:

$$N = \frac{39.85\, H^{1/2}}{De}$$

where:

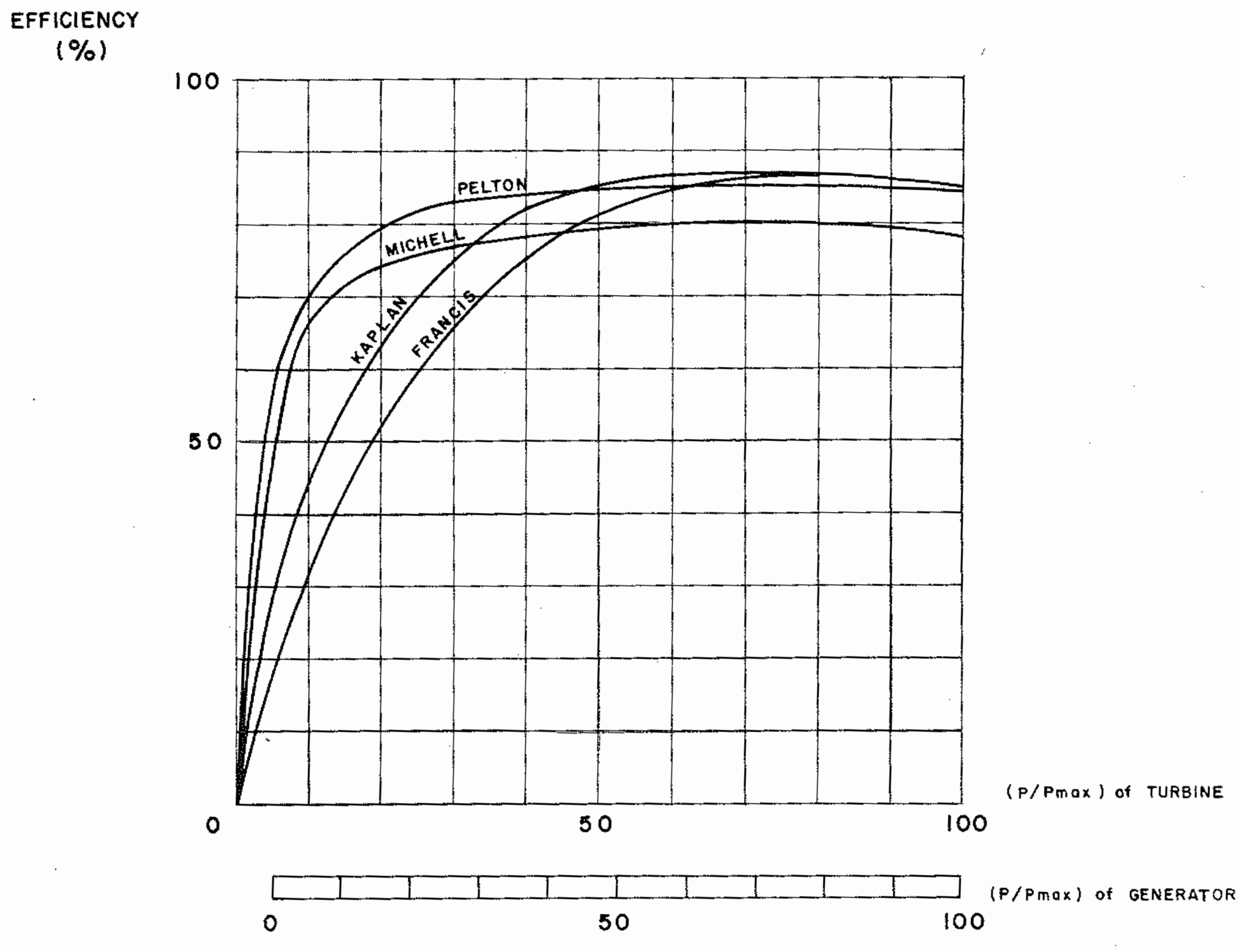
- N is the optimal number of revolutions, in RPM
- De is the outside (external) runner diameter, in meters
- H is the utilizable net head, in meters

When a turbine is designed to rotate at a synchronous speed, the outside runner diameter is determined by rearranging the foregoing formula. When the transmission between the turbine and the generator is based on a system of belts or gears, the runner diameter is assumed and the above formula is applied.



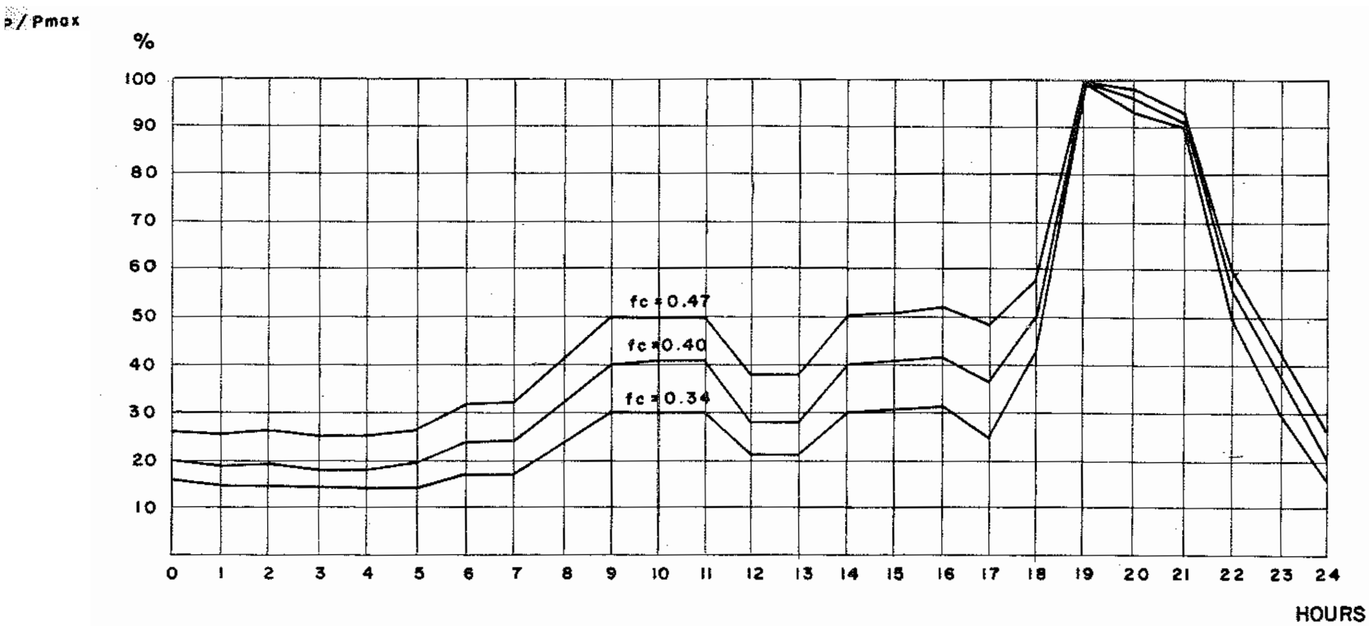
SELECTION OF THE TYPE OF TURBINE

FIGURE 1

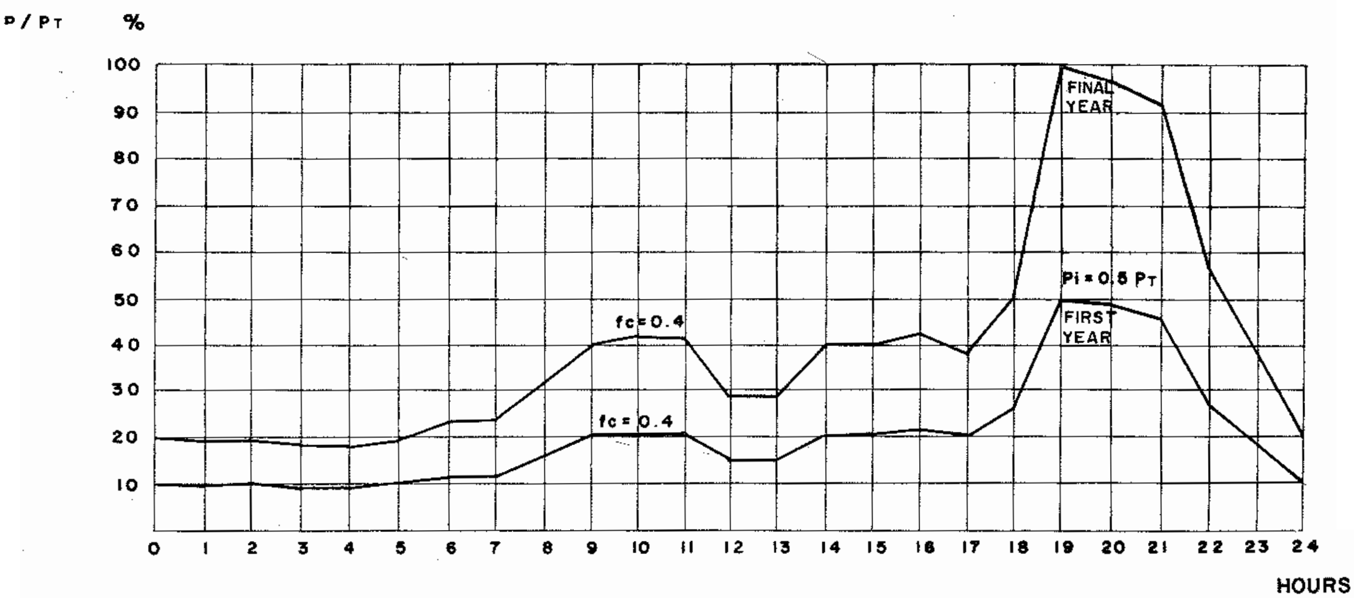


PART-LOAD EFFICIENCY OF HYDRAULIC TURBINES

FIGURE 2

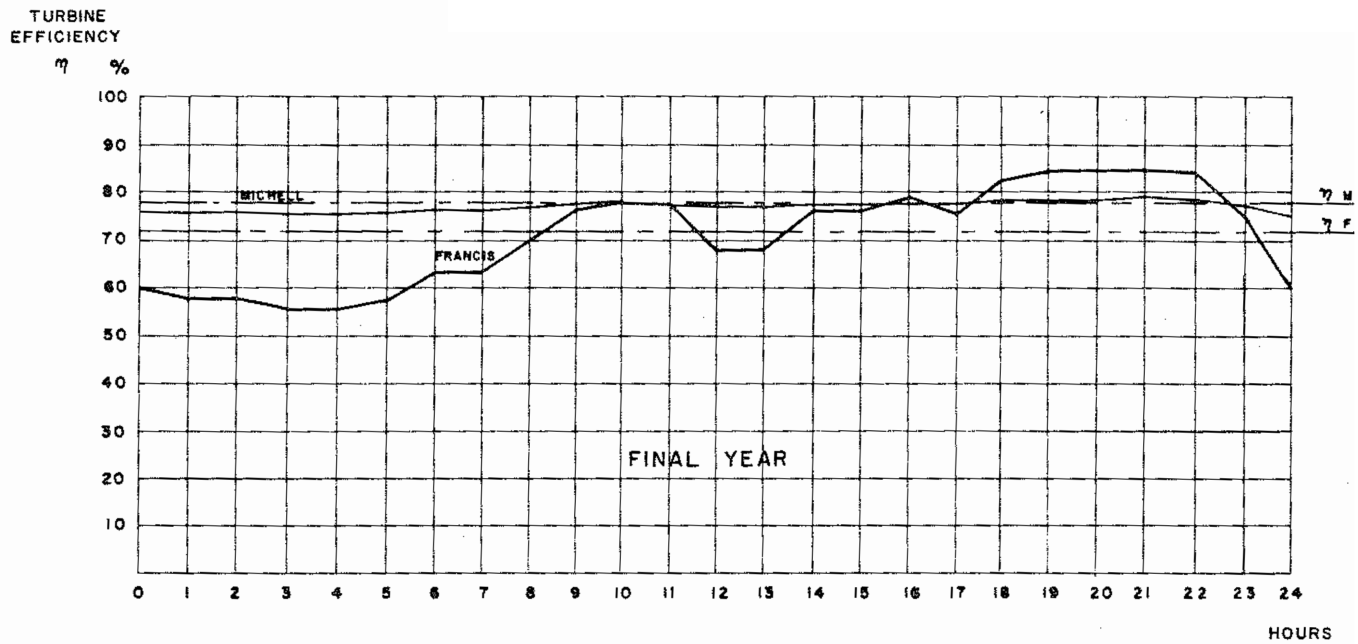


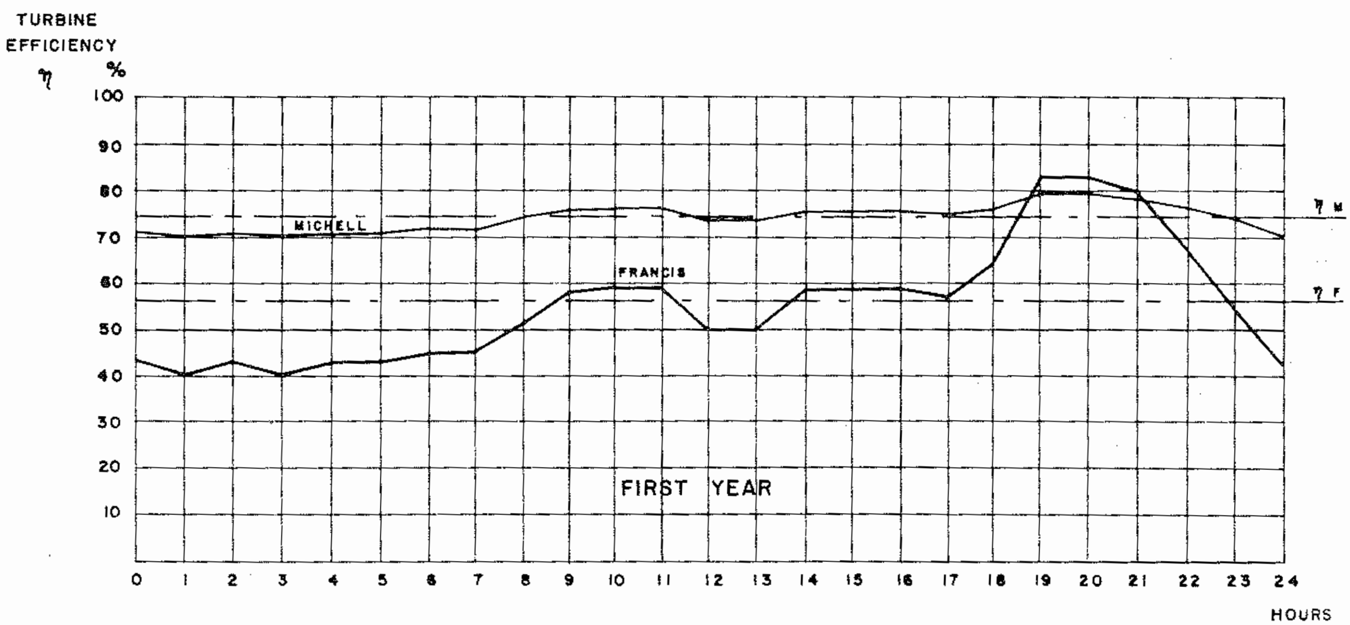
UNIT LOAD DIAGRAMS



TURBINE LOAD DIAGRAM FOR
AN ELECTRICAL SYSTEM WITH
fc = 0.4

FIGURE 3





AVERAGE DAILY EFFICIENCY ACCORDING TO

LOAD VARIATIONS IN AN ELECTRICAL SYSTEM WITH fc= 0.4

FIGURE 4

2. HYDRAULIC CALCULATIONS AND DESIGN

The hydraulic turbine calculations are used to determine the dimensions of its main elements, on the basis of the design that characterizes each type of turbine.

The Michell–Banki turbine design is based on the fact that the injector accelerates and regulates the water flow that enters the turbine, and orientates the jet of rectangular cross section towards the runner blades, giving a first impulse to the blades; then the water runs across the inside of the runner and gives a second impulse to the blades before the water flows towards the turbine discharge.

2.1 Speed Diagrams

A turbine's runner profiles are determined on the basis of the speed diagrams for each point of the runner. To determine these diagrams, it is necessary to define the speed at which water leaves the injector; this is determined on the basis of the Bernouilli Equation, applied between the reservoir surface, where the waterspeed is approximately zero, and the injector exit:

$$\frac{P_o}{\gamma} + \frac{C_o^2}{2g} + Z_o = \frac{P_i}{\gamma} + \frac{C_i^2}{2g} + Z_i + \Delta H_t + \Delta H_i$$

where:

- C_o and C_i represent the velocity of a water particle at the reservoir surface and at the injector exit, respectively.
- P_o and P_i represent the pressures at the reservoir surface and the injector exit, respectively. In this case, both pressures can be considered equal to atmospheric pressure when the turbine discharges without a suction pipe.
- Z_o and Z_i represent the topographical levels at the reservoir surface and at the injector position, respectively; and the difference between them is equal to the gross head.
- γ and g represent the specific weight of water and the acceleration of gravity, respectively
- ΔH_t is the pressure loss due to water friction with the penstock walls.
- ΔH_i is the pressure loss due to water friction with the injector walls.

On the basis of all these considerations, it can be determined that the speed at which water leaves the injector is as follows:

$$C_i = \sqrt{1 - \frac{\Delta H_i}{H}} \sqrt{2gH}$$

where: H is the station's effective or net head, obtained from the difference between the gross head and the penstock pressure losses. Kc is defined as the injector's speed coefficient, represented by:

$$K_c = \sqrt{1 - \frac{\Delta H_i}{H}}$$

and the speed at which water leaves the injector is expressed as follows:

$$C_i = K_c \sqrt{2gH}$$

In practical terms, Kc has values of between 0.97 and 0.98.

The exit speed of water leaving the nozzle is equal to the speed at which water enters the runner. This water jet is, in turn, directed towards the runner with an average angle termed α_2, which has practical values of around 16 degrees.

It is also known that for impulse turbines the speed is expressed as follows:

$$U_2 = Ku\, C_2 \cos \alpha_2$$

where: Ku is the tangential speed coefficient, which in the case of impulse turbines has a value of approximately 0.5.

With these velocities, the relative velocity can be determined:

$$W_2 = C_2 \sqrt{1 - Ku\,(2 - Ku) \cos^2 \alpha_2}$$

and with angle β_2 the construction of the speed diagram for the runner entrance is concluded (see Figure 5).

Inside the runner, the speed triangles are equal because:

$$U_1 = U'_1$$
$$C_1 = C'_1$$
$$\alpha_1 = \alpha'_1$$
$$\beta'_1 = 180 - \beta_1$$

Therefore, it can be concluded that:

$$\beta'_1 = \beta_1 = 90^o$$

The speed diagram for the runner exit indicates that:

$$U'_2 = U_2 = K_u\, C_2 \, \mathrm{Cos}\, \alpha_2$$

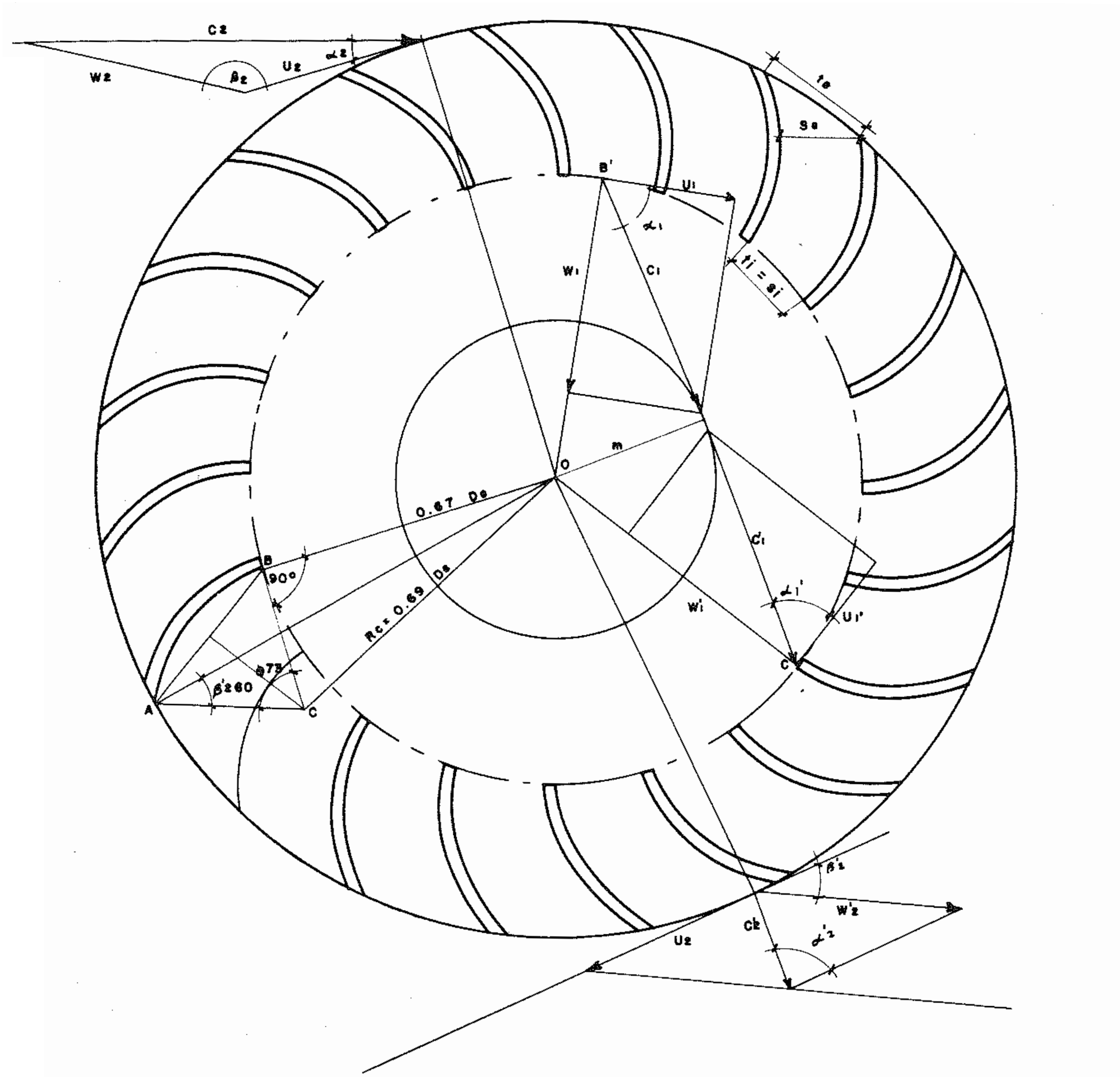
$$\beta_2 = 180 - \beta'_2$$

$$\beta'_2 = \mathrm{arc\ Sin} \left| \frac{\mathrm{Sin}\ \alpha_2}{1 - K_u\,(2 - K_u)\ \mathrm{Cos}^2\ \alpha_2 \quad ^{1/2}} \right|$$

The relative velocity would be expressed as:

$$W'_2 = K_f\, W_2$$

where: K_f is the relative velocity coefficient, which expresses the losses due to friction of the water with the runner blades, with an approximate value of 0.98.



Speed Diagram for the Michell-Banki Runner

FIGURE 5

With these velocities, the absolute waterspeed at the runner exit can be obtained:

$$C'_2 = C_2 \sqrt{K_f^2 \,(1 - K_u\,(2\text{-}K_u) \,\text{Cos}^2\ a_2\ ) + K_u^2\ \text{Cos}^2\ a_2 - 2\ K_f\,\text{Cos}^2\ a_2\ (1\text{-}K_u)\ K_u}$$

The exit angle, with respect to the runner tangent, can be determined as follows:

$$a'_2 = \text{arc Sin} \left| \frac{K_f\ \text{Sin}\ \beta'_2 \sqrt{1 - K_u\,(2\text{-}K_u)\,\text{Cos}^2\ a_2}}{\sqrt{K_f^2 + K_u\,\text{Cos}^2\ a_2\ (K_u - K_f^2\ (2 - K_u) - K_f}} \right|$$

All of these velocities can be expressed practically when the constants are defined. For example, if an average angle is assumed as equal to 16^{o}C, with a tangential speed coefficient K_u of 0.5, and a relative velocity coefficient K_f of 0.98, the following practical equations are obtained:

$$\begin{aligned}
C_2 &= 4.34 \sqrt{H} \\
U_2 &= 2.09 \sqrt{H} \\
W_2 &= 2.40 \sqrt{H} \\
W'_2 &= 2.35 \sqrt{H} \\
C'_2 &= 1.20 \sqrt{H} \\
\beta'_2 &= 29.83^{o} \simeq 30^{o} \\
\beta_1 &= 90^{o}
\end{aligned}$$

As can be observed, the speed diagrams only depend on the head, and the angles are independent of the head flow conditions.

To determine the turbine's hydraulic efficiency, the general turbine equation is applied:

$$\eta_h\ gH = U_2\ C_2\ \text{Cos}\ a_2\ - U'_2\ C'_2\ \text{Cos}\ a'_2$$

and this yields:

$$\eta_h \doteq 2\ K_c^2\ \text{Cos}^2\ a_2\ K_u\ (1\text{-}K_u)\ (1 + K_f)$$

In addition to the hydraulic efficiency, to determine the total turbine efficiency the volumetric losses, shock losses and mechanical losses must also be taken into account. In the case of the Michell—Banki turbine, efficiency can reach 82 o/o with good manufacturing work manship.

2.2 Injector Geometry

Different geometries are available for the injectors used in Michell—Banki turbines; some of these are illustrated in Figure 6. It can be seen that some of them have a gate blade, others a regulating vane with various shapes; and in some cases the injector is designed without a regulating blade. Injectors can also be designed with one or two compartments in order to improve the turbine's efficiency when operating at partial loads. In Figures 7—a and 7—b, the variation in turbine efficiency as a function of the number of compartments and their proportional dimensions can be appreciated.

In defining the geometry, it is necessary for the design to consider good conduction and acceleration of the water flow as well as an adequate orientation and regulation of this flow towards the runner blades.

When defining the injector geometry with a regulating vane, it is necessary to define the profile taking into account a balance between the pressure losses in the water flows derived from the vane's dividing action, so as to have equal losses and to obtain the same exit speed in both sections.

This manual shows the geometry of an injector with a regulating vane, which has been calculated on the basis of the methodology described above and considering a vane whose aerodynamic profile will reduce the actuating torque required to regulate the water flow. With the aid of a computer, it has been demonstrated that the shape shown in Figure 8a is a function of the runner diameter (see Figure 8b) and does not vary when it is designed for different heads and flows. The only dimension which varies as a function of head and flow, with which the turbine is designed, is the injector width, which it is calculated as follows:

$$B = \frac{Q}{P\ (\pi\, De - eZ)\ Ko\ Kc \sqrt{2\,gH}\ Sin\ a_2}$$

where:

The inner width of injector B is expressed in meters

- Q is the maximum flow to pass through the turbine, in m^3/s
- p is a factor of the admission arch, which, for the injector dealt with in this manual, has a value of 1.
- De is the outside runner diameter, expressed in meters
- e is the thickness of the runner blades, expressed in meters
- Z is the number of runner blades
- Ko is the percentage of the outside circumference of the runner through which the water enters (intake arch)

A practical formula for determining the width of the injector with a geometry as shown in Figure 8 is expressed by:

$$B = \frac{0.96\ Q}{De\ \sqrt{H}}$$

For injectors with different geometry, a practical formula can be determined on the basis of the following expression:

$$B = \frac{constant\ \ Q}{De \sqrt{H}}$$

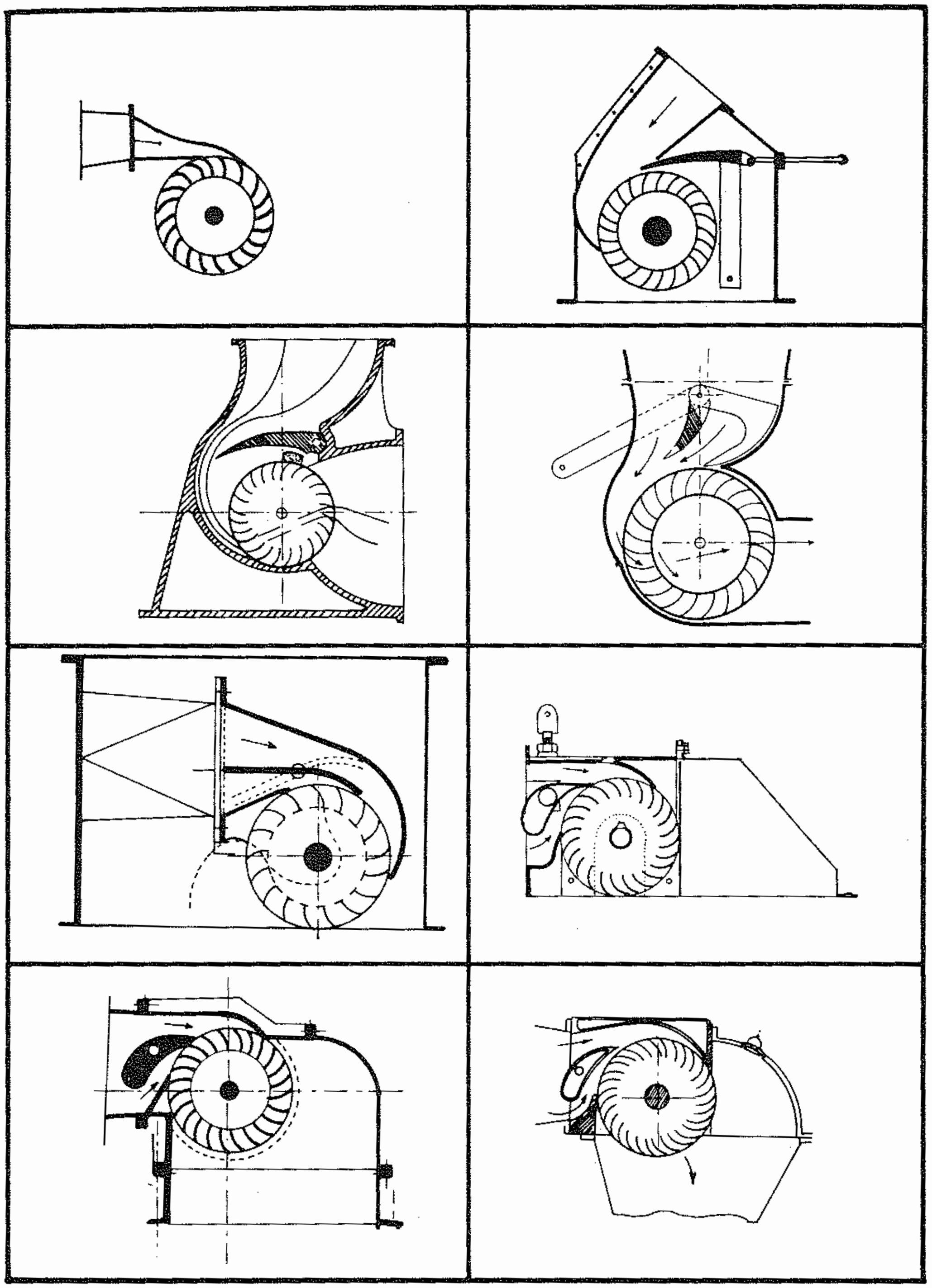
In this case, using the general expression for calculating the injector width, the constant value must be found for each injector geometry.

The injector's dimensions for different runner diameters are shown in Table No.2. In the case that runner diameters different from those indicated are used, the dimensions can be determined by means of the linear interpolation method.

2.3 Runner Geometry

The runner geometry is determined on the basis of the angles obtained from the diagrams derived under item 2.1. Thus, the inner diameter (Di) in relation to the outer diameter (De) is expressed as follows:

$$\frac{Di}{De} = \sqrt{\frac{(2\,K_u\,Cos^2\,a_2 - 1) + \sqrt{1 - 4\,Cos^2\,a_2\,(1-K_u)\,K_u}}{2\,K_u^2\ Cos^2\ a_2}}$$



PROFILES OF INJECTORS FOR MICHELL-BANKI TURBINES

FIGURE 6

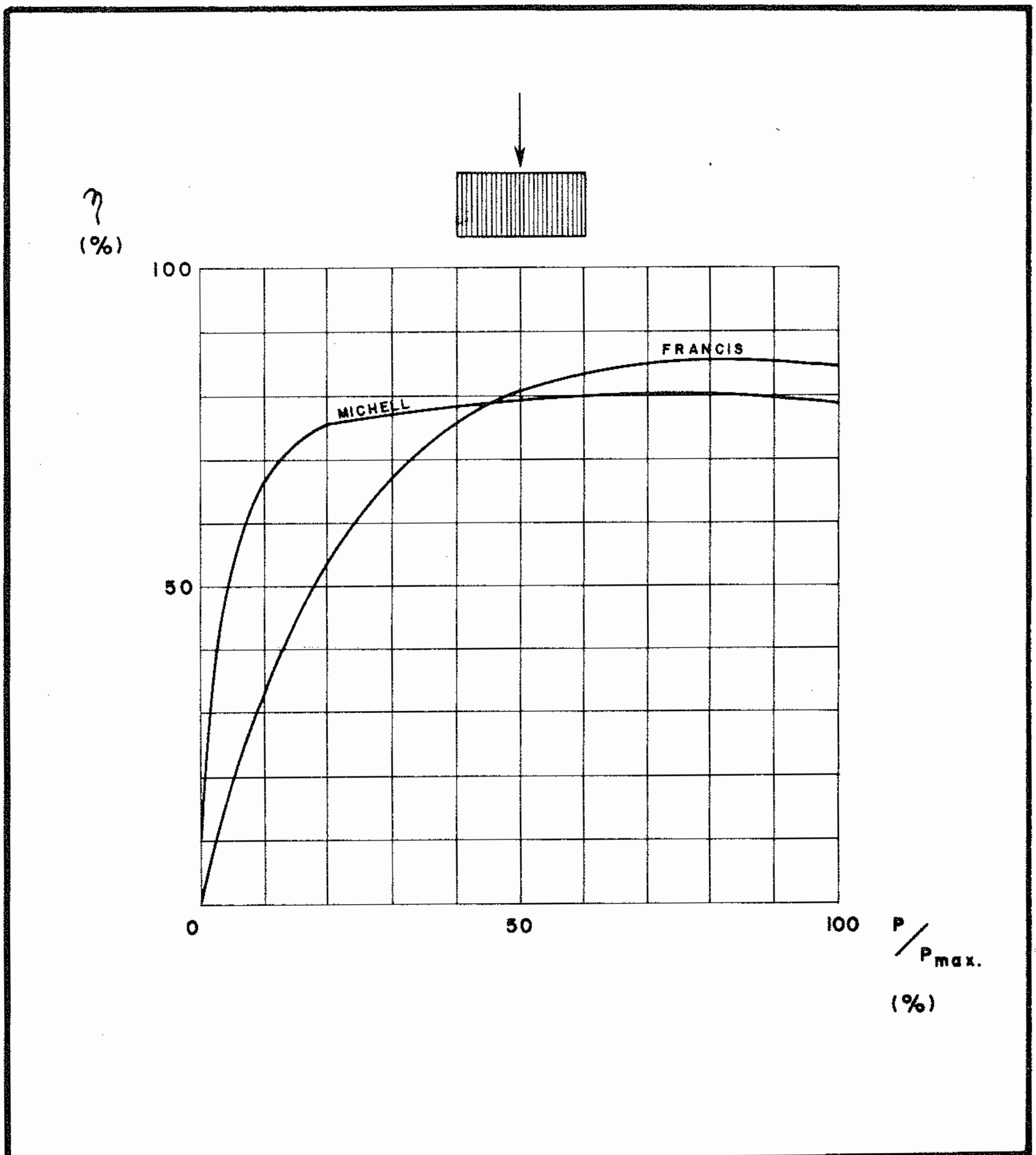


FIGURE 7A

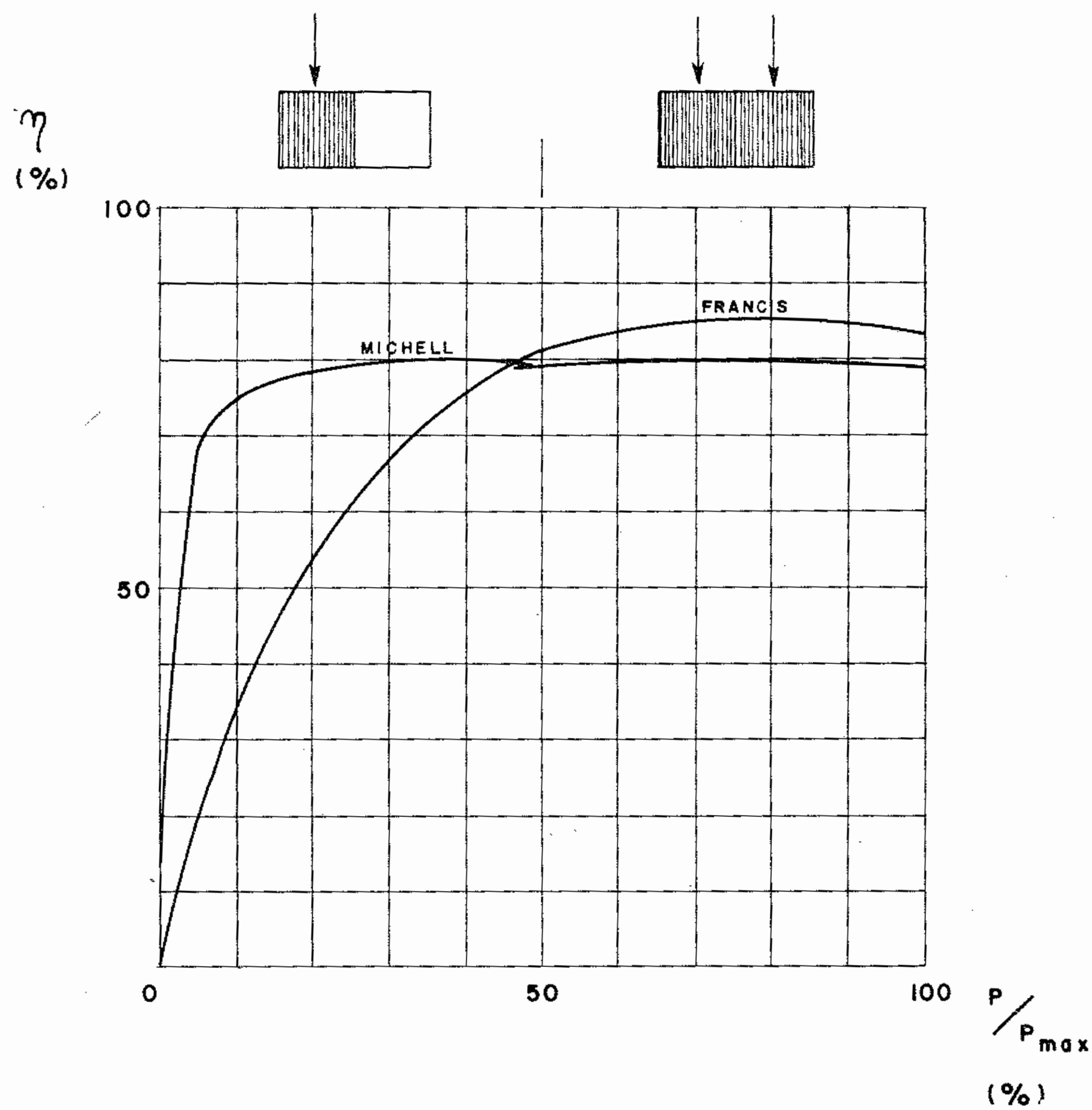


FIGURE 7B

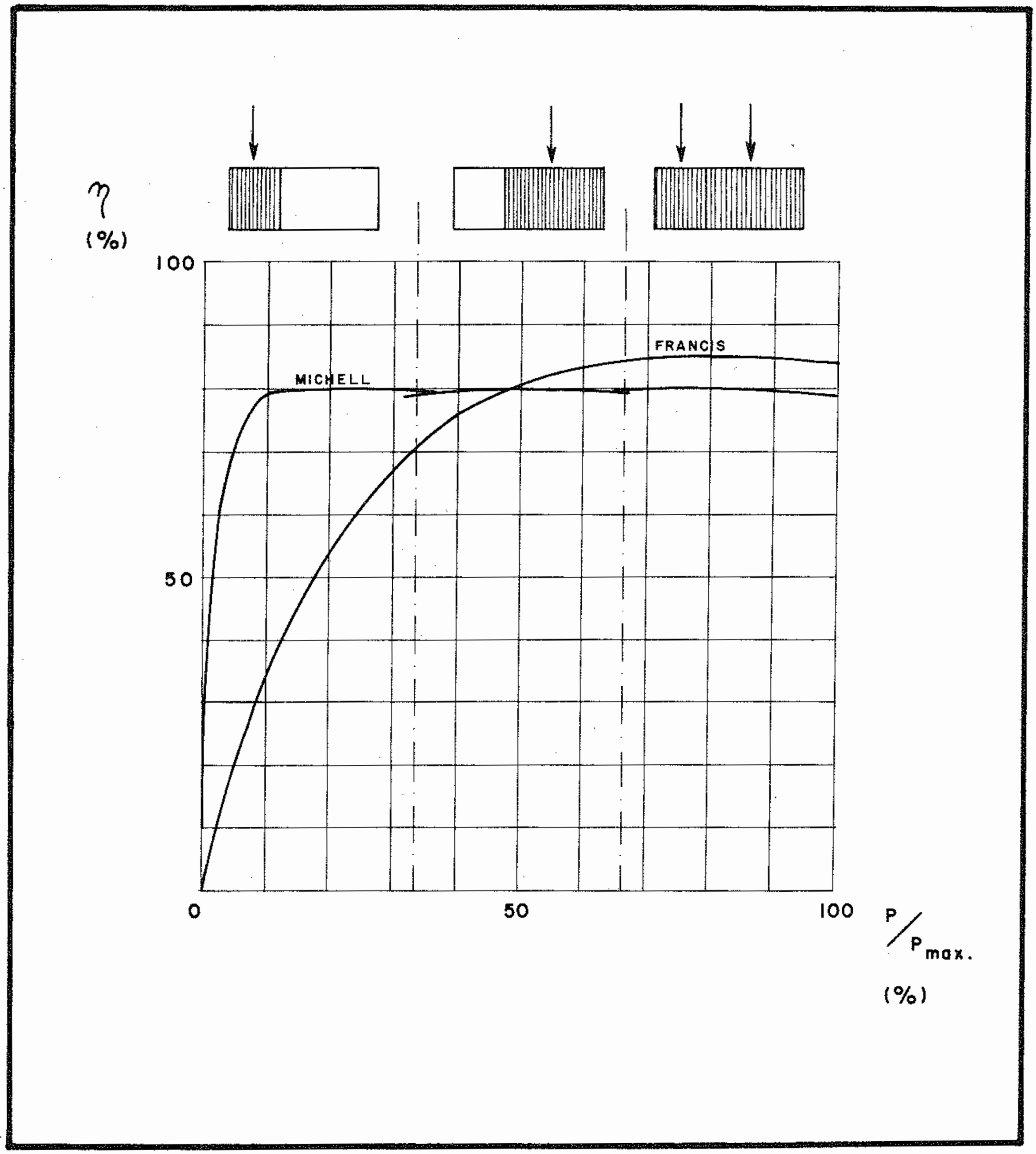


FIGURE 7C

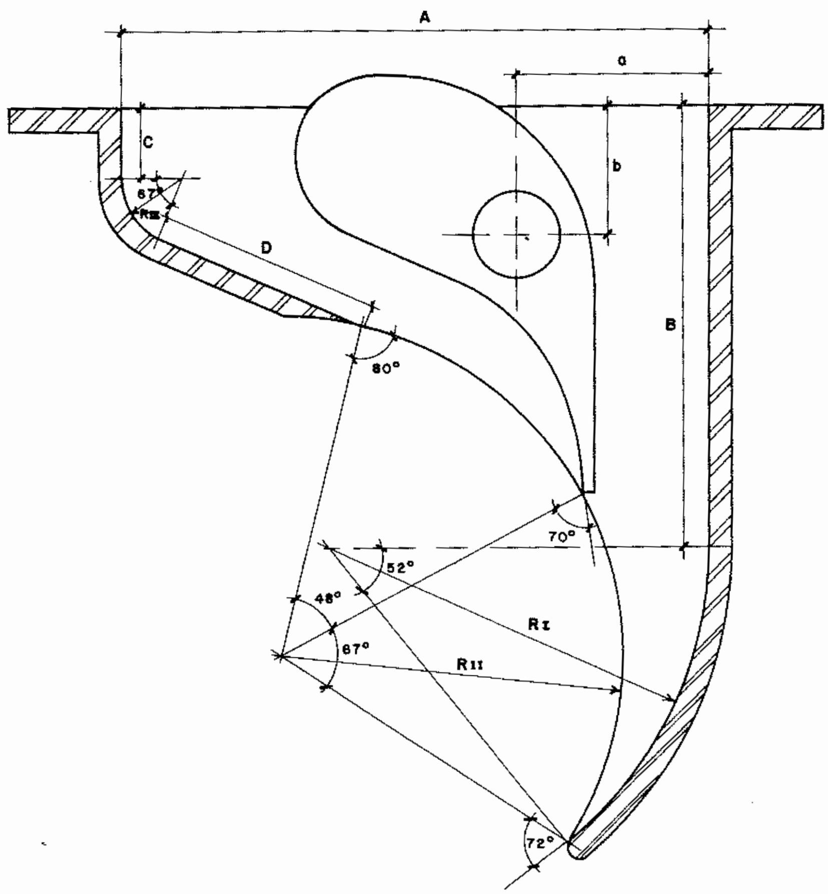
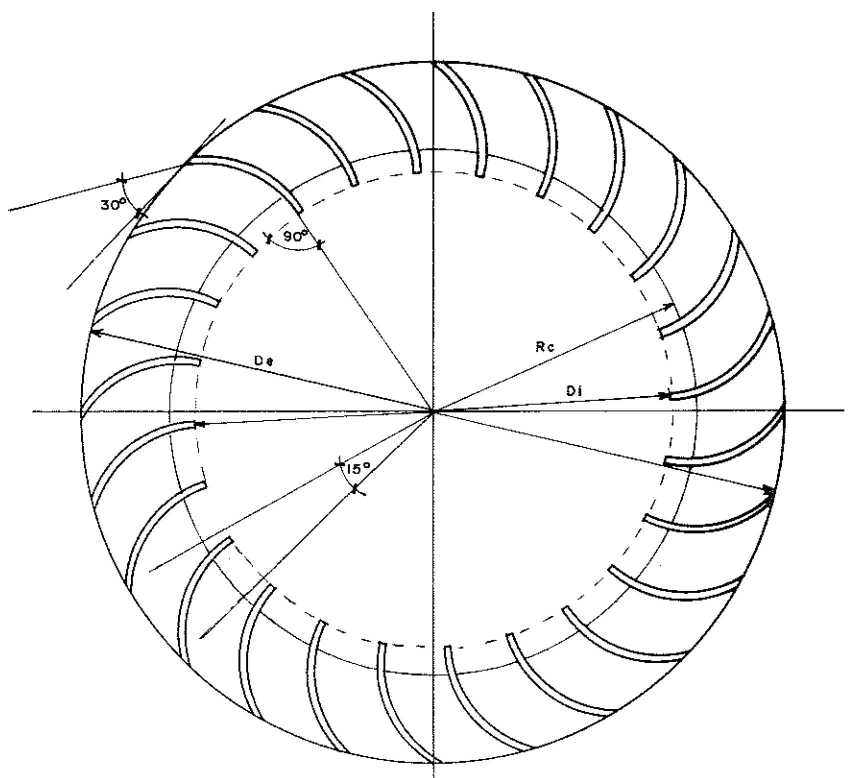


FIGURE 8A



RUNNER PROFILE

FIGURE 8B

RUNNER PROFILE DIMENSIONS

DIMENSION (mm)	RUNNER DIAMETER - De (mm)			
	300	400	500	600
Di	100	133	167	200
Rc	110	147	183	220
r	49	65	82	98

INJECTOR PROFILE DIMENSIONS

DIMENSION (mm)	RUNNER DIAMETER - De (mm)			
	300	400	500	600
A	261	348	435	522
B	195	260	325	390
C	31	41	52	62
D	102	136	170	204
a	85	113	142	170
b	55	73	92	110
R_I	168	224	280	336
R_{II}	151	201	252	302
R_{III}	28	37	47	56
R_1	94	125	157	188
R_2	39	52	65	78
R_3	31	41	52	62
R_4	60	80	100	120
R_5	100	133	167	200
R_6	70	93	117	140
R_7	133	177	222	266
x	35	47	58	70
y	116	155	193	232
ρ_1	51	68	85	102
ρ_2	98	131	163	196
ρ_3	20	27	33	40

TABLE 2

RUNNER PROFILE DIMENSIONS

DIMENSION (mm)	RUNNER DIAMETER - De (mm)			
	300	400	500	600
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R_I	168	224	280	336
R_{II}	151	201	252	302
R_{III}	28	37	47	56
R_1	94	125	157	188
R_2	39	52	65	78
R_3	31	41	52	62
R_4	60	80	100	120
R_5	100	.133	167	200
R_6	70	93	117	140
R_7	133	177	222	266
x	35	47	58	70
y	116	155	193	232
ρ_1	51	68	85	102
ρ_2	98	131	163	196
ρ_3	20	27	33	40

TABLE 2

Considering α_2 equal to 16^0 and K_u equal to 0.5, the inner diameter can be obtained with the following expression:

$$Di = 0.66 \ De$$

The ratio of the runner blades' curve is also expressed as a function of the runner diameter and the blade angle:

$$r = \frac{De}{4 \cos \beta'_2} \left| 1 - \left| \frac{Di}{De} \right|^2 \right|$$

Substituting the known values, this can be expressed in practical terms as:

$$r = 0.163 \ De$$

The angle of the runner blade's curve is obtained with the following formula:

$$\phi = 2 \text{ arc tg} \left| \frac{\cos \beta'_2}{\left| \frac{Di}{De} \right| + \sin \beta'_2} \right|$$

When the angle α_2 is equal to 16^0, this angle has a value of 73^0.

With these dimensions, the runner profile can be defined for a given outer diameter, either assumed or calculated with the optimal speed of rotation formula shown in Chapter 1. As has been observed, just as with the injector, when the runner diameter is known, all of its dimensions can automatically be found.

The inside width of the runner is obtained after finishing the detailed injector design; the space required for welding the blades to the disk must be taken into account. One practical way of estimating the width of the runner is to consider it 50 o/o greater than the injector width.

In designing Michell–Banki turbines, one must bear in mind that from the theoretical point of view, there exists a limiting factor insofar as the intake arch, expressed as follows:

$$\sphericalangle \ BOC = 2 \text{ arc tg} \left| \frac{\sqrt{1 - K_u (2-K_u) \cos^2 \alpha_2}}{\left| \frac{Di}{De} \right| K_u \cos \alpha_2} \right|$$

The maximum percentage of intake arch can be obtained with the following expression:

$$K_o = \frac{\sphericalangle \ BOC}{360^0}$$

For the values α_2 equal to 16^0 and K_u equal to 0.5, a K_o value of 0.334 is obtained.

For the injector shown in this chapter, the intake percentage is 0.30.

If one considers that the trajectory of a water particle inside the runner follows a straight line, then the maximum diameter of the shaft that crosses the runner must be:

$$d = Di \cos \left| \frac{\measuredangle BOC}{2} \right|$$

or, in practical terms:

$$d = 0.328 \, De$$

2.4 Casing Geometry

The geometry of the casing or cover for a Michell–Banki turbine is designed considering the arch of the water flow leaving the runner and its trajectory.

The total runner arch is obtained with the following formula (see Figure 8):

$$\theta_t = \theta_0 + \theta_1 + \theta_2 + \measuredangle BOC$$

where:

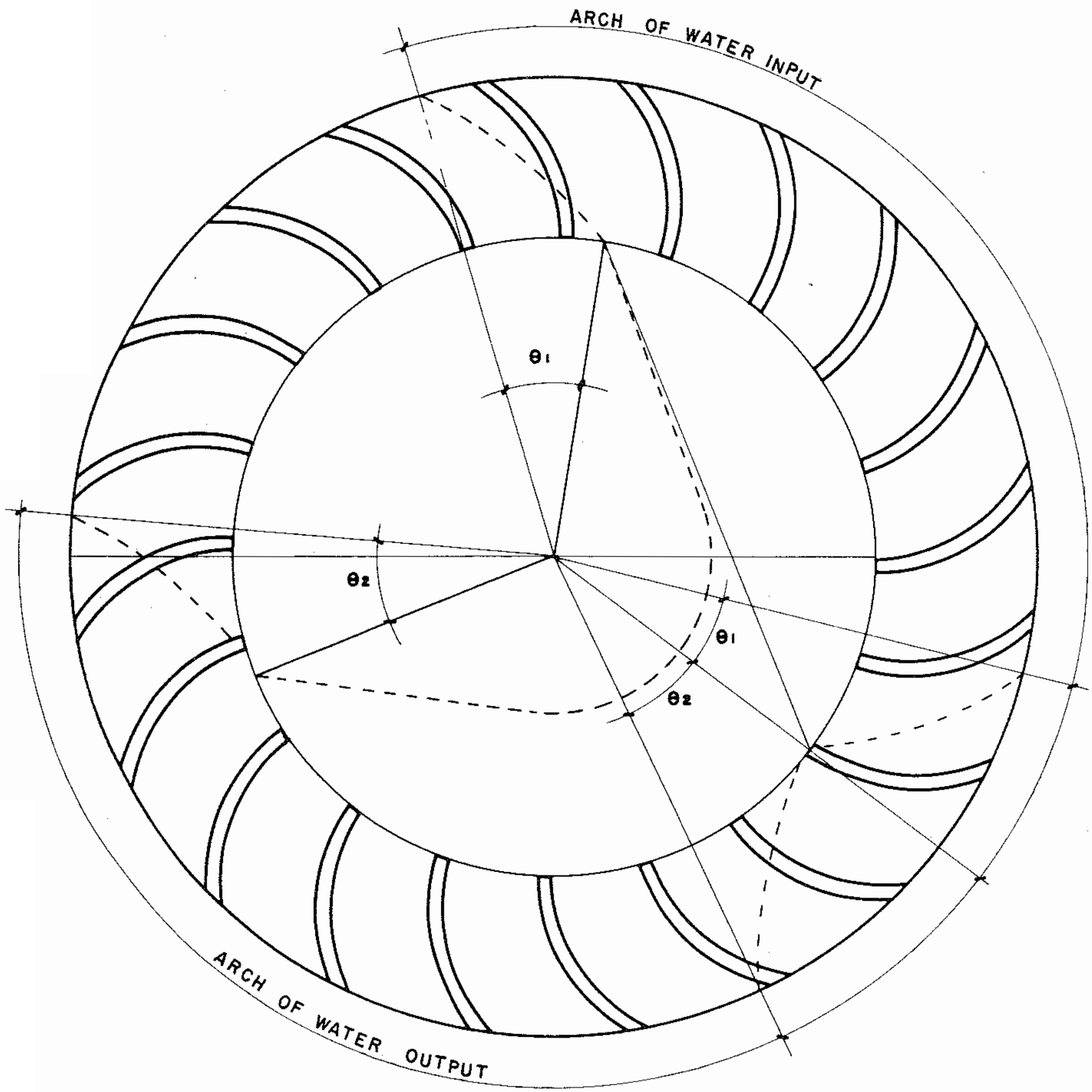
$$\theta_0 = Ko \times 360^o$$

$$\theta_1 = \frac{r\,\phi}{W_2} \; \frac{360}{60} \; N \qquad \phi \text{ in radians}$$

$$\theta_2 = \frac{\theta_1}{K_f}$$

or, in practical terms:

$$\theta_t = \theta_0 + 163^o$$



Working Arch with a Michell-Banki Runner having an Intake Arch of 1/3

FIGURE 9

3. DETAILED DESIGN AND MECHANICAL CALCULATIONS

A turbine is designed to define the dimensions of each one of its parts, which, when assembled can form the turbine as a machine. The design must take into account suitable assembly systems, as well as the aspects of hermeticity and lubrication.

The mechanical calculations are done in order to verify if the material used for each part is capable of bearing the stresses to which it will be subjected. These calculations are a complement to the detailed design, and with them, additional dimensions are obtained, over and above the ones defined by the design itself.

The detailed design of a Michell—Banki turbine is outlined in the drawings included in the appendix to this manual. The criteria to be considered and the main mechanical calculations that are recommended are discussed below.

In Drawing TM-01-01 the general turbine assembly can be appreciated. The injector is indicated with a broken line. Its structure permits the bearings support to be located in the side walls, and thus the injector and the runner are mounted in one single assembly so as to guarantee good alignment between the two. It can also be observed that the regulating mechanism allows the opening and closing of each injector compartment to be programmed by means of two cams, thereby allowing the turbine to maintain good efficiency when operating with a low load percentage. The casing is designed so that the water flow at the runner exit will be oriented towards the discharge.

Drawing TM-01-02 shows a cross-section of the injector, the runner and the casing, wherein the geometry of the injector and the runner can be observed. The regulating vane of the injector is noteworthy, for it will be in charge of regulating and directing the water flow towards the runner blades. It can also be seen that at the entry and exit of the turbine there are flanges with threaded holes, where the inlet pipe and the casing are installed. The latter has elongated holes so as to guarantee a good hermetic seal in the assembly.

It should be pointed out that when the head is small, a suction pipe can be used in the turbine discharge in order to increase the utilizable net head.

In some cases, no suction pipe is contemplated in the turbine design due to the fact that the turbine operates most of the time with low-percentage part loads, and the head it would gain would be negligible.

Drawing TM-01-03 shows a cross-section view indicated in Chart TM-01-01, where the assembly details of all the injector parts can be observed, along with the assembly of the runner, the shaft and the bearing supports. This drawing also indicates the way in which the bearings should be assembled (one fixed and one free), as well as the stuffing box system in the injector regulating vane support.

3.1 Design and Calculation of the Injector.

The TM-02 series of drawings correspond to the detailed drawings of the injector. In Drawing TM-02-01 the injector dimensions can be observed with the regulating vane unassembled. It also shows a transverse cross-section wherein complementary dimensions are indicated. In this drawing the details of the side walls can also be observed; these walls will receive the bearing supports, which will guarantee correct mounting between the injector and the runner.

In Drawing TM-02-02 a view of the injector at the turbine entrance is shown where the location of the threaded holes is detailed. It also provides complementary dimensions to those shown in the preceding drawing. It indicates the details of the location of the two compartments of the injector, which are separated by the housing for the bearing support of the regulating vanes. In the case of one-compartment injectors, this support would not be necessary.

Drawing TM-02-03 shows an upper view of the injector to complete the design.

The mechanical calculation of the walls considers each one of them as a flat plate with fixed edges and uniformly loaded. An additional thickness is also assumed in order to anticipate wear due to water abrasion.

Drawing TM-02-04 indicates the dimensions of the regulating vanes' profiles for each injector compartment. This piece can be cast, and it is recommended that aluminum bronze be used, the chemical and mechanical characteristics of which can be obtained from Table No. 3. However, appropriate stainless steel materials can also be used in turbines (as noted in the same chart).

The diameter of the shaft of the regulating vane is calculated on the basis of the maximum torque required for regulation, according to the following formula:

$$T = 31 \, De \, Q \sqrt{H} \qquad \text{in Kg - m}$$

and its variation, as a function of the degree of opening, is represented in Figure 10. This formula is applied to each compartment of the injector, and the water flow Q is equal to the maximum flow through the compartment.

The diameter of the shaft is obtained with the following formula:

$$d_i^3 = \frac{16 \, T}{\pi \, Sd}$$

where:

Sd is the design stress of the material used in the regulating vane, considering that the latter and the shaft constitute a single piece.

As is known, the design stress of a shaft with a key-seat is equal to:

$$Sd = 0.20 \, Sy$$

where Sy is the yield stress of the shaft material.

Due to limitations of space, the maximum diameter allowed in the shaft of the regulating vane for each runner diameter is shown in the following table:

De (m.m.)	di (m.m.)
300	38
400	50
500	63
600	76

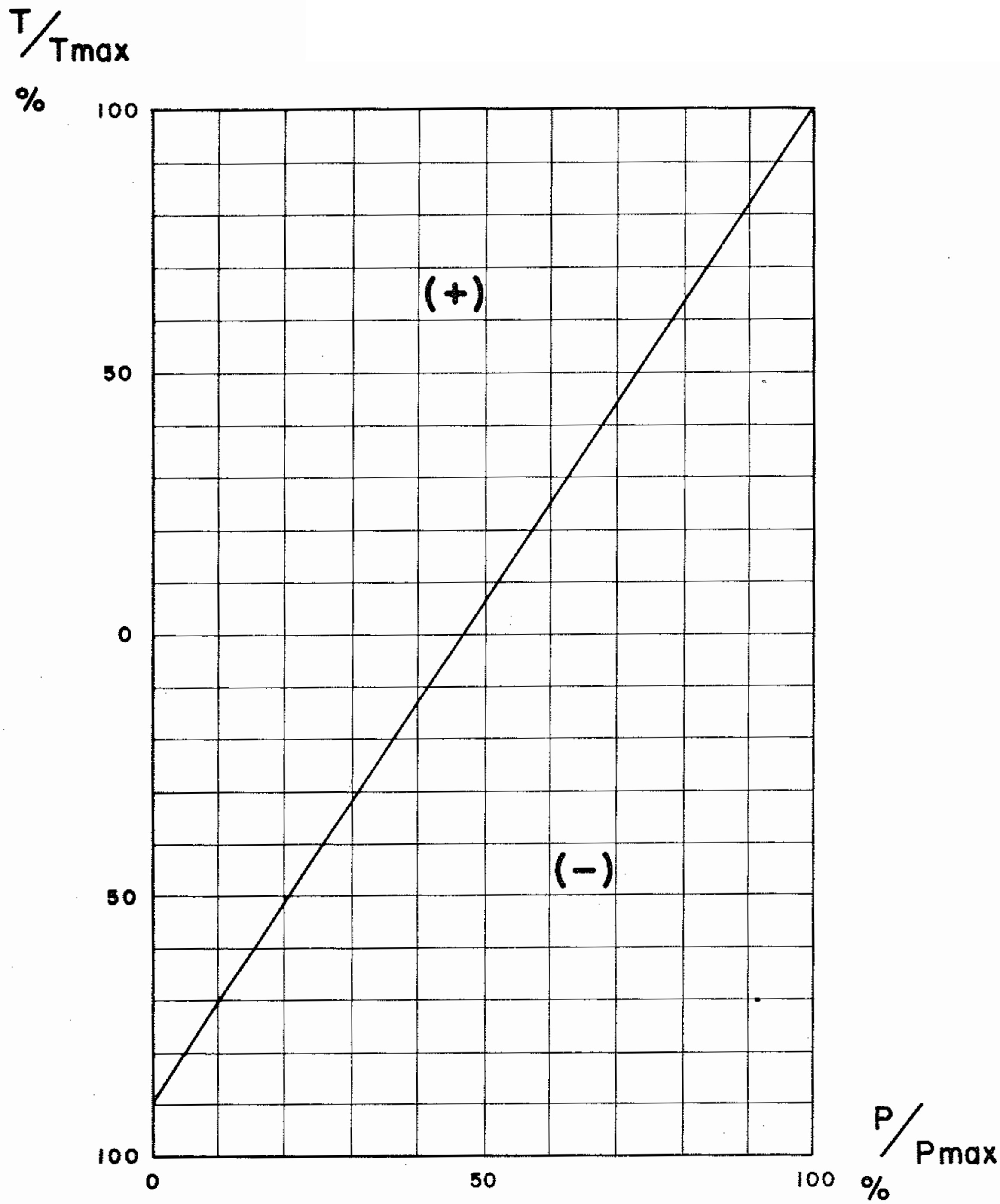
COMPOSITION OF ALLOYS COMMONLY USED IN HYDRAULIC TURBINES

MATERIAL	C %	Mn %	Si %	Cr %	Ni %
STEEL WITH 13 % Cr	0.10	0.5	0.4	12.5	0.9
STEEL WITH 18 % Cr, 8 % Ni	0.07	0.5	1.0	18.0	9.0
STEEL WITH 2 % Ni	0.24	0.7	0.3	0.2	0.2
STEEL WITH 1.5 % Mn	0.24	1.6	0.3	0.2	0.4
BRONZE - ALUMINUM	Al 10.0	Fe 8.0	Mn 5.0	Ni 2.0	Cu DIFFERENCE

MECHANICAL PROPERTIES OF ALLOYS USED IN HYDRAULIC TURBINES

MATERIAL	YIELD STRESS kg/mm^2	TENSILE STRESS kg/mm^2	MINIMUM ELONGATION L = 5 d %	MINIMUM IMPACT RESISTANCE kg/cm^2	BRINELL HARDNESS kg/mm^2	FATIGUE LIMIT kg/mm^2
STEEL WITH 13 % Cr.	45	65-75	15	4	190-30	30
STEEL WITH 18% Cr, 8% Ni	15	40-50	30	18	130-170	13
STEEL WITH 2 % Ni	35	55-65	18	6	155-195	22
STEEL WITH 1.5 % Mn	34	50-60	22	6	140-180	18
BRONZE - ALUMINUM	30	60-70	7	1	190-230	15

TABLE 3



PERCENTAGE VARIATION IN TORQUE IN THE AXIS OF THE INJECTOR GUIDE VANE, ACCORDING TO LOAD VARIATIONS IN THE MICHELL - BANKI TURBINE

FIGURE 10

Finally, Drawing TM-02-05 details the design of each injector part. These are: the lateral and central bearings supports, which are mounted on the injector's side wall, the sliding bearings, the stuffing box and the flow divider for the two-compartment injectors.

3.2 Design and Calculation of the Runner

Drawings TM-03-01 and TM-03-02 show details of the runner design. The first provides its dimensions and the second the layout of its parts. For the runner, fabrication can be assumed to be based on stainless steel plates with shaft hubs of mild steel, avoiding the contact of the hub with the water so as to reduce corrosion effects.

There are several alternatives for fitting the runner blades on the disk. One of these is proposed in the aforementioned drawings, and it consists of milling the blade profile on the disk and then assembling the blades and fixing them with an external weld. This provides a better finish to the runner. It should be pointed out that runners can also be cast in one piece.

The thickness of the blades is generally assumed and later a stress verification is made; testing the blade as a beam fixed at both ends by means of welding, and uniformly loaded. The force to which each blade is subjected, considering the unfavorable case presented when the runner is braked for some reason and the turbine is working with the injector totally open, is expressed by (see Figure 11):

$$F = \frac{\gamma\, Q\, C_2 \,\mathrm{Cos}\, \theta'}{g_0\, K_0\, 2}$$

where:

- F is the component y, of the water forces on each blade, in kg
- Q is the maximum flow through the nozzle, in m^3/s
- C_2 is the waterspeed at the injector exit, in m^3/s
- Z is the number of blades in the runner
- g_0 is a constant equal to $9.81 \dfrac{kg - m}{kgf - s^2}$
- K_0 is the intake arch percentage
- θ' is an angle determined by:

$$\theta' = a_2 + \theta^0$$

Where:

$$\theta^\circ = \mathrm{arc\ Cos} \left| \frac{re^2 + r_3{}^2 - ri^2}{2\, re_2\, r_3} \right|$$

with:

$$r_3 = 2r\ \mathrm{Sin}\ (\phi/2)$$
$$re = De/2$$
$$ri = Di/2$$

The maximum stress to which the runner blade will be subject due to the action of the water's force on the blade length can be obtained from the following formula:

$$\sigma\ max = \frac{F\ Br\ c}{12\ Igx}$$

where:

Br is the runner width, determined with the detailed design and expressed in meters
c is determined by:

$$c = (r + e) - Cg$$

where Cg is the center of gravity of the blade, determined as follows:

$$Cg = \frac{120\ \left[(r + e)^3 - r^3 \right] (\cos\theta_1 - \cos\theta_2)}{\phi\ \pi\ \left((r + e)^2 - r^2 \right)}$$

where:

e is the blade thickness, in meters
r is the radius of curvature of the blade, in meters

$$\theta_1 = \frac{180 - \phi}{2}$$

where:

$$\theta_2 = \phi + \theta_1$$

ϕ is the angle of curvature of the blade, in radians
Igx is the moment of inertia of the blade, determined by:

$$Igx = \frac{(r + e)^4 - r^4}{8} \left| \phi - \frac{\mathrm{Sen}\ 2\theta_2 - \mathrm{Sen}\ 2\theta_1}{2} \right|$$

Note that in this case ϕ is expressed in radians.

For the stress verification, the maximum stress on the blade must have a value of less than 66 o/o of the yield stress Sy of the material selected for the blade.

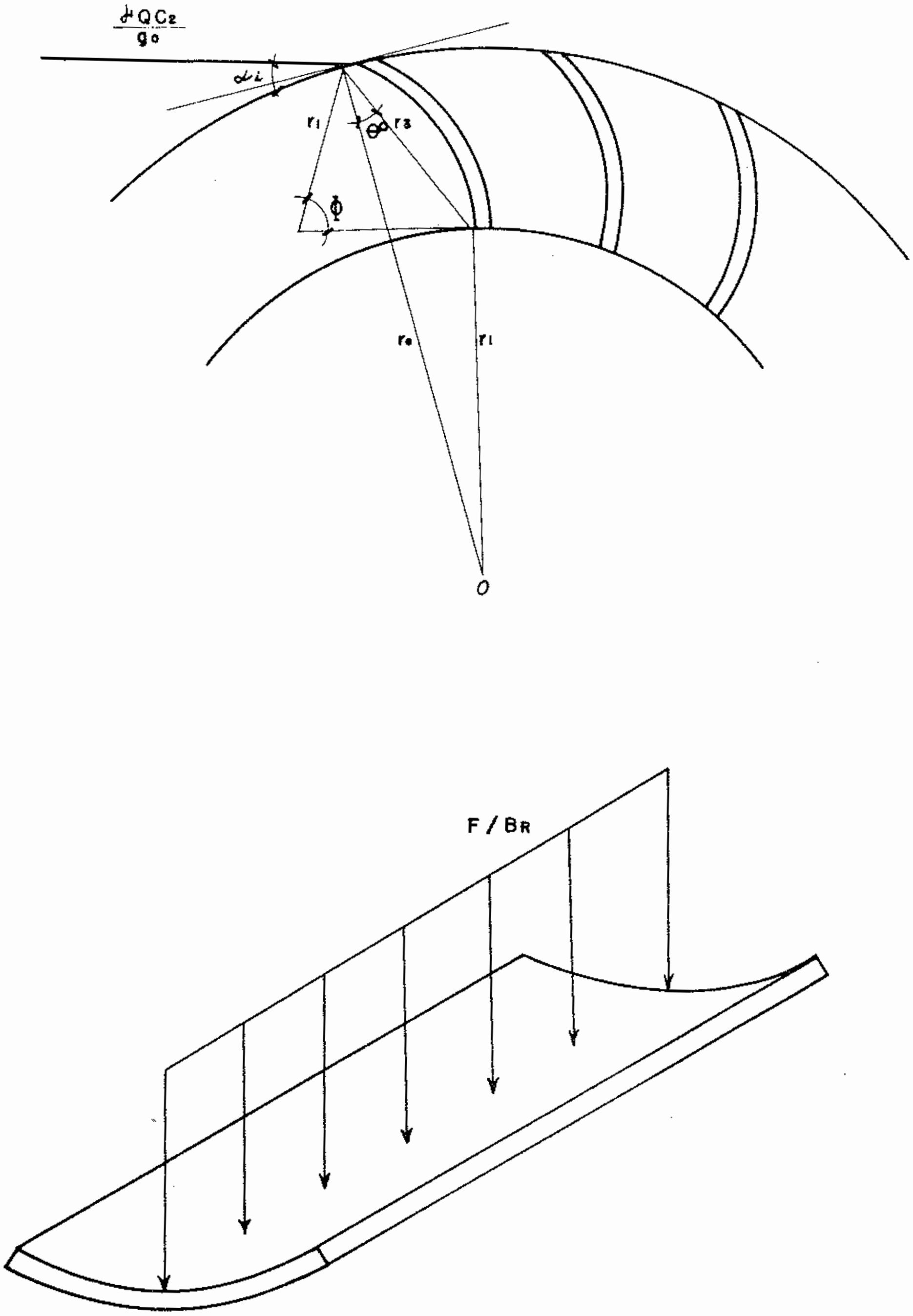
For all practical purposes, and in the cases where the runner has the same geometry as the one developed in this manual, it can be considered that:

$$F = 46.5\ Q \sqrt{H}$$

and the values for Cg, Igx, and c can be obtained from Table No. 4, where these values have been calculated for different blade diameters and thicknesses.

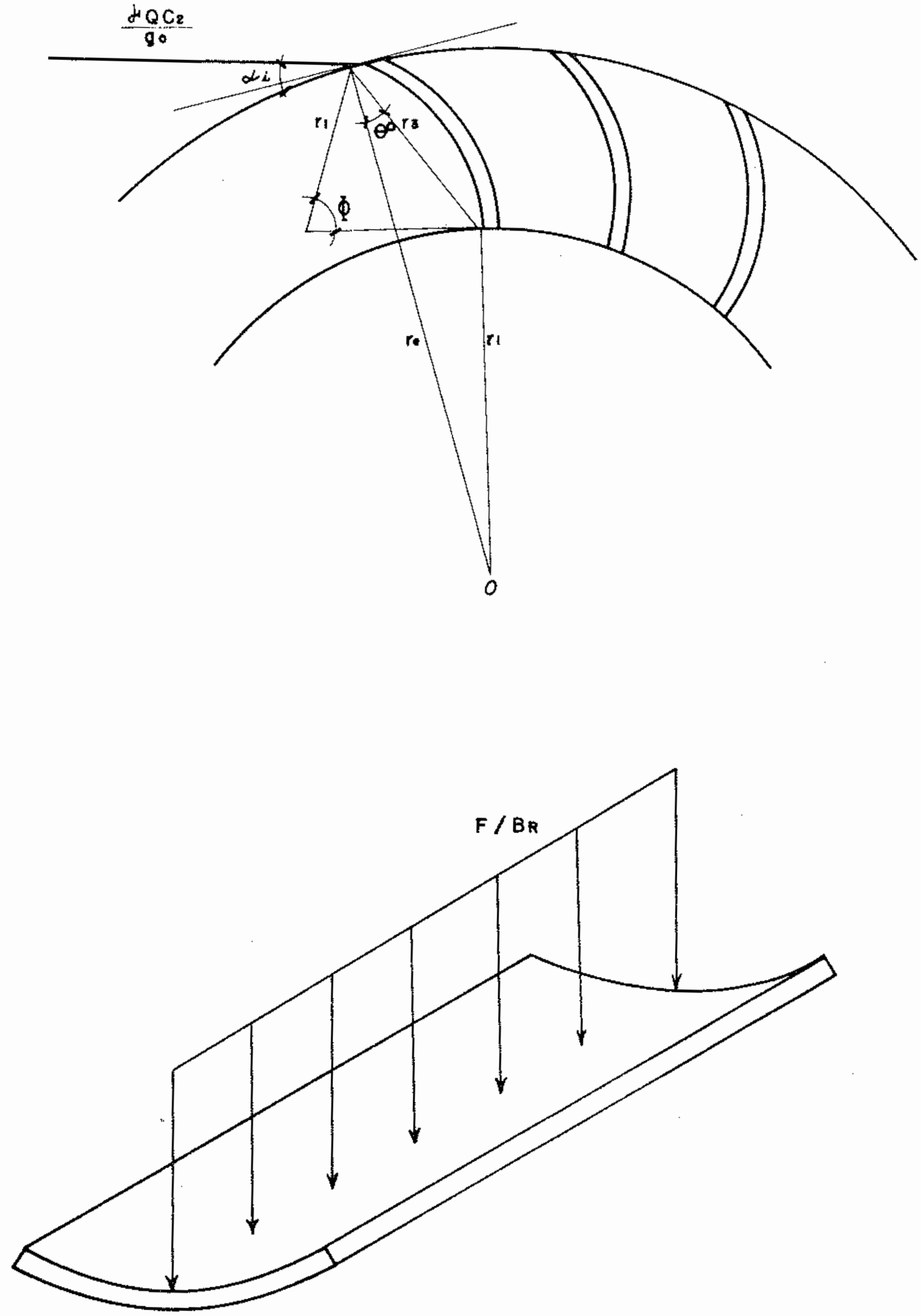
The thickness of the disk is assumed with the criterion of avoiding its deformation by the effects of heat during the welding process to fix the blades on it.

The runner hub is designed on the basis of the diameter of the main shaft, and it is recommended that the outer diameter be equal to twice the diameter of the shaft.



FORCES ACTING ON THE RUNNER BLADE

FIGURE II



FORCES ACTING ON THE RUNNER BLADE

FIGURE II

CENTER OF GRAVITY AND MOMENT OF INERTIA OF THE RUNNER BLADE

RUNNER DIAMETER De (mm)	BLADE THICKNESS e (mm)	AREA OF STRESS A (cm^2)	CENTER OF GRAVITY Cg (cm)	MOMENT OF INERTIA Igx (cm^4)	TURNING RADIUS C (cm)
300	2	1.27	4.66	0.1134	0.43
300	3	1.93	4.70	0.1822	0.48
300	4	2.59	4.75	0.2633	0.53
300	6	3.96	4.85	0.4751	0.64
400	2	1.68	6.18	0.2610	0.54
400	3	2.55	6.22	0.4098	0.59
400	4	3.42	6.27	0.5760	0.64
400	6	5.21	6.37	0.9779	0.75
500	2	2.10	7.70	0.5018	0.65
500	3	3.17	7.75	0.7780	0.70
500	4	4.25	7.79	1.0778	0.75
500	6	6.46	7.89	1.7686	0.86
600	2	2.52	9.23	0.8587	0.75
600	3	3.79	9.27	1.3215	0.81
600	4	5.08	9.32	1.8146	0.86
600	6	7.70	9.41	2.9159	0.97

NOTE: These values are valid for runner blades with $\alpha i = 16°$

TABLE 4

The key-seat canals that are located therein should be sized according to standard key seats, whose width will be approximately one fourth the diameter of the shaft and whose length will be 1.2 to 1.3 times the diameter of the shaft.

To facilitate the assembly of the runner on the shaft, the inner diameter of one of the hubs should be designed to be slightly larger than that of the outer; this makes it necessary to design the shaft with scaled diameters.

3.3 Design and Calculation of the Main Shaft

Drawing TM-04-02 shows the details of the shaft design, based on Drawing TM-01-03, which shows the details of its assembly with the bearings and the runner.

The turbine shaft is designed considering that the turbine will transmit its power to the generator by means of a coupling or a transmission system based on belts or gears. With this criterion, a typical diagram of force and momentum can be obtained, as illustrated in Figure 11.

The mechanical calculation of the shaft is done using the ASME formula, with which the minimum shaft diameter is determined. This formula is as follows:

$$d^3 = \frac{16}{\pi\, Sd} \sqrt{(Km\ Mmax)^2 + (Kt\ Tmax)^2}$$

where:

Mmax is the maximum flector momentum presented in the shaft, in kg-m; it is obtained with the following formula:

$$Mmax = \sqrt{M_x^2 + M_y^2}$$

with:

$$M_x = \frac{Fr\ .\ a}{2}$$

$$M_y = \frac{Pr\ .\ a}{2}$$

Pr is the runner weight, in kg.
Fr is the tangential force on the runner, in kg:

$$Fr = \frac{1948\ P_T}{N\ De}$$

P_T is the turbine power, in kW.
N is the number of revolutions of the turbine, in RPM
De is the outside runner diameter, in meters
Tmax is the maximum torsional moment applied to the shaft, in kg-m; it is obtained with the following formula:

$$T = \frac{974\ P_T}{N}$$

Km is the bending momentum flector for a stable load estimated with a value of 1.5

Kt is the bending momentum factor for a stable load estimated with a value of 1.0

Sd is the design stress, in kg/m^2, of the material used for the shaft, estimated as 20 o/o of the yield stress value when a key-seat canal is used

To define the steps of the shaft diameters, the dimensions of the bearings and standard retaining rings commercially available should be considered, and that because of turbine operating reasons, the maximum shaft diameter permitted inside the runner will be given by:

$$d = 0.328 \, De$$

Once the diameter of the shaft has been determined, it is recommended that this be checked in light of the critical speed of the shaft, which should be higher than the tripping speed, which, in the case of the Michell–Banki turbine, has a value equal to 1.8 times the rated (nominal) turbine speed.

The first critical speed, in RPM, for the diagram of forces in Figure 11, would be expressed as follows:

$$N\,crit = \frac{29.88}{y\ 1/2}$$

where:

N crit is expressed in RPM

y is the deflection produced in the shaft by the action of the weight and force of the runner. For the Figure 11 diagram, y is equal to:

$$y = \frac{Wa^2}{6EI}(3l_0 - 4a) \qquad \text{in meters}$$

where:

$$W = \sqrt{Pr^2 + Fr^2}$$

$$E = 2.1 \times 10^{10} \ Kgr/m$$

I = the moment of inertia of the shaft section, in m^4

$$I = \frac{\pi d^4}{64}$$

It is recommended that the critical speed be approximately 40 o/o higher than the turbine's tripping speed.

3.4 Design of the Bearings Support

The design of a bearings support requires the previous selection of the bearing. This is done by determining the required dynamic base capacity:

$$C = (X F_r + Y F_a) \left| \frac{60 N L_h}{10^6} \right|^p$$

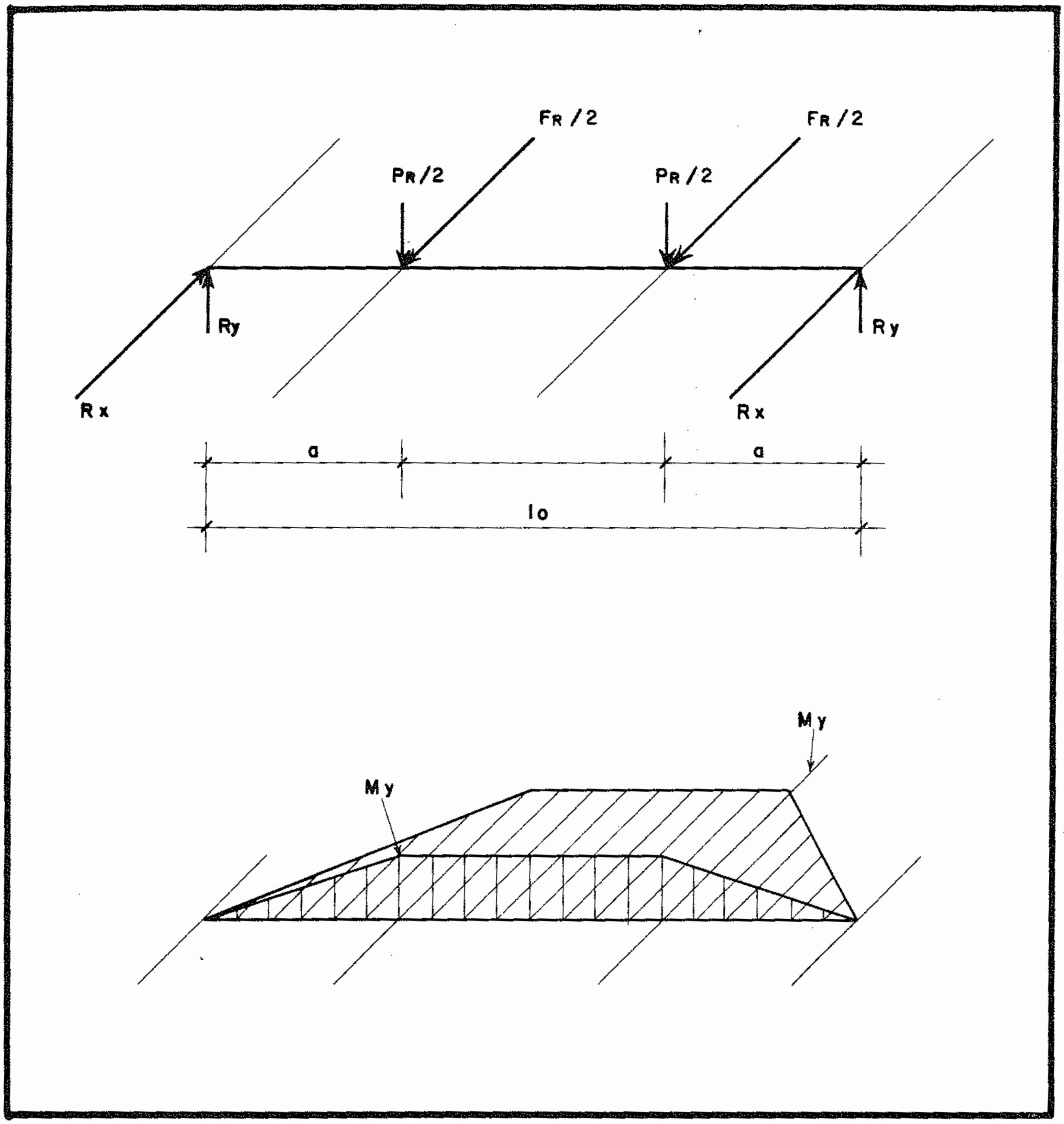


DIAGRAM OF FORCES AND MOMENTS IN THE TURBINE AXIS

FIGURE 12

where:

C	is the minimum dynamic base capacity required for the bearing, expressed in kWh.
X	is the radial bearing coefficient, considered as 1("one")
Y	is the axial bearing coefficient, which in the case of a turbine does not exist, because no axial load F_a exists
N	is the turbine's number of revolutions per minute
L_h	is the nominal expected life, in hours
p	is 1/3 for ball bearings and 3/10 for roller bearings
F_r	is the radial load on the bearing, determined from the diagram of forces and momentum in Figure 11

$$F_r = \sqrt{R_x^2 + R_y^2}$$

With the dynamic base capacity, the shaft diameter, and the maximum number of revolutions per minute at which the turbine operates, the appropriate bearing can be selected from commercial catalogues.

With the dimensions of the bearing, as found in the catalogue, the bearing support is designed to ensure a simple assembly system and to avoid the entry of water to the bearings. Drawing TM-05-01 presents the design for a bearings support where housing has been provided for two retainers or seals: one for water and one for grease; between these, a water trap has been designed to prevent the entry of water to the bearings.

3.5 Design of the Regulating Mechanism

As mentioned in previous sections, the injector of the Michell—Banki turbine can be designed with one or two compartments. In those cases where the injector is designed with only one compartment, the regulating mechanism consists of only a lever installed in the shaft of the regulating vane, which is attached directly to the speed regulator. When the injector is designed with two compartments, it is necessary to design a regulating mechanism which combines the opening and closing processes of both compartments, in order to obtain good efficiencies when the turbine is operating at part loads.

The drawings of the TM-06 series show the details of the design of the parts which, when assembled, comprise the regulating mechanism. These parts are the lever support, the levers, and the lever arms.

Drawings TM-06-03 and TM-06-04 illustrate the dimensions of the regulating cams or levers used in those turbines having two-compartment injectors, where one has a width equal to half that of the other. The canal, or slot, of these cams is designed so that the opening process will occur in three stages.

—In the first stage, the injector keeps the larger compartment completely closed, while the smaller compartment begins its total opening.

—In the second stage, the injector operates by closing the smaller compartment and simultaneously the larger compartment begins its opening process.

—In the third stage, the injector keeps the larger compartment completely open while the smaller compartment again begins its opening process; finally, both compartments remain completely open.

Drawing TM-06-03 corresponds to the cam that regulates the larger compartment, and Drawing TM-06-04 to the cam that regulates the smaller one.

Drawings TM-06-07 and TM-06-08 show the regulating cams for those injectors that have two compartments of equal size. In this case, the regulating process has two stages.

—In the first stage, the nozzle keeps one of the compartments closed while the other is opening.

—In the second stage, the nozzle keeps one of the compartments open while the other is opening, and finally both compartments remain completely open.

The closing process, just as in the preceding example, is carried out inversely to the opening process.

In Drawing TM-01-01 it can be observed that when the regulating cam turns, it makes the bearing located at the end of the main arm turn also; when the arm revolves, it activates the lever of the regulating vane in the injector. This also occurs on the backside, where the lever of the regulating vane of the other compartment is located.

To select the bearing that is displaced along the canal or slot of the cams, it is necessary to determine the radial force that will act on it. This force can be determined on the basis of an analysis of the lever arms starting with the formula for the maximum torque required to turn the regulating vane. Once the radial force on the bearing has been determined, then it can be selected in the same way as the turbine bearings were selected.

Drawing TM-06-06 illustrates the details of the lever arm design. The threaded rod and nut system, located in the regulating lever of the injector and on the secondary arm, has the purpose of fine adjustment of the dimensions during assembly. The dimensions of the section of the lever arms can be calculated considering rods subject to traction and bending.

Drawing TM-06-01 shows the assembly of the structure that supports the regulating cams; it is designed to house the sliding bearings made of phosphorus bronze, in a mechanical tube, and the point where the pivot of the main arm is located.

In Drawing TM-06-02, the dimensions of the base structures can be observed; these can be fabricated out of mild steel plate.

The shaft which supports the cams of the regulating mechanism, and which will be coupled to the speed regulator, is designed taking into consideration the diagram of forces shown in Figure 12 and applying the ASME formula:

$$d^3 = \frac{16}{\pi\,Sd}\sqrt{(Km\ Mmax)^2 + (Kt\ Tmax)^2}$$

$$Mmax = aF$$

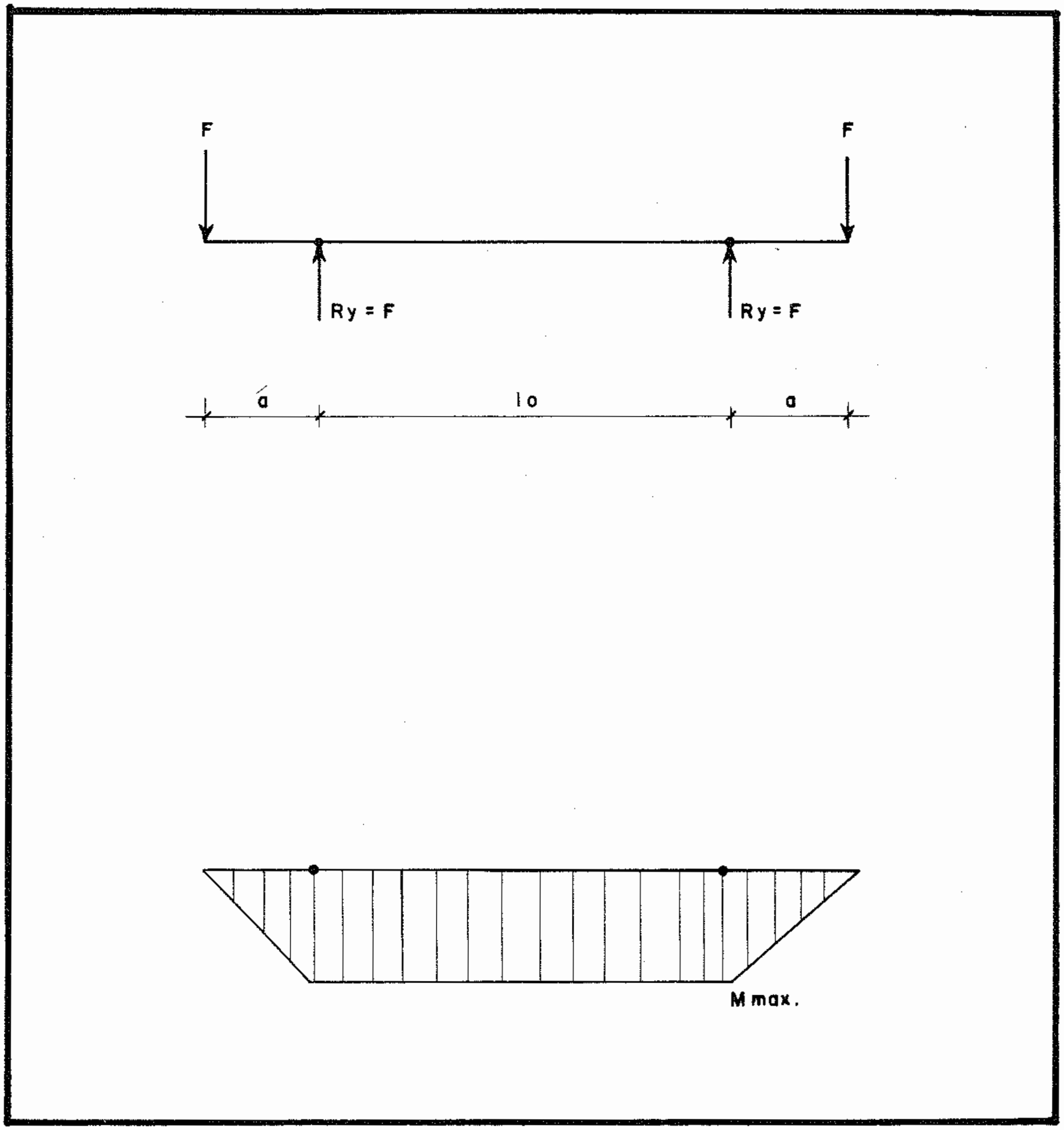


DIAGRAM FORCES AND MOMENTS IN THE REGULATING CAM AXIS

FIGURE 13

where:

F is the radial force on the bearing that is displaced along the canal or slot of the regulating cam; this value is found from an analysis of the forces in the lever arms of the regulating mechanism

Tmax(the torque) can be considered negligible

The Km factor is considered equal to 1, so that:

$$d^3 = \frac{16aF}{\pi\ Sd}$$

The diameter steps for this shaft is determined considering its assembly with the sliding bearings and regulating cams.

The sliding bearings are designed considering force F as a radial load, since no axial load exists.

The design of the regulating mechanism is concluded by determining the dimensions of the commercial pins that are used at the joints of the lever arms.

3.6 Design of the Casing

Drawings TM-07-01 and TM-07-02 show the alternative designs for the casing. Therein, it can be observed that the holes of the flanges are elongated so as to obtain a tight adjustment during its assembly with the other parts of the turbine.

4. STANDARDIZATION AND SELECTION

Once the design of the Michell—Banki turbine is known, criteria can be established to define a standardized series of this type of turbine, to select it and, in some cases, to relocate it in a specific project.

Currently, many turbine manufacturers have opted for defining standardized series, which have the advantage of optimizing designs and thus reducing engineering and manufacturing costs. Standardization makes it possible for the project engineer to choose a suitable turbine; and with basic knowledge of its design, he is also able to relocate this turbine in other projects.

4.1 Standardization

Standardization of Michell—Banki turbines consists of designing a suitable number of turbines so that they are complementary in terms of their field of application and so that on the whole they will cover the range of applications for this type of turbine.

To establish a series of standardized turbines, the formula for the specific number of revolutions can be used, expressed as a function of the flow rate Q:

$$N_q = N \frac{Q^{1/2}}{H^{3/4}}$$

where the optimal value for the turbine's number of revolutions is substituted:

$$N = \frac{29.85\sqrt{H}}{De}$$

and where:

De is the outside runner diameter expressed in meters
H is the net head, in meters

With these two expressions, the specific number of revolutions N_q is obtained, expressed as a function of the runner diameter:

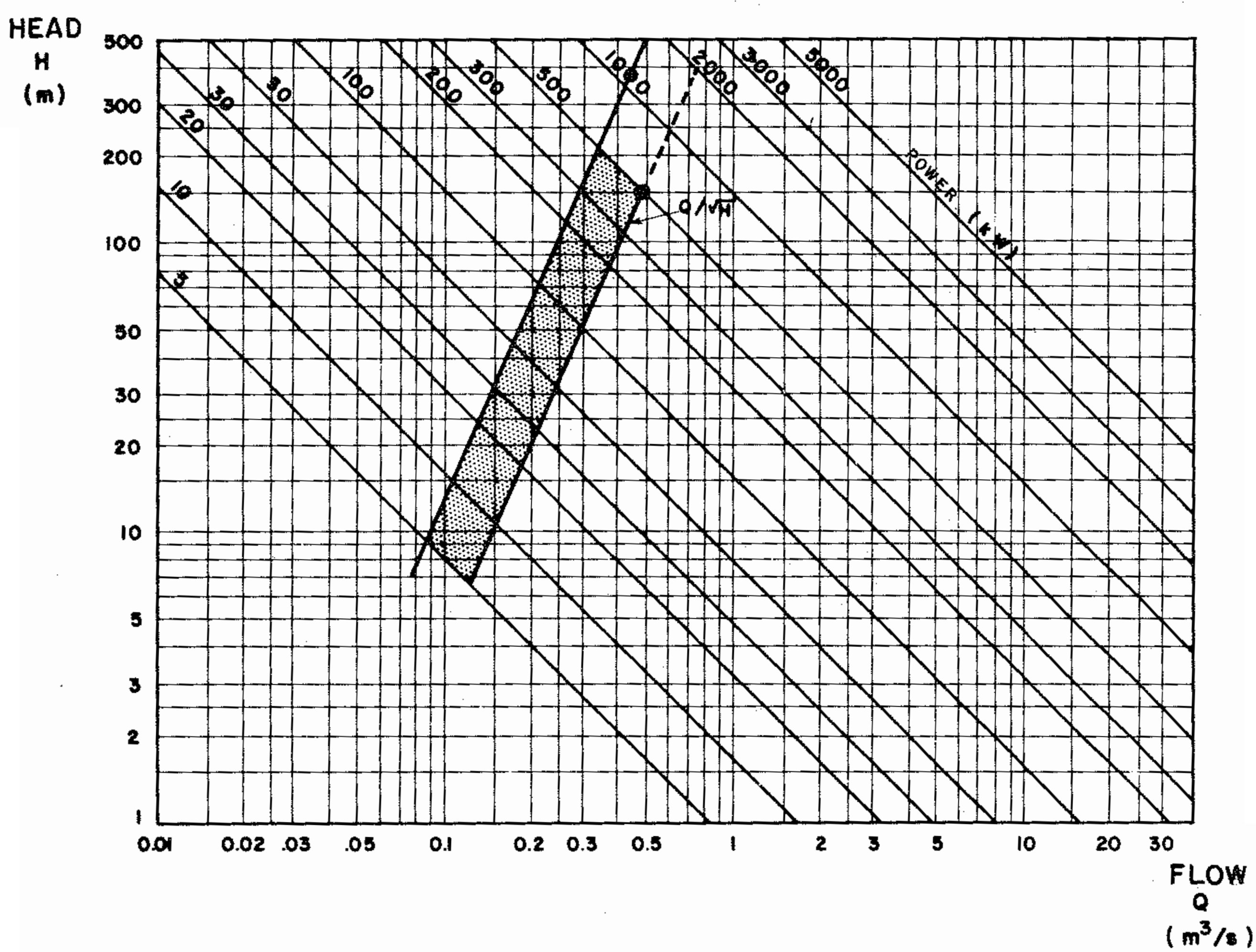
$$N_q = \frac{39.85\ Q^{1/2}}{De\ H^{1/4}}$$

From this expression, it can be deduced that when a Michell—Banki turbine is designed for given head and flow conditions, by assuming the runner diameter, a specific number of revolutions is being defined; and this corresponds to the turbine dimensions. Thus, from the point of view of hydraulics, the turbine could operate in any combination of head and flow that would fulfill the following expression:

$$\frac{Q}{\sqrt{H}} = \left| \frac{De\ N_q}{K} \right|^2 = \text{constant}$$

If the same turbine were operated with a partial load, it could satisfy a greater number of combinations of heads and flows, as illustrated in Figure 14.

As a result, other turbines can be designed to complement its application; and thereby, it would be possible to cover the range of applications of the Michell—Banki



HEAD
H
(m)
500
300
200
100
50
30
20
10
5
3
2
1
5
10
20
30
50
100
200
300
500
1000
2000
3000
5000
POWER (kW)
Q/√H
0.01
0.02
.03
.05
0.1
0.2
0.3
0.5
1
2
3
5
10
20
30
FLOW
Q
(m³/s)

FIGURE 14

turbine. Figure 15 shows that each area corresponds to the field of application of a given standard turbine.

In this case, transmission between the turbine and the generator is obtained by means of belts or gears. When it is desired to use transmission with direct coupling, it will be necessary to subdivide the field of application, using different runner diameters, according to the synchronous speeds of the generator.

To define the total number of areas of application of standardized turbines, it is necessary to define the lower limit of the percentage of partial load with which it is recommended that the turbine be operated so as to satisfy the maximum design demand corresponding to the station. This percentage can be deduced from the characteristic curve of the turbine efficiency with a partial load, and considering that the maximum demand will occur 15 to 20 years after the turbine has been installed, according to the demand projections for the project.

In the case of the Michell–Banki turbine, standardization must consider that due to mechanical and geometric reasons, there exists a maximum diameter limit for the shaft of the regulating vane of the injector, which in turn defines a maximum injector width for each runner diameter assumed.

It should also be pointed out that the standardization of Michell–Banki turbines offers the advantage of being able to establish several injector widths for each standard runner diameter, which in turn define the size of each standard turbine. Thus, for the sake of illustration, Michell–Banki turbines have been standardized herein, with eight turbine sizes defined, the range of application of which can be observed in Figure 15. The main dimensions of these standard turbines have been included in the drawings and tables in the appendix of this manual.

This standardization has considered as an upper limit a maximum head of 100 meters and the units will not develop more than 1000 kW of power to be delivered by the generator to the electric power system.

It should also be noted that the standardization undertaken has contemplated transmission by means of belts or gears between the turbine and the generator; and according to the head with which each turbine operates, it would turn at a different number of revolutions. The dimensions indicated in the tables of the appendix correspond to those obtained from the mechanical calculations, considering the most unfavorable conditions for the product $Q\sqrt{H}$, which as observed in the foregoing section define the mechanical stress in each of the turbine's components. This shows that several diameters can be obtained for the shaft for the turbine geometry, as a function of the head with which it is installed.

4.2 Selection and Relocation

When catalogues are available with illustrations similar to that indicated in Figure 15, the selection of the Michell–Banki turbine can be defined by the intersection of the head and flow for the project.

To determine the number of units that will be installed in a given station, it is necessary to consider the demand evaluation study for the project, because it will determine the percentage of part load with which the turbine will operate when it reaches the maximum daily demand during the first year; and if this part load percentage is more than 30 o/o, it is recommended that only one unit be used. If it is above 15 o/o, two units can be used and in exceptional cases, when this ratio is 7.5 o/o, three units can be used. These recommendations are based on the analysis of the characteristic efficiency curves of the turbine operating at part load, and on the load variations that can be observed in typical load diagrams for small hydro power stations.

When catalogues including illustrations like the aforementioned one are not available, it is recommended that a turbine be requested from an equipment supplier, providing him with the following details:

a) Turbine brake power
b) Net head of the station
c) Number of units required
d) Required speed regulation system
e) Physical and chemical characteristics of the water (amount of solids, acidity levels, etc.)

The manufacturers or suppliers should also be requested to include the following technical details in their quotation.

- Turbine brake power
- Utilizable net head
- Maximum flow for operating at full load
- Optimal rotation speed
- Efficiency
- Part load operation curves
- Inertia G D^2
- General dimensions and weights
- Materials used in its components, e.g. runner, injector, regulating vane, shaft, casing, etc.
- Availability of spare parts
- Instruments required for operation
- Type of tools required for maintenance

Ultimately, the final selection of the turbine should result from a technical and economic evaluation that would take into account the following criteria:

- Investment costs and payment conditions
- Delivery time
- Cost of spare parts
- Efficiency
- Possibilities for obtaining or manufacturing spare parts locally
- Previous experiences in terms of the lifetime of other turbines produced by a given manufacturer
- Infrastructure required for maintenance
- Complexity of operation

A particular case of turbine selection occurs when it is not obtained from a manufacturer or supplier but rather selected from a group of turbines that are in good condition but out of service, because the hydro power station where they were operating was enlarged or replaced by a substation of an electric power system. In these cases the turbine can be relocated and installed in a new project.

Unlike electric generators, in order to relocate a turbine, in addition to the brake power data, it is absolutely necessary to know also the head and maximum flow conditions with which it initially operated. This can be verified by observing Figure 16, which shows the flow-head data, including power and maximum-load operation curves for a geometrically given turbine.

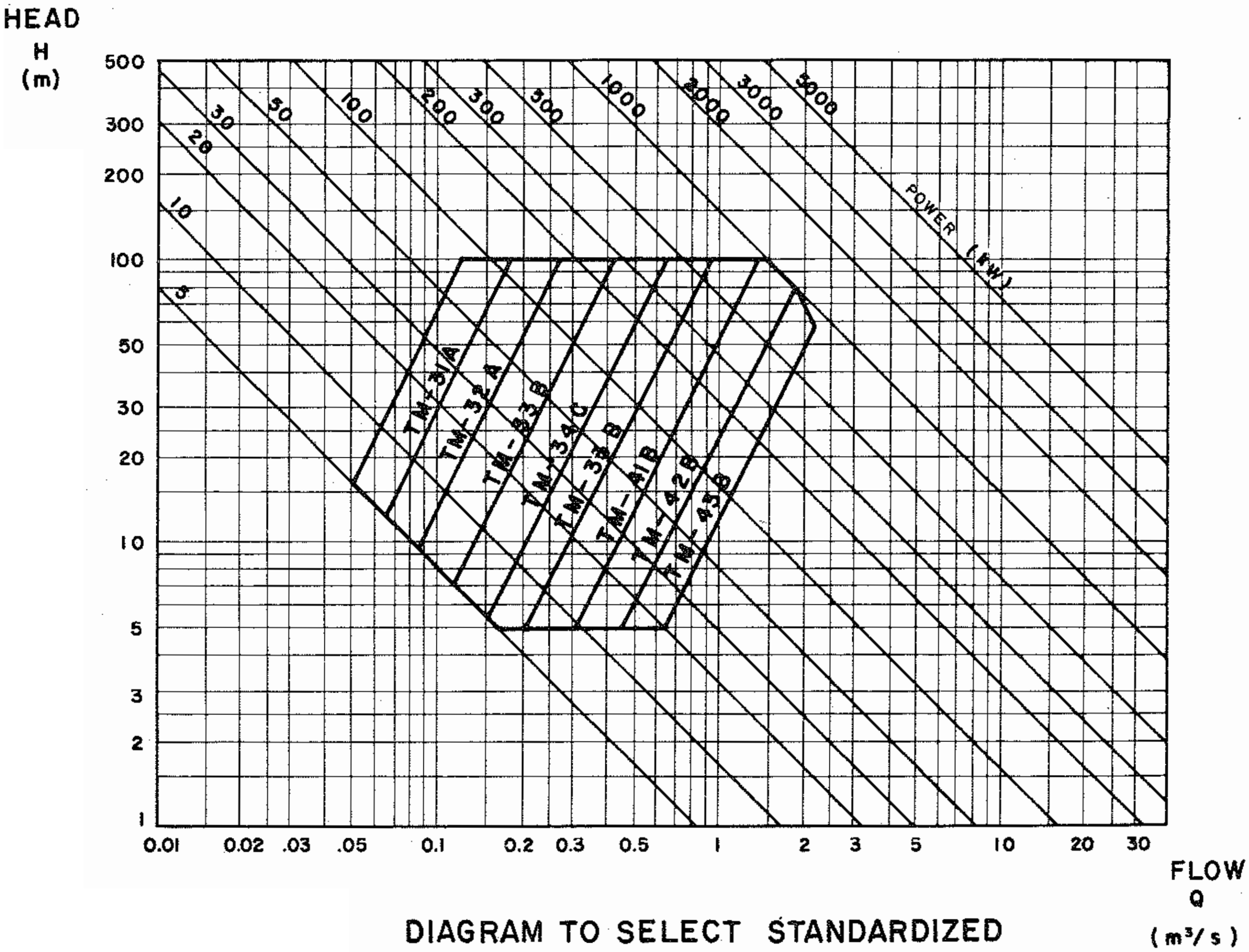


DIAGRAM TO SELECT STANDARDIZED
MICHELL-BANKI TURBINES

FIGURE 15

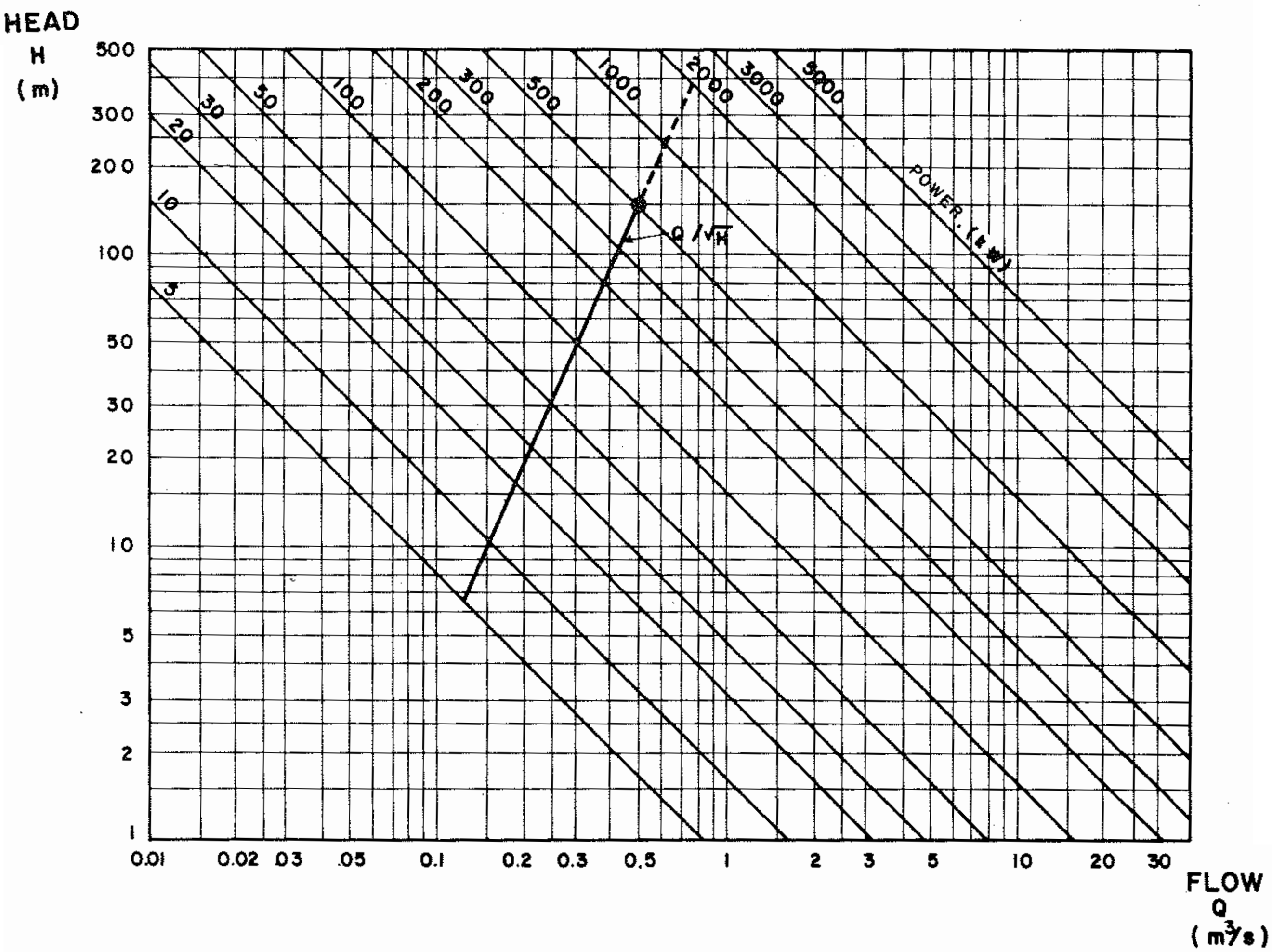

HEAD
H
(m)
500
300
200
100
50
30
20
10
5
3
2
1
0.01
0.02
.03
.05
0.1
0.2
0.3
0.5
1
2
3
5
10
20
30
3
10
20
30
50
100
200
300
500
1000
2000
3000
5000
POWER (kW)
$Q/\sqrt{H}$
FLOW
Q
(m^3/s)

FIGURE 16

In order to know if a turbine is in condition to be relocated in a hydroelectric project, it is necessary to verify two things: one of a hydraulic nature, by means of which it is determined if the turbine geometry will allow the head and flow conditions of the project to be satisfied; and the other, of a mechanical nature, which makes it possible to determine if the materials used in the turbine will be sufficiently resistant to withstand the new operating conditions.

The hydraulic-type verification can follow the procedure outlined below:

—When the head and power or flow data with which the turbine operated are known, the $Q/\sqrt{H}$ characteristic can be determined. When these data are unavailable, it will be necessary to determine them on the basis of the runner geometry, for which purpose the turbine will be re-designed.

—With the head and flow data for the project where the turbine will be relocated, the $Q/\sqrt{H}$ characteristic is determined and compared with the corresponding characteristic obtained for the turbine.

—The turbine will be useful for the project if the percentage obtained for the following expression is less than 100 o/o:

$$\text{o/o} = \frac{Q/\sqrt{H}\text{ of the project}}{Q/\sqrt{H}\text{ of the turbine}} \quad 100$$

This percentage represents the part load percentage with which the turbine will operate when satisfying the design power capacity of the project.

—To select the turbine, it will be necessary to determine, with the aid of part load operation curves for the turbines, if the percentage of the maximum load that the turbine will reach when developing the project's design power, will guarantee a good operating efficiency.

The mechanical-type verification consists of the mechanical re-design of the turbine, and it will only be necessary when the design power of the project is greater than the maximum power output developed by the turbine in its initial installation. The mechanical re-design will consider mainly the calculation of the shaft on the basis of critical speed, keeping in mind that the turbine will operate with a different head and will therefore have a new optimal speed of rotation.

When the turbine is relocated, the transmission system to couple it to the generator will also have to be designed so as to guarantee the operation of the turbine at its optimal speed.

5. MANUFACTURING RECOMMENDATIONS

The last stage of the design of a turbine consists of elaborating manufacturing drawings which transmit to the manufacturer all of the indications and specifications necessary to produce each part of the turbine. These specifications are as follows: the materials with which each part is produced, specifications for fits, tolerances and finishing, and the production process to follow in order to make each part.

To manufacture turbines of the Michell–Banki type, a general engineering workshop is needed, with universal machine tools such as a lathe, a milling machine, a hydraulic, a drilling machine and a planer. Welders with experience in working with stainless steel are also required. As will be detailed further on, there are some parts that can be produced by a casting process; therefore, the workshop should have the infrastructure necessary for this process or should sub-contract the production of these pieces with a specialized foundry.

Below, a series of recommendations and alternatives are listed for the industrial production of each part of the Michell–Banki turbine.

5.1 Manufacturing the Injector

To manufacture the injector of a Michell–Banki turbine, it is recommended that materials similar to those indicated in Table 3 be used.

The structure of the injector can be produced on the basis of bent and welded steel plates, or through a casting process with final finishing by hand. In the case of production on the basis of bent and welded plates, mild steel can be used for the flanges and side walls where the bearing supports are located; the rest of the structure would be made from stainless steel.

To manufacture the regulating vane of the injector, it is recommended that a casting process be used. Later, surface finishing will be done to obtain a smooth surface and avoid cavitation spots. It is not recommended to manufacture it from bent and welded plates because its fabrication can become too complicated, being such a small component.

The supports for the side and central bearings are pieces that should be produced through a casting process. The final dimensions will be obtained by machining with machine tools. The material to be used to produce them will be the same used for the injector structure.

The stuffing box is a simple part whose production by casting is recommended, using cast iron.

The bearings are phosphorous bronze rings mounted on the support. They are not complex to manufacture, and they only require a good surface finish with suitable tolerance.

In designing two-compartment injectors, a flow divider is used, which can be manufactured on the basis of bent and welded plates, using stainless steel materials.

Once the production alternatives for the different pieces making up the injector of the Michell–Banki turbine have been defined, the manufacturing drawings can be elaborated. They should have the following features:

—A drawing of the general assembly of the injector should be presented, with reference to the drawings where the specifications are detailed, as well as to the fits, tolerances and finishes for each piece of the injector. This drawing should include a list of the machine components that intervene in the assembly: nuts and bolts, washers, gaskets, etc. Assembly details should also be included and the required tolerances should be specified.

—The drawing of the part known as the injector structure should indicate the level of finishing and specify the tolerances for each dimension. In the case of fabrication using bent and welded plates, details and specifications for the welded joints should be indicated, as well as the electrode recommended for welding. When manufacturing is by casting process, the final dimensions of the piece should be included, and drawings of the casting model to be used in this process should be added. The list of materials should indicate the specifications of the materials to be used in manufacturing this part.

—The drawings for the regulating vane should indicate the levels of finishing and tolerances for the final dimensions of this piece. These tolerances should be compatible with the tolerances of the holes of the pieces where the vane will be mounted. The list of materials should specify the materials with which this piece will be made. It will also be necessary to include another drawing for the model to be used for its production by means of casting.

—The manufacturing drawings for the bearings supports should clearly indicate the levels of finishing and tolerances required for their assembly with the other pieces. The materials list should indicate the specifications of the materials from which these parts will be made.

—The drawing for the part known as the stuffing box should have the same characteristics as the previous drawing, and this is true also for the drawings of the bearings and the flow divider.

5.2 Manufacturing the Runner

As indicated in the detailed design, the elements that make up the runner are the blades, the disks, and the hubs. The first two can use the same or similar materials to those indicated in Table 3.3, and the third can use mild steel, in those cases where the runner is fabricated out of welded plate.

Casting is one manufacturing alternative for Michell—Banki runners. However, it has the disadvantage that the runner is a very small piece. Thus, its final finishing is very complicated and expensive. For that reason, the manufacturing process based on welded stainless steel plate is recommended.

When the runner is manufactured with a welding process, it is recommended to mill the blade profiles in the disk and, after assembling them, to do the welding outside the runner to attach them to the disk. Afterwards, the runner hub is mounted on the disk, previously executing some machining to guarantee that the hub is centered on the disk. Lastly, the final machining is carried out on the runner, as to ensure its final dimensions.

In some cases, the runner is designed with an intermediate disk in order to reduce the stresses on the blades. The mounting is similar to the lateral blades, i.e., first milling the blade profiles, and after they are assembled, inside welding is done.

The manufacturing drawings for the runner should have the following features:

—A general assembly drawing should be presented, indicating the final dimensions and tolerances for the runner. Details should also be provided for the welding joints of the blades to the disk and for the disk and hub.

—The drawing for the blade should indicate the tolerance of the radius and angle of curvature. It is preferable to dimension the angle with one degree in excess, in order to obtain the final diameter desired after the final finishing of the runner.

—The drawing for the disks should indicate the radius of curvature centers which will serve as a guide for the suitable milling of the canals where the blades will be mounted. In specifying the width of the canal, the desired tolerance should be indicated. Just as in the preceding case, it is recommended that the disk diameter be made 2mm. larger in order to be able to machine the runner after the general assembly and obtain the final dimension desired for the outside diameter.

—The drawing for the hub should indicate the tolerances of the holes where the shaft will be mounted and the tolerance for force fitting the hub on the disk, thus facilitating the final welding process. The type of thread should also be specified for the hub setscrew and the levels of finishing for the key seat canal and the shaft hole.

5.3 Manufacturing the Main Shaft

The main shaft of the Michell—Banki turbine can be manufactured out of an adequate carbon shaftsteel, which should have good resistance to bending stresses.

The manufacturing drawing for this component should contain indications relative to tolerances in the diameter of each diameter step, levels of finishing, and curvature radius for each change of section.

In the part of the shaft where the runner hubs will be fitted, it is recommended that a tolerance be considered to permit forced fitting during assembly.

5.4 Manufacturing the Bearings Supports

As indicated in the detailed design, the bearings supports of the Michell—Banki turbine can be manufactured through a casting process or by welding plates to a solid, hollow piece, as shown in drawing TM-05-01 of the detailed design.

To manufacture this part, cast iron can be used; and in the case it is fabricated from plate, mild or structural steel can be utilized. Once the manufacturing has been concluded, it is recommended that the entire piece be heat treated.

The manufacturing drawings should indicate the specifications for tolerances and levels of finishing, mainly for the area where the bearing is located. In the case of production through a casting process, grey cast iron should be used; and it is only necessary to machine the inside of the support, as indicated in the drawings. The list of materials should contain the specifications of the materials to be used in production.

5.5 Manufacturing the Regulating Mechanism

As indicated in the detailed design, the regulation mechanism is composed of several pieces, which can be manufactured by a welding process using structural steel.

The manufacturing drawings should contain details of the types of welded joints to be used in the pieces of the mechanism, as well as the areas where good surface finishing are required. The tolerances of the dimensions should also be indicated.

5.6 Manufacturing the Casing

The casing of a Michell–Banki turbine is manufactured with mild steel plate, and once the manufacturing process has been finished by welding, it is subjected to a dip galvanizing process, in order to protect it from corrosion.

The manufacturing drawings should present details of the welded joints and of the tolerances and spacing of the holes.

Finally, the manufacturing of the turbine culminates with the assembly of all of its parts, for which it is necessary to elaborate a drawing of the general assembly, including a list of complementary materials required, e.g., gaskets, bolts, washers, retainers, etc.

APPENDIX 1

A PRACTICAL EXAMPLE OF THE DESIGN AND CALCULATION OF A MICHELL–BANKI TURBINE

The aim of the present appendix is to illustrate, through an example, the process of design and calculation of a Michell–Banki turbine for a specific small hydro power station project.

A resource assessment study was done for the selected hydroelectric project; and it indicated that the head that can be tapped by the station is 40 meters, and that the minimum annual flow that can be captured by the station is 1.5 m^3/s, in keeping with a variation in flow similar to that indicated in Figure 1A.

The study done to evaluate demand determined a growth similar to that indicated in Figure 2A, with the initial maximum daily demand being 80 kW with a maximum daily output of 400 kW after 25 years. In addition, it has been considered that the load diagram has a load factor equal to 0.43.

With these data, it was determined that it would be convenient for the station to install two units, due to the fact that the initial maximum daily output (80 kW) was 20 o/o of the final maximum daily output of the station (400 kW).

The type of turbine was selected considering that the first unit would reach its maximum power output in 12 years (200 kW), according to the theoretical growth in demand. Therefore, by graphing in Figure 3A the daily load diagram of the first unit, as a function of the variation in electric power demand between the first and last years, it was determined that the Michell–Banki turbine offered a better average efficiency than that obtained with a Francis turbine, as indicated in Figure 4A. On the basis of this analysis, and considering that the Michell–Banki turbine can be manufactured at a lower cost than that of the Francis turbine, two Michell–Banki turbine units were selected for this project.

With these parameters, the design of the Michell–Banki turbine began with the determination of the brake power, for which purpose the following formula was used:

$$P_T = \frac{Pg}{\eta_g \quad \eta_{tr}}$$

where:

- Pg is the maximum power output that the generator can deliver to the electric power system, i.e., 200 kW.
- η_g is the 200-kW generator efficiency, equal to 93 o/o.
- η_{tr} is the efficiency of transmission, considering a system of belts, equal to 95 o/o.

Replacing the values, a brake power of 226.4 was obtained for the turbine.

With this power, the design flow was determined with the following formula:

$$Q = \frac{P_T}{9.807 \; H \, \eta_T}$$

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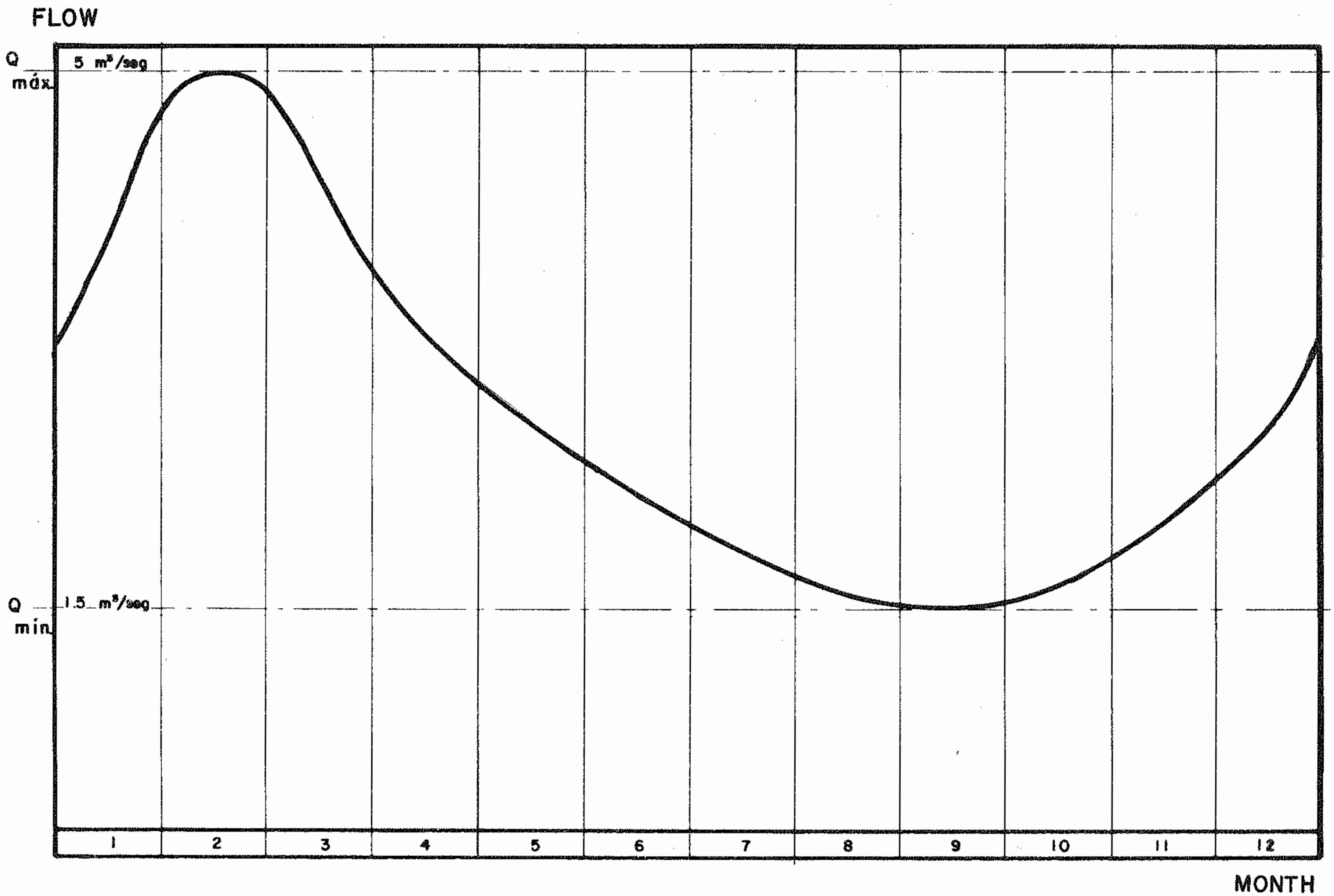
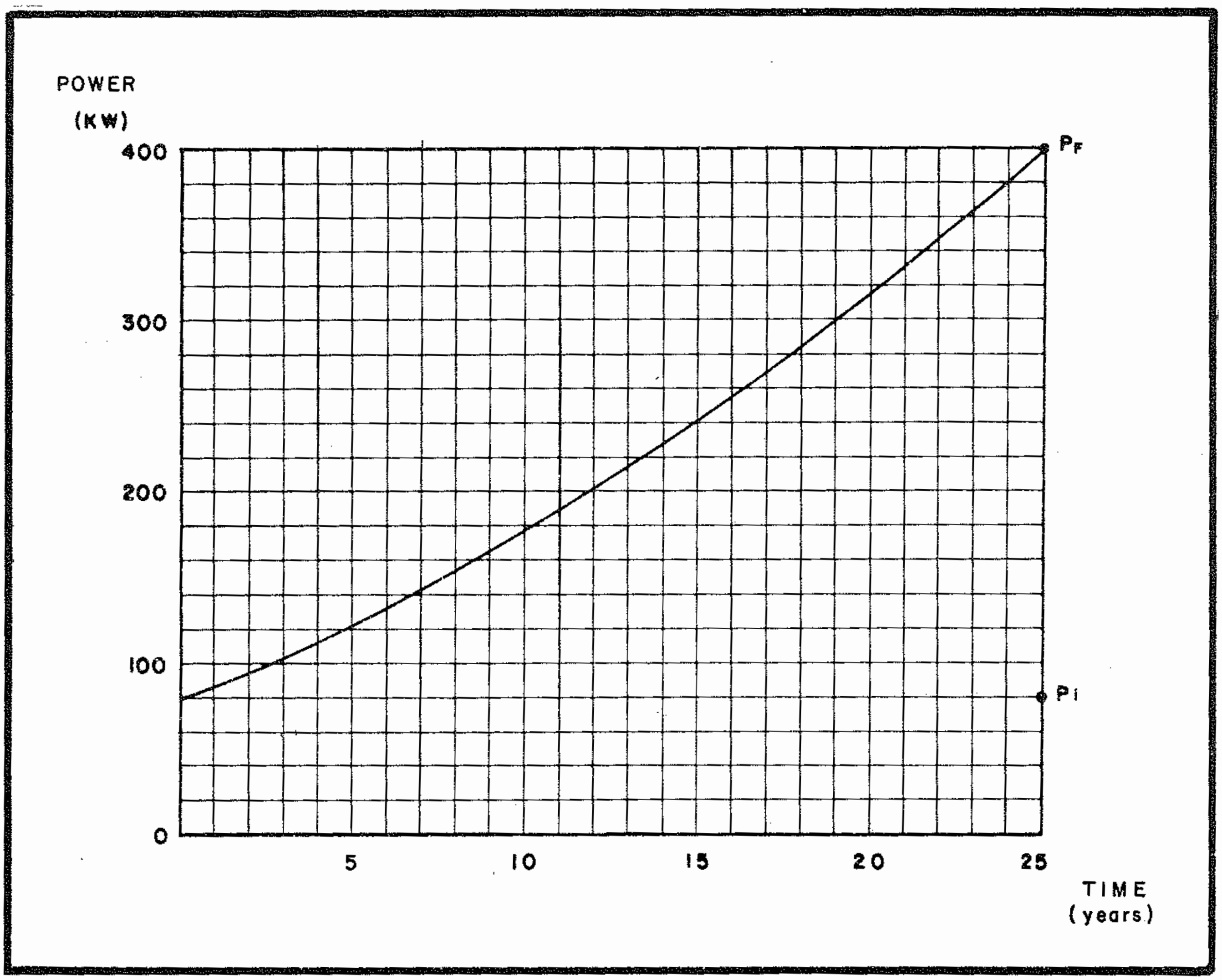


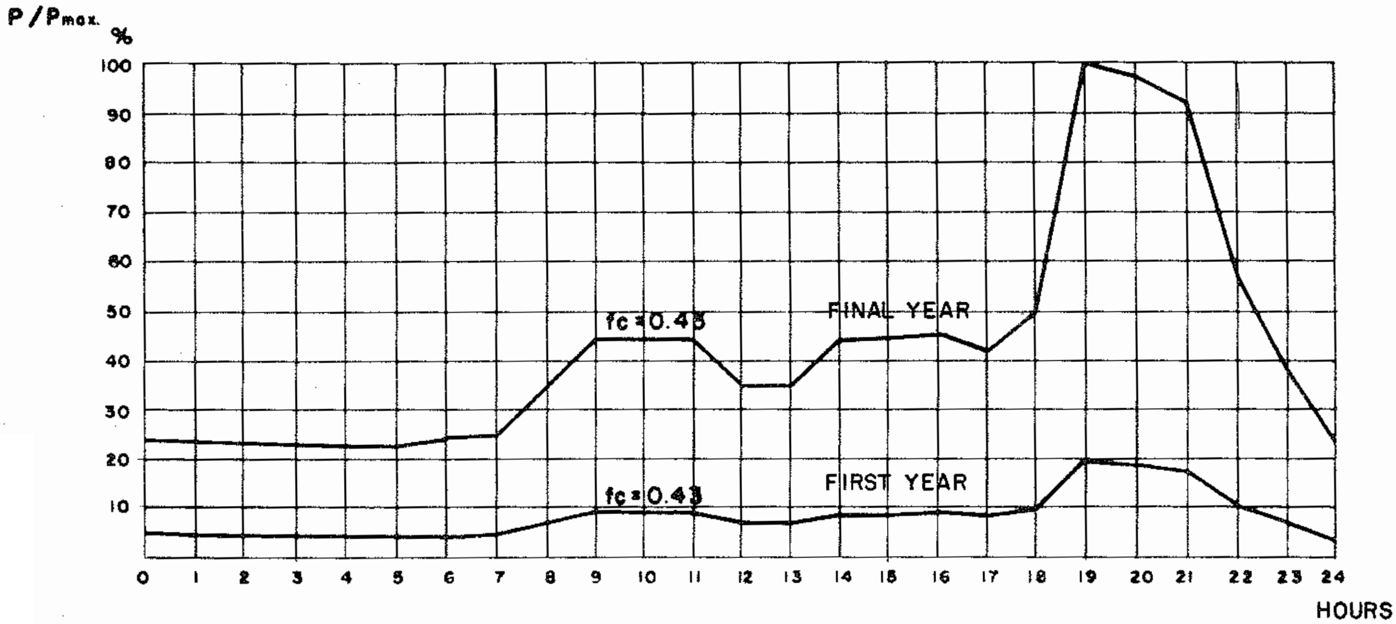
FIGURE 1A



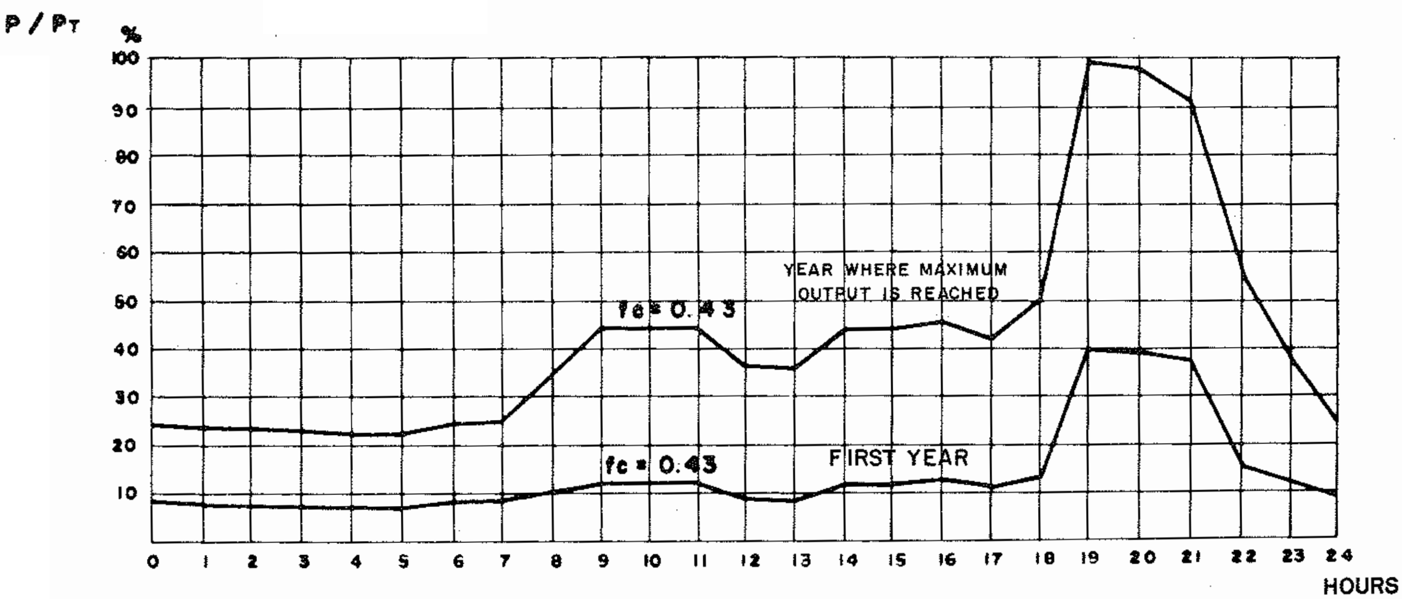
PROJECTION OF DEMAND

FIGURE 2A

LOAD DIAGRAM

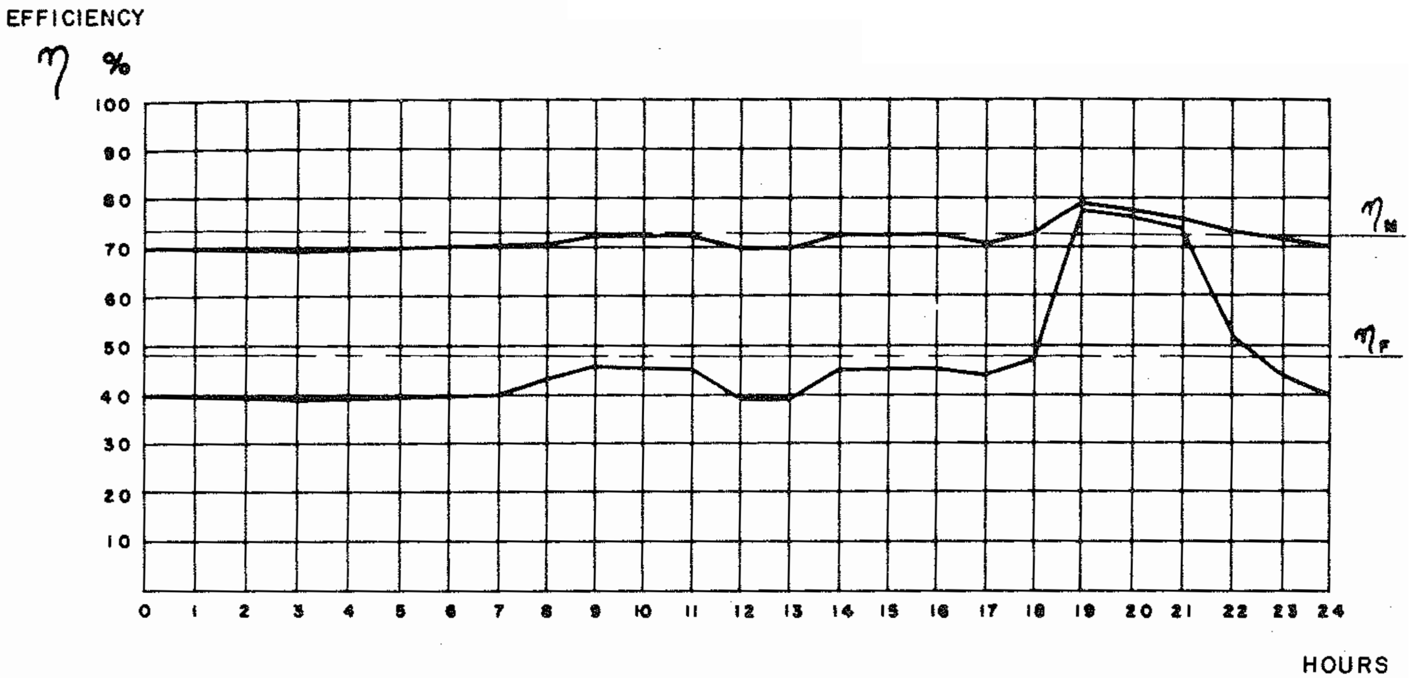


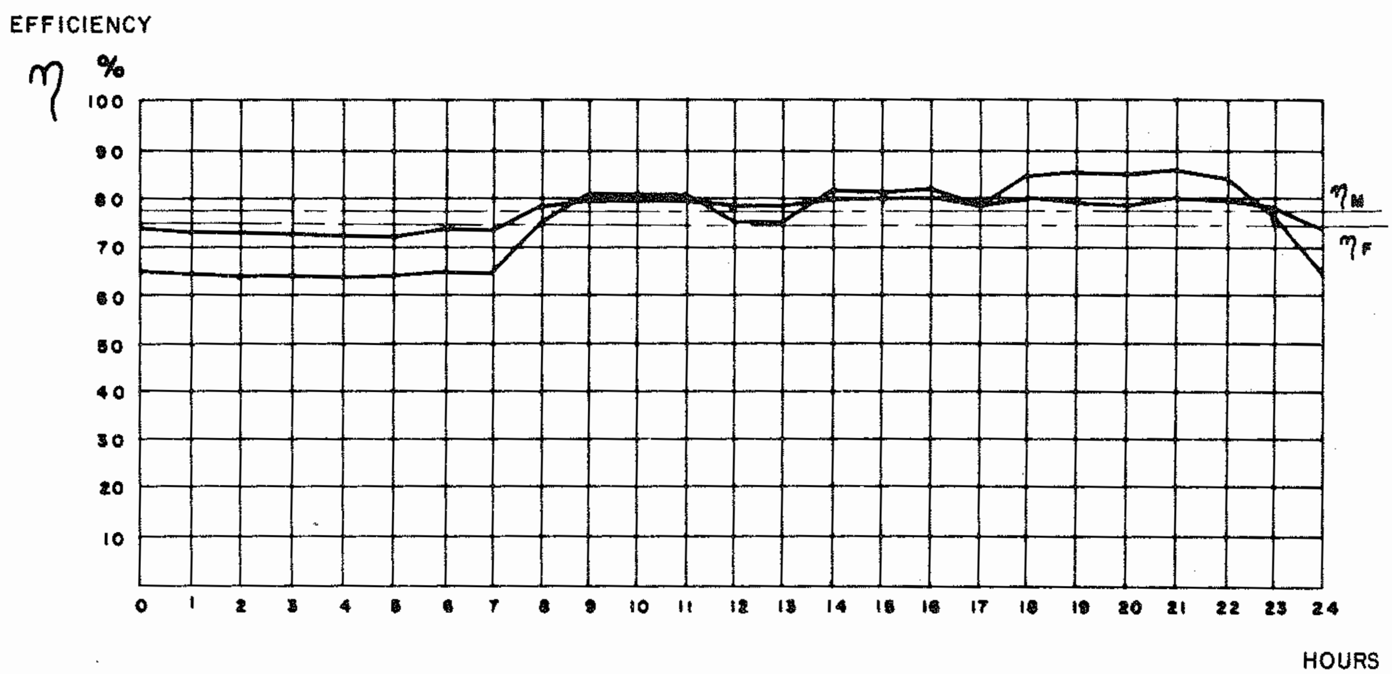
VARIATION IN DAILY LOADS AT THE POWER STATION AS A FUNCTION OF ITS MAXIMUM CAPACITY



VARIATION IN DAILY LOADS IN THE FIRST UIT AS A FUNCTION OF ITS MAXIMUM CAPACITY

FIGURE 3A





EFFICIENCY OF MICHELL-BANKI AND FRANCIS TURBINES AS A FUNCTION OF THE LOAD DIAGRAM FOR THE FIRST UNIT

FIGURE 4A

Substituting the value for the head, $H = 40$ meters, a turbine efficiency of 78 o/o at full load, and the brake power, a maximum flow of 0.740 m^3/s was obtained.

The optimum number of revolutions with which the turbine should turn was determined by applying the formula:

$$N = \frac{39.85\ H^{1/2}}{De}$$

In this case, a runner diameter of 0.3 meters was assumed; and by replacing the values in the formula, an optimum number of revolutions of 840 RPM was obtained.

To verify the applicability of the Michell–Banki turbine, the specific number of revolutions was calculated:

$$N_q = N \frac{Q^{1/2}}{H^{3/4}}$$

yielding the figure $N_q = 26$; this value falls within the range of application of the Michell–Banki turbine.

With the assumed runner diameter value, the shape of the runner and the injector were determined, for which purpose it was necessary to determine the width of the injector with the practical formula:

$$B = \frac{0.96\ Q}{De \sqrt{H}}$$

yielding an injector width of 0.375 meters.

Considering a two-compartment injector for the turbine design, in which one was half as wide as the other, it was obtained that the width of the one would be 125 mm. and that of the other 250 mm.

In order to verify if the guide vane could withstand the stress presented in the wider compartment, the maximum torque required to regulate the flow was calculated using the following formula:

$$T_{max} = 31\ D_e\ Q \sqrt{H}$$

In this case, the flow Q corresponded to the flow through the wider compartment:

$$Q = \frac{2}{3}\ (0.740) = 0.493\ m^3/s$$

Thus, the maximum starting torque would be equal to 29 km-m.

By calculating the maximum starting torque of the guide vane, it was possible to verify the mechanical stress to which the shaft of the guide vane would be subject. For this, the material was taken to be bronze to aluminum, with a flow stress of $S_y = 30$ kg/mm² (according to Table 3); and the diameter of the shaft was taken as 0.038, the maximum permitted for a runner diameter of 0.3 meters.

Then the stress presented in the shaft S was determined as follows:

$$S = \frac{16\ T}{\pi\ d_i^3}$$

Replacing the data, we had:

$$S = \frac{(16)\ (29)}{\pi\ (0.038)^3} = 2.7 \times 10\ \ kg/m^2$$

Thus, S = 2.7 kg/mm^2.

The design stress of the material would be:

$$Sd = 0.2\ \ Sy = 6\ kg/mm^2 > S, \text{which is correct.}$$

With this verification, the runner diameter and the width of the injector compartments were defined. The rest of the dimensions could be obtained from Table 2 and from Figures 8A and 8B of the manual.

With the aid of Drawing TM-01-03 of the detailed design of the turbine shown in Appendix 2, the runner width was determined, yielding a value of Br = 0.47 meters.

For the runner, a blade thickness of 0.003 meters was assumed, to determine the maximum stress to which it would be subject, using the following formula:

$$\sigma\ max = \frac{F\ Br\ c}{12\ Igx}$$

where:

$$F = 46.5\ \ Q\sqrt{H}$$

In this case, Q was the maximum flow through the turbine. The force F then had a value of 217.6 kg.

From Table 4, the values could be obtained for the center of rotation and for the moment of inertia Igx, with a maximum stress of 2.245 kg/cm^2, i.e., 22.45 kg/mm^2, which is greater than the flow stress of the material (21 kg/mm^2). In this case, it would be useful to place an intermediate disk in the runner in order to reduce this stress, thereby dividing the runner width into two compartments, with the maximum width of the larger compartment being 0.270 meters, so that in this case the maximum stress would be 12.91 kg/mm^2.

The design stress of the material, considering bending, would be:

$$Sd = 0.66\ Sy = (0.66)\ (21) = 13.86\ kg\ /mm^2$$

which indicates that the material can withstand the maximum blade stress.

The next step consisted of calculating the diameter of the main shaft of the turbine, by applying the ASME formula:

$$d^3 = \frac{16}{\pi\ S_d} \sqrt{(K_m\ M_{max})^2 + (K_t\ T_{max})^2}$$

where:

$$M_{max} = \sqrt{M_x^2 + M_y^2}$$

with:

$$M_x = \frac{F_r \cdot a}{2}$$

$$M_y = \frac{P_r \cdot a}{2}$$

Pr being the estimated runner weight (20 kg) and
Fr being the tangential runner stress calculated by:

$$F = \frac{1948 \ P_T}{N \ D_e}$$

and yielding $F = 3063$ kg .

In the case of the turbine of this example, it was determined that "a" had a value of 0.15 meters in the detailed design.

Replacing data, we obtained a moment M_{max} of 230 kg-m

The maximum torque was determined by the following formula:

$$T_{max} = \frac{974 \ P_T}{N}$$

and yielded $T_{max} = 459$ kg-m

With this value, and assuming a material with a flow stress Sy equal to 21 kg/mm², the shaft diameter was obtained. The design stress would be:

$$S = 0.2 \ Sy = 4.2 \ \text{kg/mm}^2$$

The shaft diameter obtained was approximately 0.089 meters, and a diameter of 90 mm. was chosen for the bearings section.

The shaft was scaled considering the runner assembly and the fact that the part of the shaft that goes through the runner had a dimension smaller than:

$$d = 0.328 \ De = 0.098 \text{ meters}$$

Checking the shaft in terms of critical speed, we obtained that:

$$N \, crit = \frac{29.88}{y^{1/2}}$$

and the value of "y" was determined with the following formula:

$$y = \frac{W a^2}{6EI} (3 \, 1_0 - 4a)$$

where: $W = \sqrt{Pr^2 + Fr^2} = 3063$ kgr

$E = 2.1 \times 10^{10}$ kg/m²

$$I = \frac{\pi d^4}{64} = \frac{\pi (0.090)^4}{64} = 3.2 \times 10^{-6} \ m^4$$

yielding $y = 6.098 \times 10^{-4}$ meters

This gave us a critical speed of approximately 1210 RPM, which it is impossible for this turbine to obtain, and thus showed that the shaft dimension was correct.

The next step was to select the bearings, for which purpose the dynamic base capacity was determined with the following formula:

$$C = (XF_r + Y F_a) \left| \frac{60 \text{ N } L_h}{10^6} \right|^p$$

where "X" was considered equal to the null axial force and the exponent "p" equal to 3/10, if considering roller bearings.

The rated duration of working hours was considered as 200,000 hours.

The radial force F was determined as 1535 kg., with which the dynamic base capacity of the roller had a value of 20,620 kg; catalogue rollers SKF No. 23220 CK could be used.

Finally, to conclude the mechanical design of the turbine, it was determined that the force that acts on the regulating cam had a value of 285 kg., so that the rigid ball bearings SKF No. 6205 were selected for the cam.

The cam shaft was calculated with the following formula:

$$d^3 = \frac{16 \text{ a F}}{\pi \text{ Sd}}$$

In this case, the detailed design yielded a = 0.1meters, with which the shaft diameter had a value of 0.033 meters, rounded off to 35 mm. for a shaft using material whose flow stress Sy = 21 kg/mm².

The stage subsequent to these calculations was to refine the detailed design of each component of the turbine on the basis of the data obtained.

The process of design and calculation of the Michell–Banki turbine could then be considered concluded.

In the event that one of the standardized turbines had been considered for selection, turbine TM-41B (Figure No. 15 of the manual) would have been chosen, the dimensions of which can be obtained from the tables attached to the detailed drawings of the turbines; and in this case, the regulating cams from drawings TM-06-07 and TM-06-08 would have been considered.

APPENDIX 2

DETAILED DRAWINGS AND TABLES OF DIMENSIONS FOR STANDARDIZED MICHELL–BANKI TURBINES

The aim of this appendix is to show the detailed drawings of each piece comprising the Michell–Banki turbine, along with the corresponding tables of dimensions for each standardized turbine, the range of application of which is shown in Table 15 of the manual.

It is necessary to note that the dimensions included in these tables are only a guide, and they can suffer variations in keeping with the criteria of the engineer or technician who is designing the turbine. Likewise, the detailed design can be modified on the basis of particular experiences and on the basis of the selection of a better alternative for manufacturing each turbine component.

These detailed drawings should be complemented by manufacturing drawings, which should indicate the manufacturing alternative selected, as well as the tolerance and finishing required for each part.

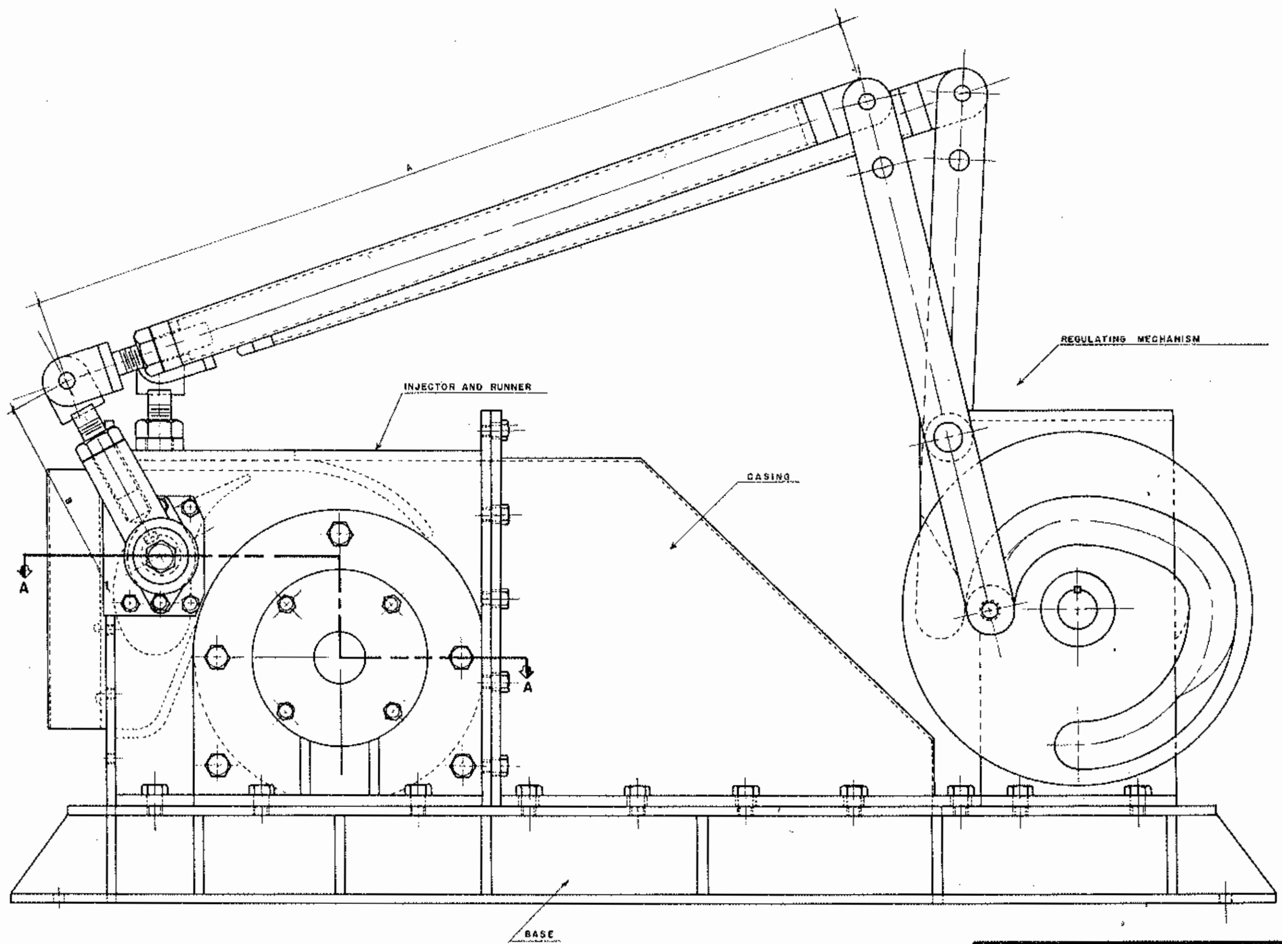
MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-01-01

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	868	868	868	868	868	1157	1157	1157
B	200	200	200	200	200	267	267	267



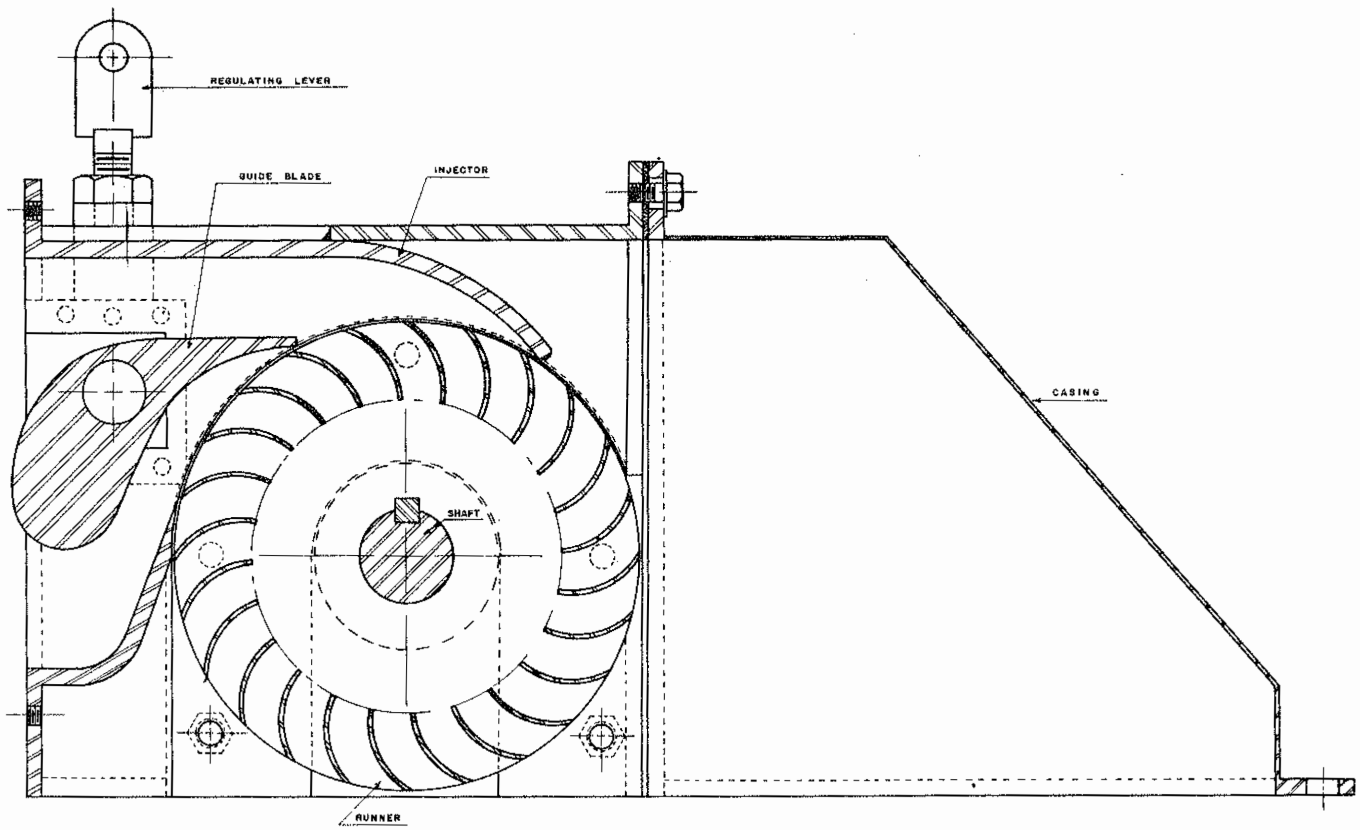
olade

DESIGN MANUAL ON

MICHELL-BANKI TURBINES

GENERAL VIEW

DRAWING : TM - 01 - 01	DESIGNED BY: CARLOS HERNANDEZ B.
	DRAWN BY: JULIO PAZMIÑO
SCALE :	DATE: APRIL ________ 82



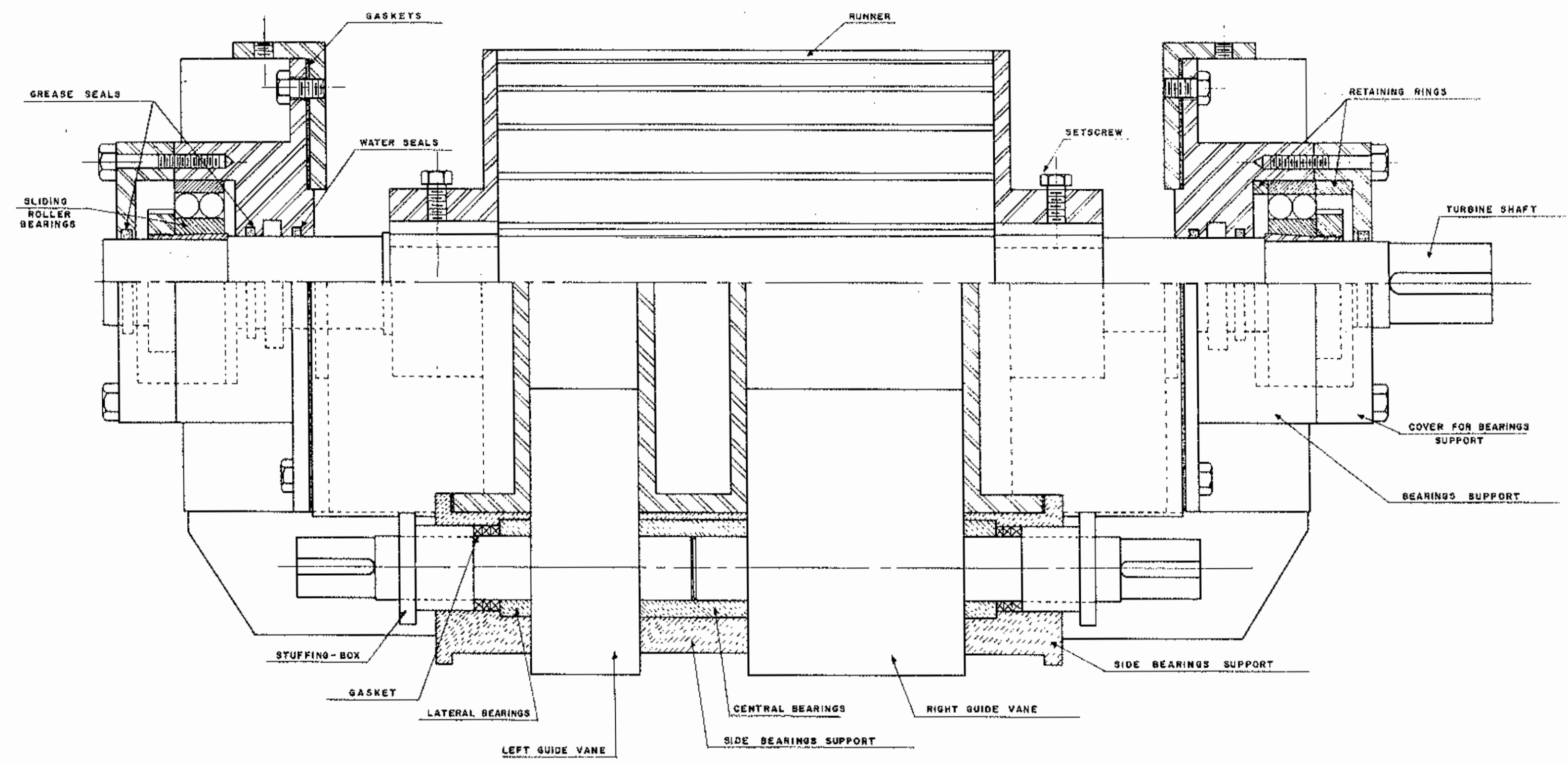
olade

DESIGN MANUAL ON

MICHELL-BANKI TURBINES

CROSS-SECTION

DRAWING : TM-01-02	DESIGNED BY: CARLOS HERNANDEZ B.
	DRAWN BY: JULIO PAZMIÑO
SCALE:	DATE: APRIL ______ 82



CROSS-SECTION A-A

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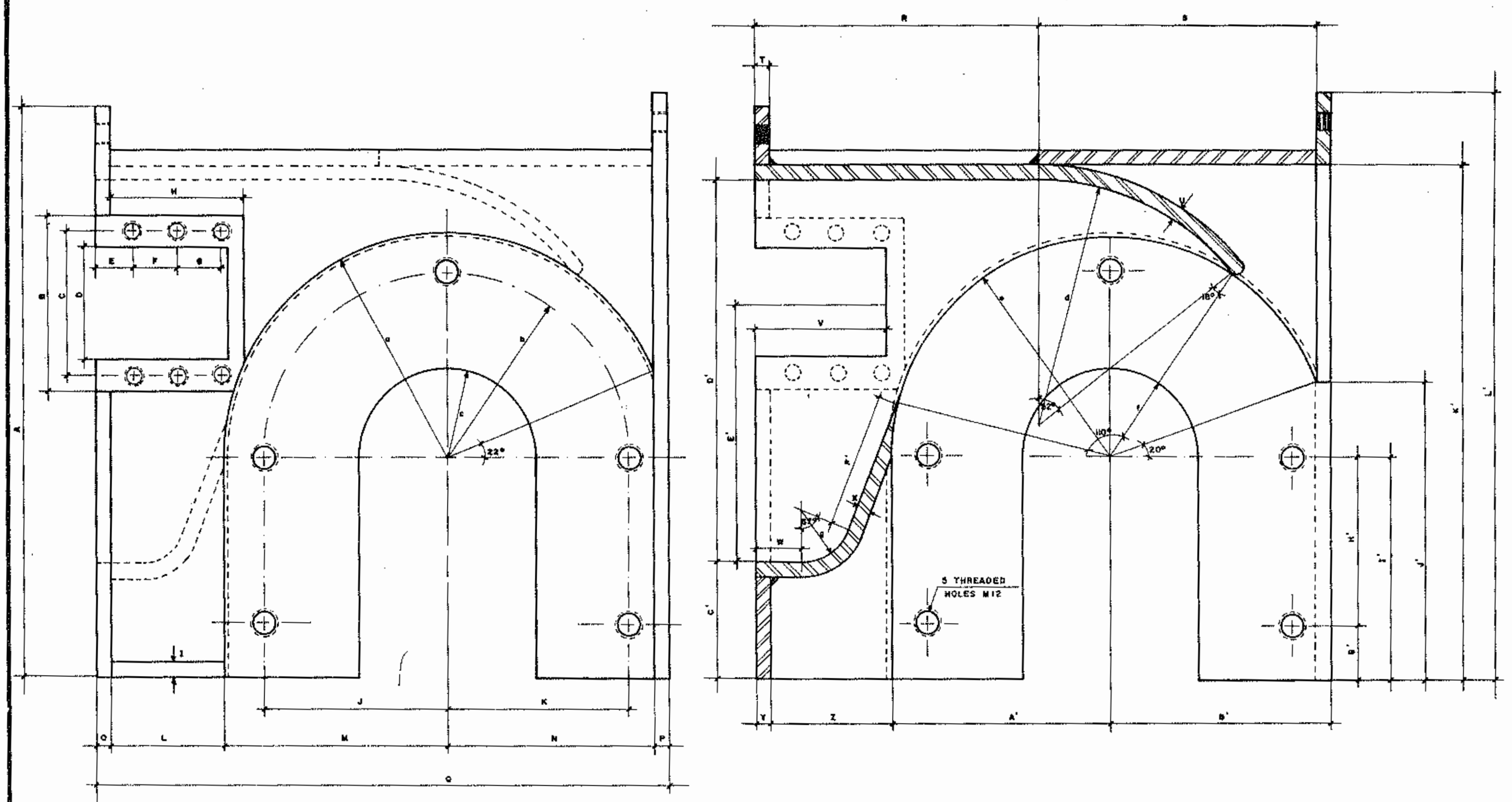
DESIGN MANUAL ON
MICHELL-BANKI TURBINES

INJECTOR AND RUNNER ASSEMBLY

DRAWING : TM-01-03	DESIGNED BY: CARLOS HERNANDEZ B.
	DRAWN BY: JULIO PAZMIÑO
SCALE:	DATE: APRIL 82



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DESIGN MANUAL ON

MICHELL-BANKI TURBINES

INJECTOR

DRAWING :	TM - 02 - 01	DESIGNED BY : CARLOS HERNANDEZ B.
		DRAWN BY : JULIO PAZMIÑO
SCALE :		DATE : APRIL ________ 82

MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-02-01

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	390	390	390	390	390	520	520	520
B	120	120	120	120	120	160	160	160
C	100	100	100	100	100	133	133	133
D	76	76	76	76	76	101	101	101
E	25	25	25	25	25	33	33	33
F	30	30	30	30	30	40	40	40
G	30	30	30	30	30	40	40	40
H	92	92	92	92	92	123	123	123
I	10	10	10	10	10	12	12	12
J	125	125	125	135	135	167	167	167
K	125	125	125	135	135	167	167	167
L	79	79	79	79	79	105	105	105
M	154	154	154	154	154	205	205	205
N	142	142	142	142	142	189	189	189
O	10	10	10	10	10	12	12	12
P	10	10	10	10	10	12	12	12
Q	395	395	395	395	395	523	523	523
R	195	195	195	195	195	260	260	260
S	190	190	190	190	190	253	253	253
T	10	10	10	10	10	12	12	12
U	10	10	10	10	10	12	12	12
V	90	90	90	90	90	120	120	120
X	10	10	10	10	10	12	12	12
Y	10	10	10	10	10	12	12	12
Z	82	82	82	82	82	109	109	109
a	154	154	154	154	154	205	205	205
b	125	125	125	135	135	175	175	175
c	60	70	80	100	100	110	125	125

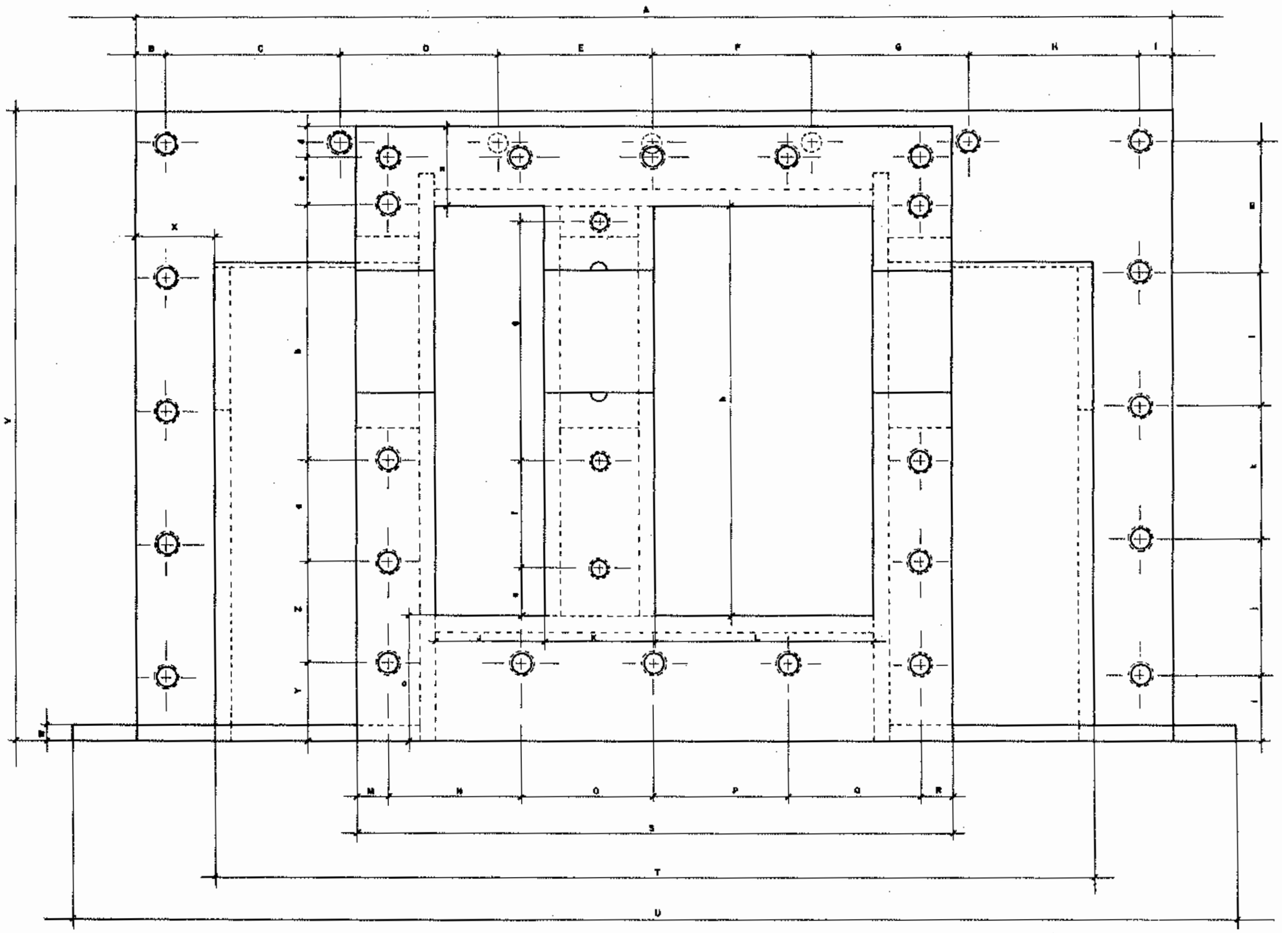
MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

(CONTINUATION) DRAWING Nº TM-02-01

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
d	168	168	168	168	168	224	224	224
e	151	151	151	151	151	201	201	201
f	60	70	80	100	100	110	125	125
g	28	28	28	28	28	37	37	37
A'	151	151	151	151	151	201	201	201
B'	152	152	152	152	152	203	203	203
C'	78	78	78	78	78	104	104	104
D'	262	262	262	262	262	349	349	349
E'	176	176	176	176	176	235	235	235
F'	102	102	102	102	102	136	136	136
G'	40	40	40	40	40	55	55	55
H'	110	110	110	110	110	145	145	145
I'	150	150	150	150	150	200	200	200
J'	203	203	203	203	203	271	271	271
K'	350	350	350	350	350	463	463	463
L'	400	400	400	400	400	533	533	533



DESIGN MANUAL ON
MICHELL-BANKI TURBINES

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VIEW OF TURBINE INTAKE

DRAWING : TM-02-02	DESIGNED BY : CARLOS HERNANDEZ B.
	DRAWN BY : JULIO PAZMIÑO
SCALE :	DATE : APRIL ______ 82

TABLE OF DIMENSIONS
(mm)

DRAWING Nº TM-02-02

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	430	490	630	740	840	900	1110	1310
B	20	20	20	20	20	25	25	25
C	x	x	x	x	x	x	x	x
D	x	x	x	x	x	x	x	x
E	x	x	x	x	x	x	x	x
F	x	x	x	x	x	x	x	x
G	x	x	x	x	x	x	x	x
H	x	x	x	x	x	x	x	x
I	20	20	20	20	20	25	25	25
J	-	-	70	70	150	170	250	350
K	70	70	70	70	70	90	90	90
L	60	90	70	140	150	170	250	350
M	20	20	20	20	20	25	25	25
N	x	x	x	x	x	x	x	x
O	x	x	x	x	x	x	x	x
P	x	x	x	x	x	x	x	x
Q	x	x	x	x	x	x	x	x
R	20	20	20	20	20	25	25	25
S	160	190	310	380	470	550	710	910
T	350	410	550	660	760	822	1032	1232
U	510	570	710	820	920	1042	1252	1452
V	400	400	400	400	400	533	533	533
W	10	10	10	10	10	12	12	12
X	40	40	40	40	40	38	38	38
Y	x	x	x	x	x	x	x	x
Z	x	x	x	x	x	x	x	x
a	x	x	x	x	x	x	x	x
b	x	x	x	x	x	x	x	x

x: space between bolts, to be defined for each case

MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

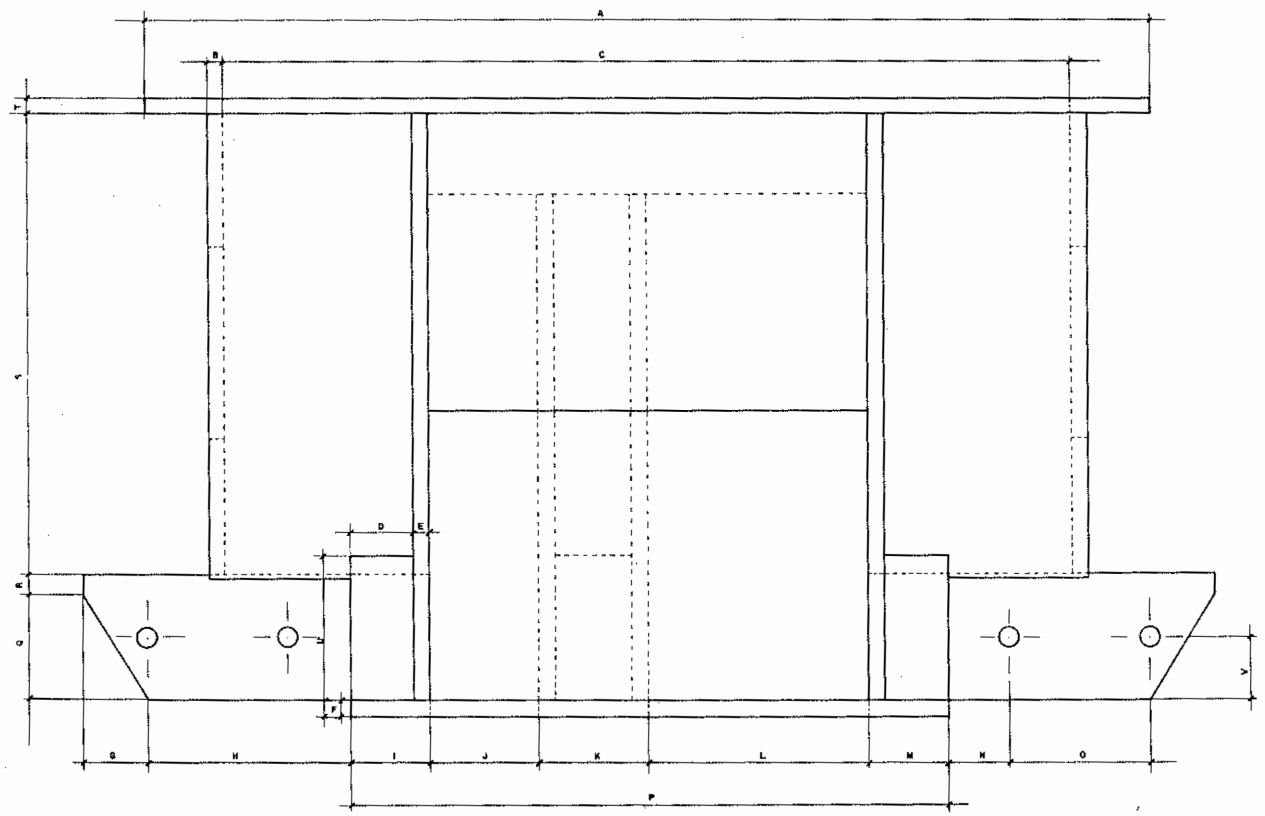
(mm)

(CONTINUATION)

DRAWING Nº TM-02-02

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
c	x	x	x	x	x	x	x	x
d	20	20	20	20	20	25	25	25
e	x	x	x	x	x	x	x	x
f	x	x	x	x	x	x	x	x
g	x	x	x	x	x	x	x	x
h	262	262	262	262	262	349	349	349
i	40	40	40	40	40	40	40	40
j	x	x	x	x	x	x	x	x
k	x	x	x	x	x	x	x	x
l	x	x	x	x	x	x	x	x
m	x	x	x	x	x	x	x	x
n	50	50	50	50	50	50	50	50
o	78	78	78	78	78	104	104	104

x: space between bolts, to be defined for each case



DESIGN MANUAL ON
MICHELL-BANKI TURBINES

VIEW OF INJECTOR FROM ABOVE

DRAWING :	TM - 02 - 03	DESIGNED BY :	CARLOS HERNANDEZ B.
		DRAWN BY :	JULIO PAZMIÑO
SCALE :		DATE :	APRIL ______ 82

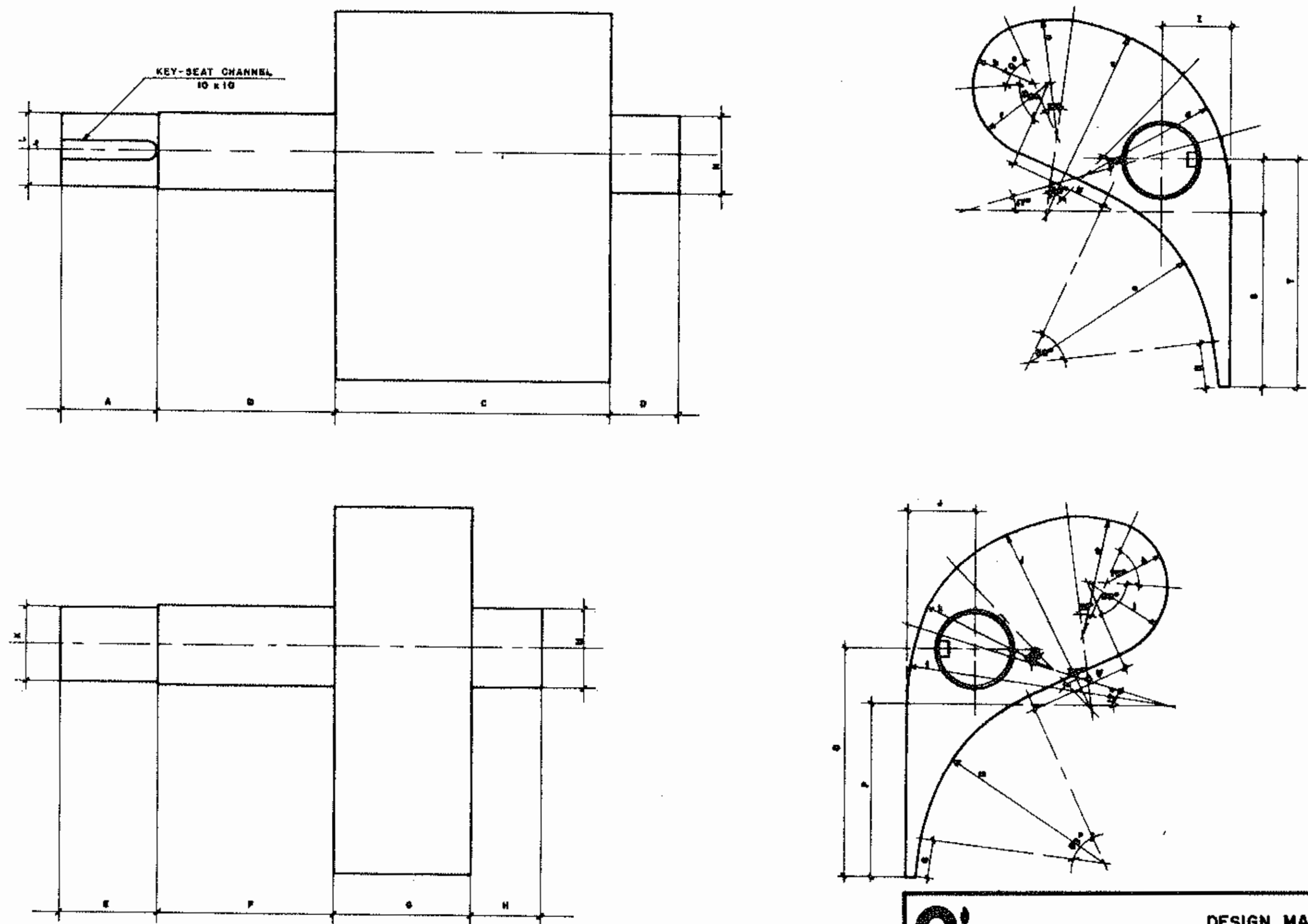
MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-02-03

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	430	490	630	740	840	900	1110	1310
B	10	10	10	10	10	12	12	12
C	330	390	530	640	740	800	1010	1210
D	40	40	40	40	40	55	55	55
E	10	10	10	10	10	10	10	10
F	10	10	10	10	10	12	12	12
G	40	40	40	40	40	60	60	60
H	135	150	160	180	185	186	211	211
I	50	50	50	50	50	65	65	65
J	-	-	70	70	150	170	250	350
K	-	-	70	70	70	90	90	90
L	60	90	70	140	150	170	250	350
M	50	50	50	50	50	65	65	65
N	40	40	40	40	40	65	65	65
O	90	90	90	90	90	120	120	120
P	160	190	310	380	470	550	710	910
Q	64	64	64	64	64	80	80	80
R	15	15	15	15	15	25	25	25
S	196	196	196	196	196	394	394	394
T	10	10	10	10	10	12	12	12
U	102	102	102	102	102	133	133	133
V	30	30	30	30	30	40	40	40



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DESIGN MANUAL ON
MICHELL-BANKI TURBINES

INJECTOR GUIDE VANE

DRAWING: TM-02-04	DESIGNED BY: CARLOS HERNANDEZ B.
	DRAWN BY: JULIO PAZMIÑO
SCALE:	DATE: APRIL ______ 82

MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-02-04

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	50	50	50	50	50	65	65	65
B	100	100	100	100	100	130	130	130
C	60	90	70	140	150	170	250	350
D	35	35	35	35	35	45	45	45
E	-	-	50	50	50	65	65	65
F	-	-	100	100	100	130	130	130
G	-	-	70	70	150	170	250	350
H	-	-	35	35	35	45	45	45
I	35	35	35	35	35	47	47	47
J	35	35	35	35	35	47	47	47
K	38	38	38	38	38	48	48	48
L	38	38	38	38	38	48	48	48
M	40	40	40	40	40	50	50	50
N	40	40	40	40	40	50	50	50
O	116	116	116	116	116	155	155	155
P	98	98	98	98	98	131	131	131
Q	20	20	20	20	20	27	27	27
R	20	20	20	20	20	27	27	27
S	98	98	98	98	98	131	131	131
T	116	116	116	116	116	155	155	155
U	51	51	51	51	51	68	68	68
V	51	51	51	51	51	68	68	68
a	60	60	60	60	60	80	80	80
b	31	31	31	31	31	41	41	41
c	100	100	100	100	100	133	133	133
d	70	70	70	70	70	93	93	93
e	94	94	94	94	94	125	125	125

MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

(CONTINUATION)

DRAWING Nº TM-02-04

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
f	39	39	39	39	39	52	52	52
g	60	60	60	60	60	80	80	80
h	31	31	31	31	31	41	41	41
i	100	100	100	100	100	133	133	133
j	39	39	39	39	39	52	52	52
k	70	70	70	70	70	93	93	93
l	133	133	133	133	133	177	177	177
m	94	94	94	94	94	125	125	125

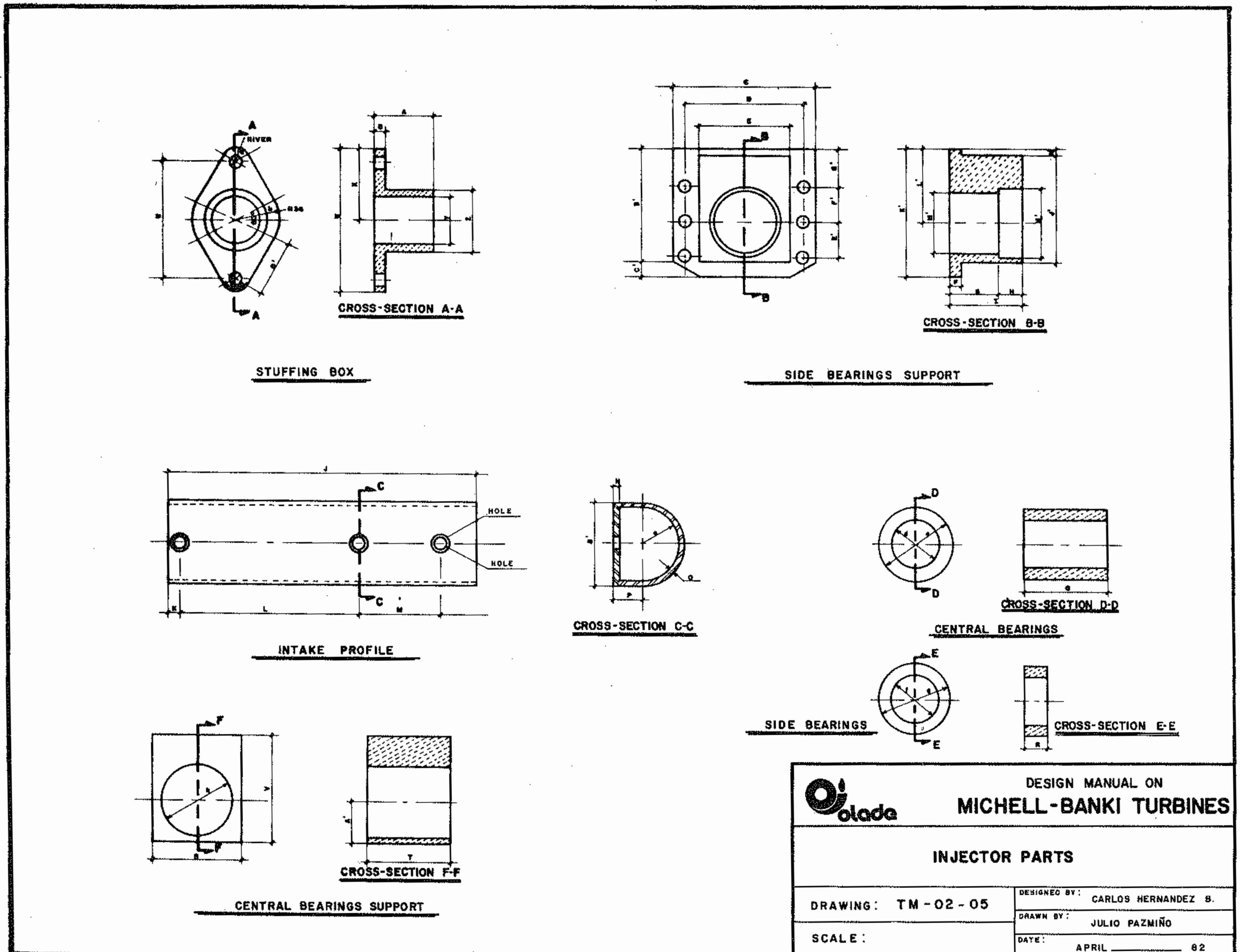

STUFFING BOX
CROSS-SECTION A-A
SIDE BEARINGS SUPPORT
CROSS-SECTION B-B
INTAKE PROFILE
HOLE
HOLE
CROSS-SECTION C-C
CENTRAL BEARINGS
CROSS-SECTION D-D
SIDE BEARINGS
CROSS-SECTION E-E
CENTRAL BEARINGS SUPPORT
CROSS-SECTION F-F
olade
DESIGN MANUAL ON
MICHELL-BANKI TURBINES
INJECTOR PARTS
DRAWING: TM-02-05
SCALE:
DESIGNED BY: CARLOS HERNANDEZ B.
DRAWN BY: JULIO PAZMIÑO
DATE: APRIL 82

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-02-05

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	60	60	60	60	60	80	80	80
B	10	10	10	10	10	12	12	12
C	120	120	120	120	120	160	160	160
D	100	100	100	100	100	130	130	130
E	76	76	76	76	76	101	101	101
F	10	10	10	10	10	12	12	12
G	42	42	42	42	42	54	54	54
H	20	20	20	20	20	20	20	20
I	62	62	62	62	62	74	74	74
J	261	261	261	261	261	348	348	348
K	x	x	x	x	x	x	x	x
L	x	x	x	x	x	x	x	x
M	x	x	x	x	x	x	x	x
N	6	6	6	6	6	6	6	6
O	3	3	3	3	3	3	3	3
P	25	25	25	25	25	30	30	30
Q	70.4	70.4	70.4	70.4	70.4	90.4	90.4	90.4
R	20.2	20.2	20.2	20.2	20.2	20.2	20.2	20.2
S	76	76	76	76	76	101	101	101
T	70	70	70	70	70	90	90	90
U	100	100	100	100	100	130	130	130
V	90	90	90	90	90	120	120	120
W	120	120	120	120	120	160	160	160
X	60	60	60	60	60	80	80	80
Y	40.2	40.2	40.2	40.2	40.2	50.2	50.2	50.2
Z	51.8	51.8	51.8	51.8	51.8	69.8	69.8	69.8
A'	35	35	35	35	35	47	47	47
B'	70	70	70	70	70	90	90	90

x: space between bolts, to be defined for each case

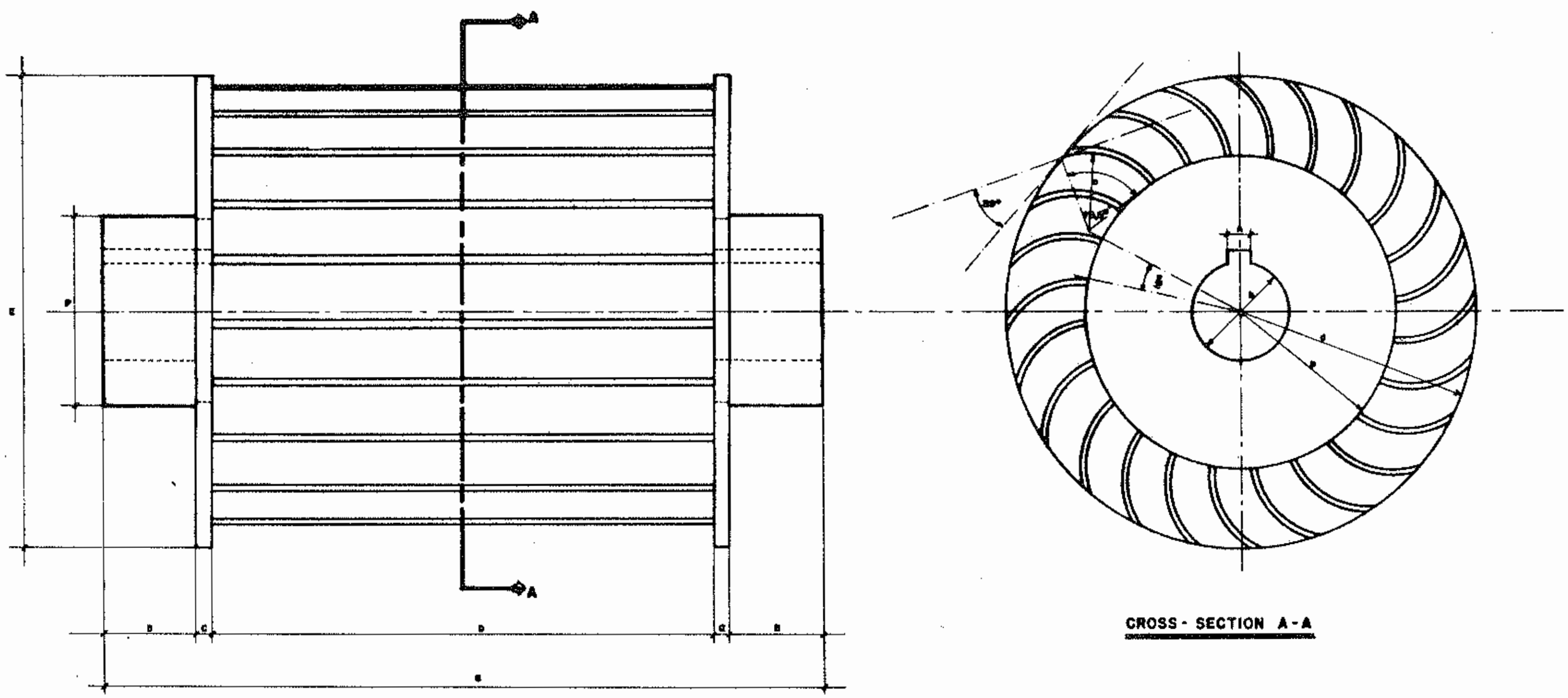
TABLE OF DIMENSIONS
(mm)

(CONTINUATION)

DRAWING Nº TM-02-05

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
C'	12	12	12	12	12	16	16	16
D'	96	96	96	96	96	126	126	126
E'	30	30	30	30	30	40	40	40
F'	30	30	30	30	30	40	40	40
G'	31	31	31	31	31	41	41	41
H'	60	60	60	60	60	80	80	80
J'	90	90	90	90	90	120	120	120
K'	108	108	108	108	108	142	142	142
L'	61	61	61	61	61	79	79	79
M'	52	52	52	52	52	71	71	71
N'	6	6	6	6	6	6	6	6
O'	45	45	45	45	45	60	60	60
a	10	10	10	10	10	15	15	15
b	36	36	36	36	36	48	48	48
c	32	32	32	32	32	42	42	42
d	40	40	40	40	40	50	50	50
e	60	60	60	60	60	80	80	80
f	40	40	40	40	40	50	50	50
g	60	60	60	60	60	80	80	80
h	60	60	60	60	60	80	80	80

x: space between bolts, to be defined for each case



CROSS-SECTION A-A

DESIGN MANUAL ON

MICHELL-BANKI TURBINES

olade

RUNNER

DRAWING : TM - 03 - 01	DESIGNED BY: CARLOS HERNANDEZ B.
	DRAWN BY: JULIO PAZMIÑO
SCALE :	DATE: APRIL ______ 82

olade

MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-03-01

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	16	18	20	24	24	24	30	30
B	70	85	95	115	120	120	145	145
C	10	10	10	10	10	10	10	10
D	90	120	240	310	400	460	620	820
E	300	300	300	300	300	400	400	400
F	120	140	160	190	200	200	240	240
G	250	310	450	560	660	720	930	1130
a	49	49	49	49	49	65	65	65
b	60	70	80	100	110	110	125	125
c	200	200	200	200	200	267	267	267
d	150	150	150	150	150	200	200	200

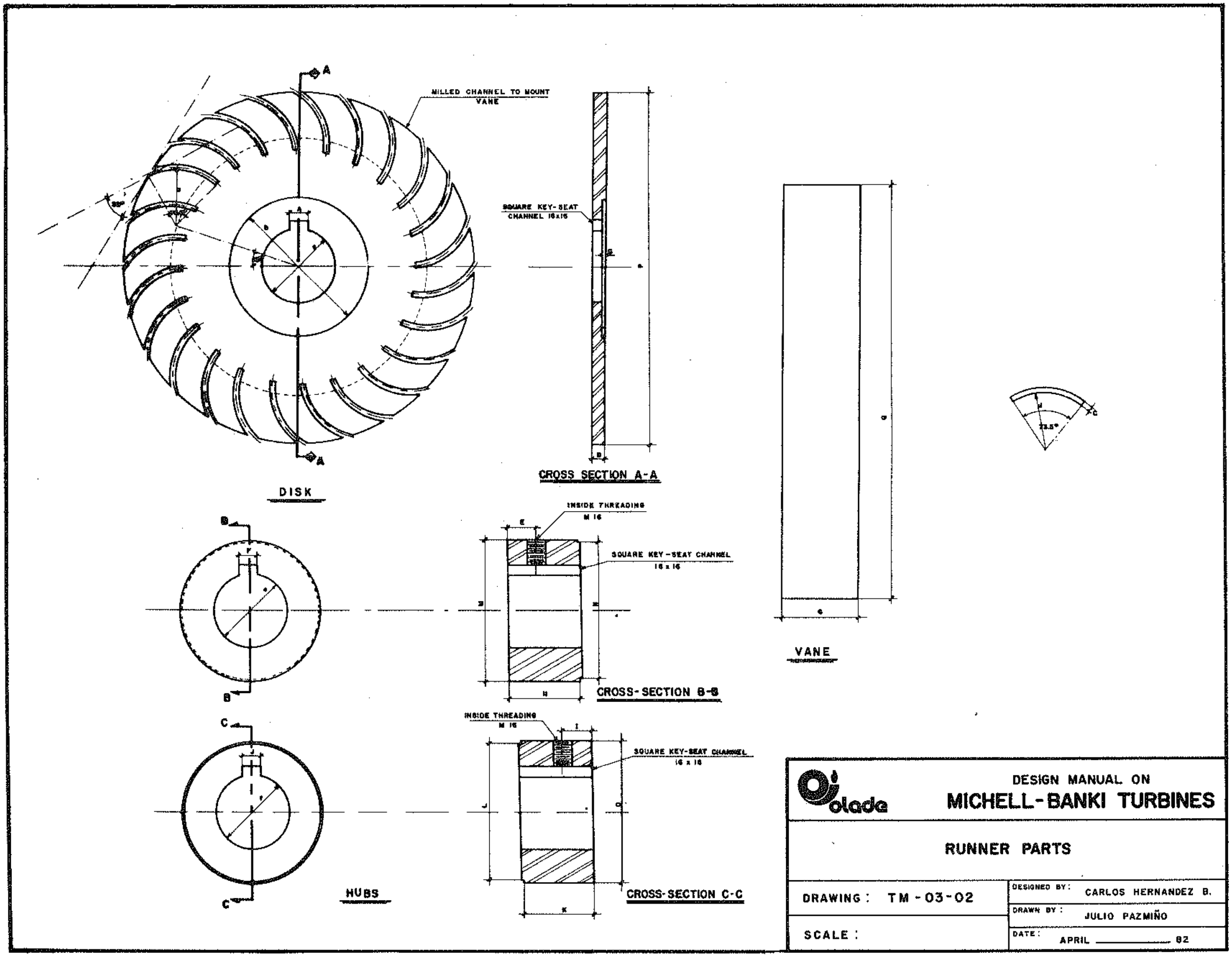
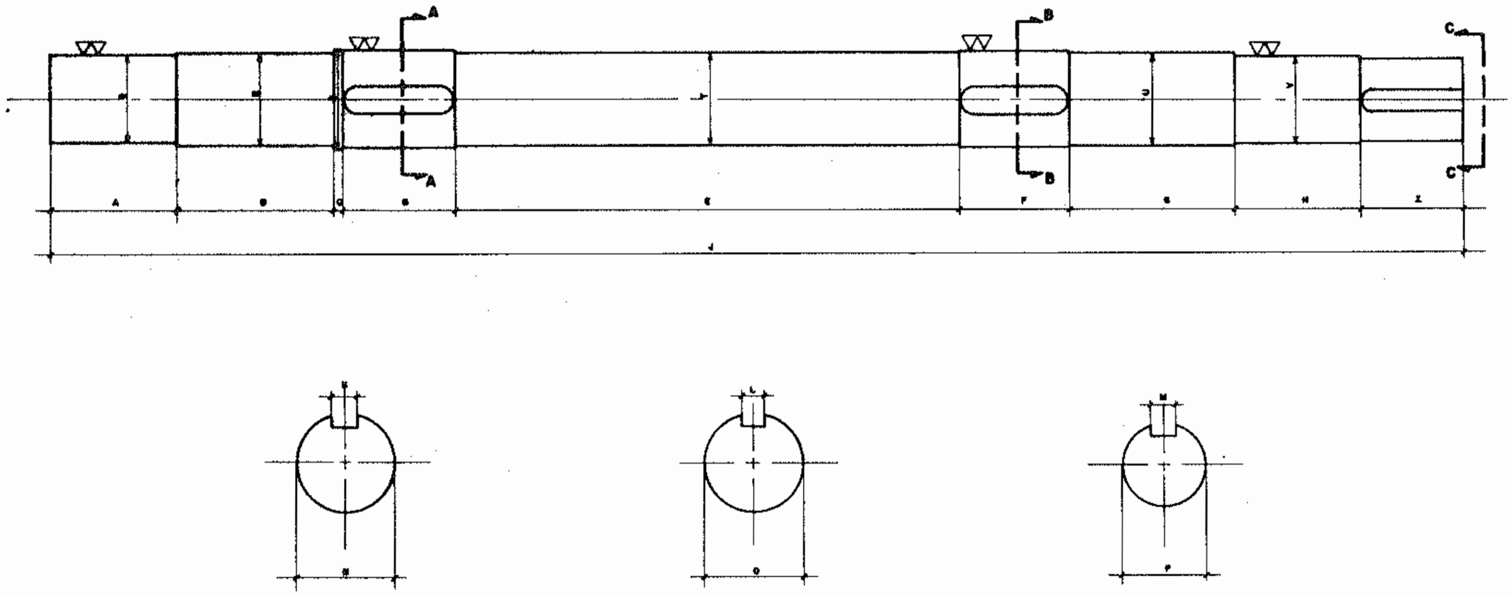

MILLED CHANNEL TO MOUNT VANE
SQUARE KEY-SEAT CHANNEL 16x16
CROSS SECTION A-A
DISK
INSIDE THREADING M 16
SQUARE KEY-SEAT CHANNEL 16 x 16
CROSS-SECTION B-B
VANE
INSIDE THREADING M 16
SQUARE KEY-SEAT CHANNEL 16 x 16
HUBS
CROSS-SECTION C-C
olade
DESIGN MANUAL ON
MICHELL-BANKI TURBINES
RUNNER PARTS
DRAWING : TM - 03-02
SCALE :
DESIGNED BY : CARLOS HERNANDEZ B.
DRAWN BY : JULIO PAZMIÑO
DATE : APRIL ________ 82

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-03-02

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	14	16	20	24	24	24	30	30
B	3	3	3	3	3	3	3	3
C	3	3	3	3	3	3	3	3
D	10	10	10	10	10	10	10	10
E	40	45	50	60	65	65	75	75
F	14	16	20	24	24	24	30	30
G	64	64	64	64	64	85	85	85
H	70	85	95	115	120	120	145	145
I	40	45	50	60	65	65	75	75
J	14	16	20	24	24	24	30	30
K	70	85	95	115	120	120	145	145
L	110	130	150	180	180	190	230	230
M	120	140	160	190	200	200	240	240
N	110	130	150	180	180	190	230	230
O	120	140	160	190	200	200	240	240
P	300	300	300	300	300	400	400	400
Q	110	140	260	330	420	470	640	840
a	50.5	50.5	50.5	50.5	50.5	67.3	67.3	67.3
b	120	140	160	190	200	200	240	240
c	60	70	80	100	110	110	125	125
d	49	49	49	49	49	65	65	65
e	60	70	80	100	110	110	125	125
f	59	69	79	99	109	109	124	124



CROSS-SECTION A-A

CROSS-SECTION B-B

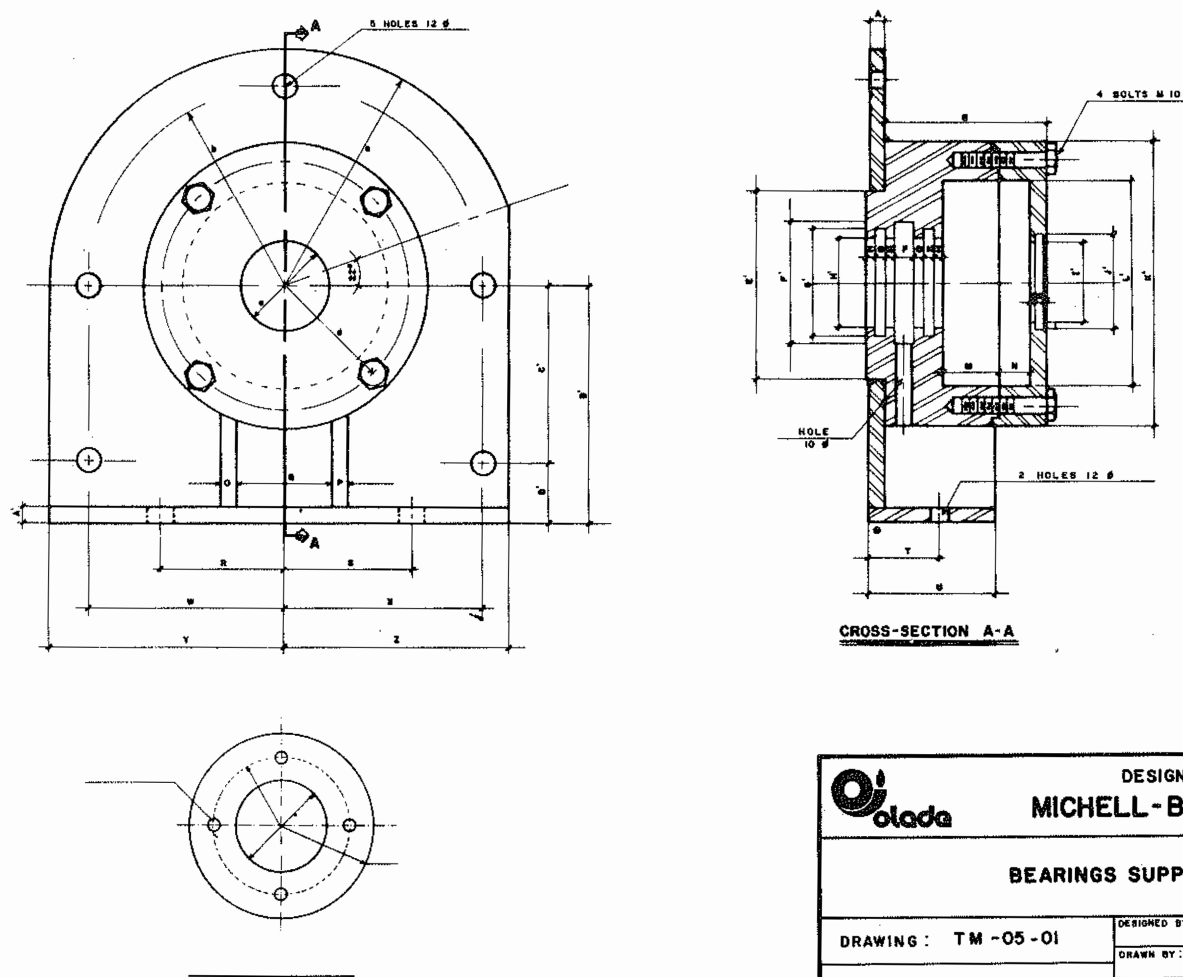
CROSS-SECTION C-C

olade	DESIGN MANUAL ON **MICHELL-BANKI TURBINES**
MAIN SHAFT	
DRAWING : TM-04-01	DESIGNED BY : CARLOS HERNANDEZ B.
	DRAWN BY : JULIO PAZMIÑO
SCALE :	DATE : APRIL _______ 82

TABLE OF DIMENSIONS
(mm)

DRAWING Nº TM-04-01

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	70	75	80	130	140	140	155	155
B	100	100	100	100	100	100	100	100
C	5	5	5	5	5	5	5	5
D	80	95	105	125	130	130	155	155
E	90	120	240	310	400	460	620	820
F	80	95	105	125	130	130	155	155
G	105	105	105	105	105	105	105	105
H	70	75	80	130	140	140	155	155
I	57	69	80	105	117	117	135	135
J	657	739	900	1135	1267	1327	1585	1785
K	14	16	20	24	24	24	30	30
L	14	16	20	24	24	24	30	30
M	12	14	18	22	24	24	30	30
N	60	70	80	100	110	110	125	125
O	59	69	79	99	109	109	124	124
P	47	57	67	87	97	97	112	112
Q	50	60	70	90	100	100	115	115
R	54	64	74	94	104	104	119	119
S	63.5	76.2	88.9	101.6	114.3	114.3	127	127
T	58	68	78	98	108	108	123	123
U	54	64	74	94	104	104	119	119
V	50	60	70	90	100	100	115	115



DESIGN MANUAL ON

MICHELL-BANKI TURBINES

olade

BEARINGS SUPPORT

DRAWING : TM -05 -01	DESIGNED BY: CARLOS HERNANDEZ B.
	DRAWN BY: JULIO PAZMIÑO
SCALE :	DATE: APRIL ______ 82

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-05-01

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	10	10	10	10	10	12	12	12
B	83	93	96	138	148	146	161	161
C	6	6	6	6	6	6	6	6
D	6	6	6	6	6	6	6	6
E	6	6	6	6	6	6	6	6
F	10	10	10	10	10	10	10	10
G	6	6	6	6	6	6	6	6
H	6	6	6	6	6	6	6	6
I	6	6	6	6	6	6	6	6
J	3	3	3	3	3	3	3	3
K	6	6	6	6	6	6	6	6
L	3	3	3	3	3	3	3	3
M	30	35	35	75	85	85	95	95
N	20	25	28	30	30	30	35	35
O	10	10	10	10	10	12	12	12
P	10	10	10	10	10	12	12	12
Q	60	60	60	60	60	80	80	80
R	80	80	80	80	80	110	110	110
S	80	80	80	80	80	110	110	110
T	45	45	45	45	45	60	60	60
U	80	80	80	80	80	110	110	110
W	125	125	125	135	135	175	175	175
X	125	125	125	135	135	175	175	175
Y	150	150	150	150	150	200	200	200
Z	142	142	142	142	142	188	188	188
a	150	150	150	150	150	200	200	200
b	125	125	125	125	125	175	175	175
c	55	65	75	95	105	105	120	120

MANUAL ON DESIGN OF MICHELL BANKI TURBINES

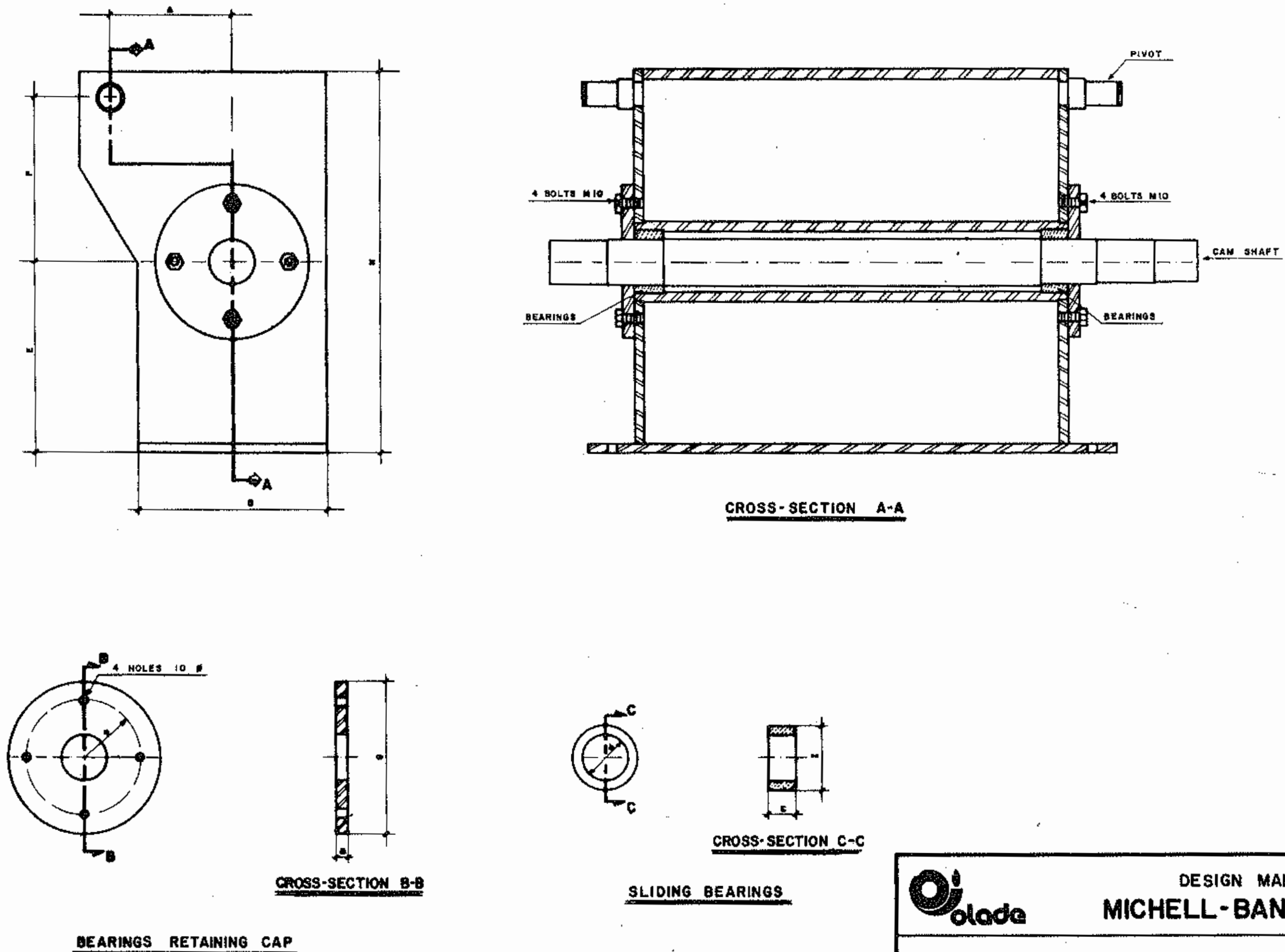
TABLE OF DIMENSIONS

(mm)

(CONTINUATION)

DRAWING Nº TM-05-01

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
d	130	150	170	210	220	230	260	260
A'	10	10	10	10	10	12	12	12
B'	40	40	40	40	40	55	55	55
C'	110	110	110	110	110	145	145	145
D'	150	150	150	150	150	200	200	200
E'	120	140	160	200	200	220	250	250
F'	75	85	95	115	125	125	140	140
G'	67	77	87	107	117	117	132	132
H'	55	65	75	95	105	105	120	120
I'	51	61	71	91	101	101	116	116
J'	63	73	83	103	113	113	128	128
K'	150	170	190	230	240	250	280	280
L'	100	120	140	180	200	200	230	230



olade

DESIGN MANUAL ON

MICHELL-BANKI TURBINES

ASSEMBLY OF REGULATING CAM STRUCTURE

DRAWING : TM - 06 - 01	DESIGNED BY: CARLOS HERNANDEZ B.
	DRAWN BY: JULIO PAZMIÑO
SCALE :	DATE: APRIL ___ 82

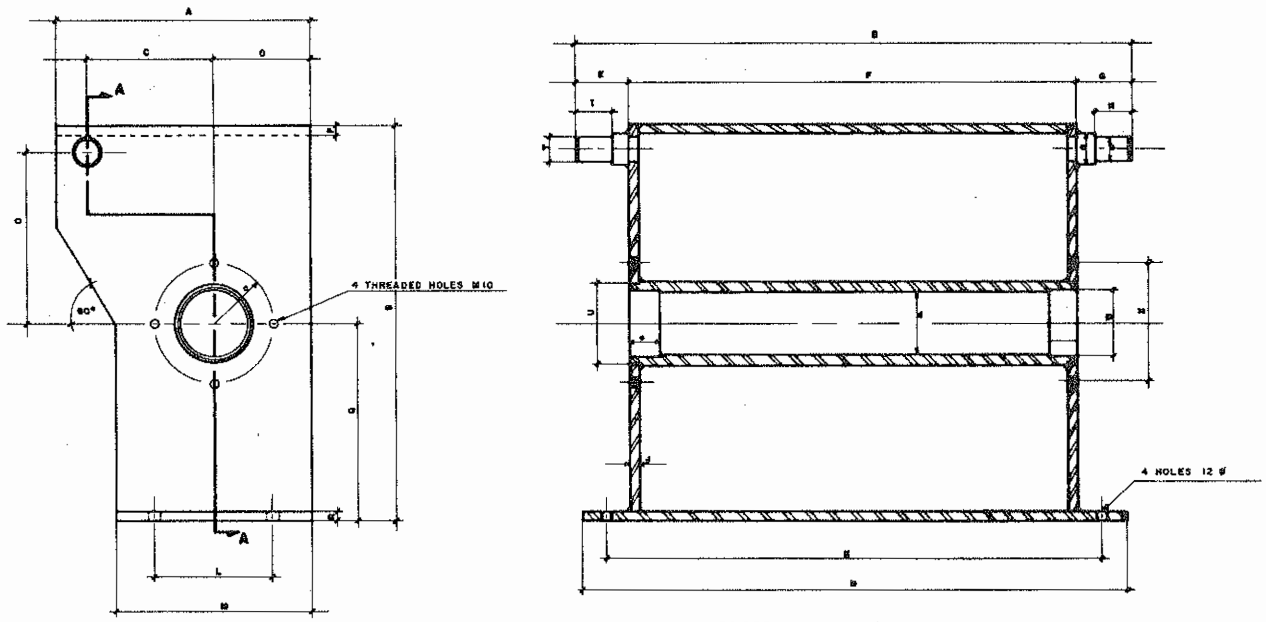
MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-06-01

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	-	-	130	130	130	174	174	174
B	-	-	200	200	200	267	267	267
C	-	-	40	40	40	60	60	60
D	-	-	10	10	10	12	12	12
E	-	-	200	200	200	267	267	267
F	-	-	175	175	175	233	233	233
G	-	-	200	200	200	220	230	230
H	-	-	400	400	400	533	533	533
I	-	-	60	65	70	79	84	84
a	-	-	75	75	75	85	90	90
b	-	-	40	45	50	55	60	60



CROSS-SECTION A-A

olade

DESIGN MANUAL ON
MICHELL-BANKI TURBINES

REGULATING CAM SUPPORT STRUCTURE

DRAWING: TM-06-02	DESIGNED BY: CARLOS HERNANDEZ B.
	DRAWN BY: JULIO PAZMIÑO
SCALE:	DATE: APRIL ______ 82

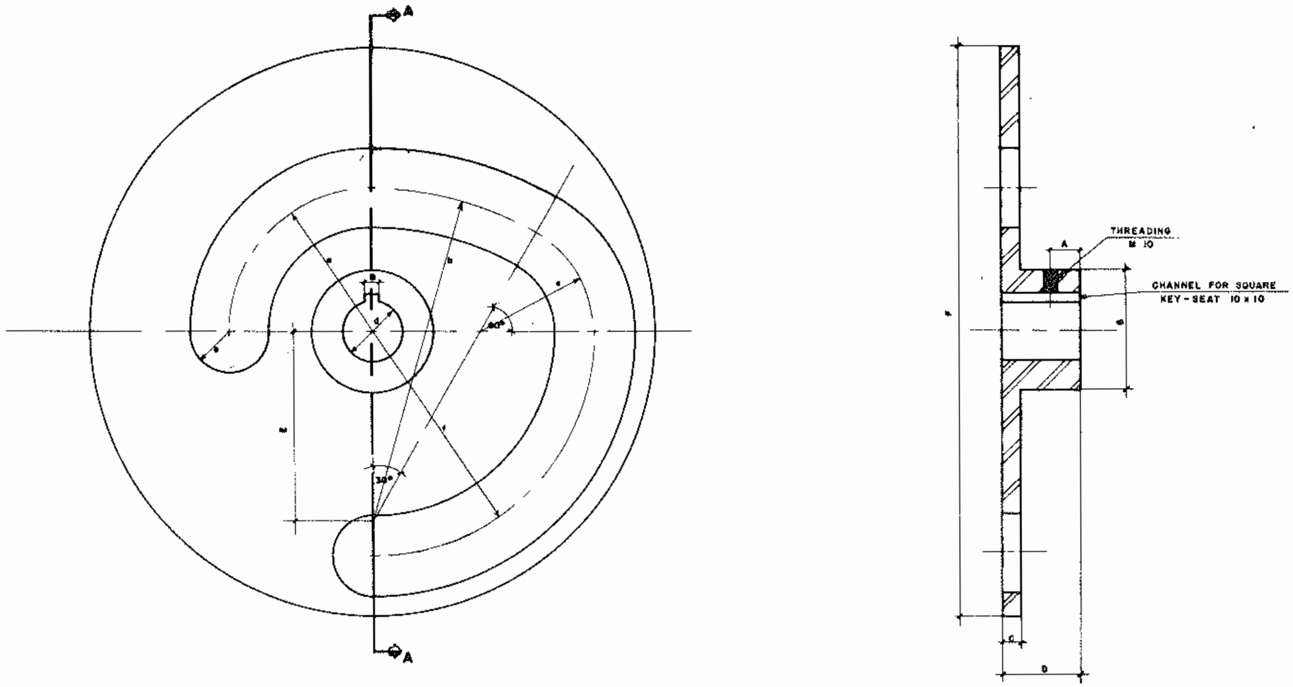
MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-06-02

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	-	-	260	260	260	347	347	347
B	-	-	513	583	673	816	976	1176
C	-	-	130	130	130	174	174	174
D	-	-	100	100	100	133	133	133
E	-	-	72	72	72	85	85	85
F	-	-	369	439	529	646	806	1006
G	-	-	72	72	72	85	85	85
H	-	-	47	47	47	55	55	55
I	-	-	47	47	47	55	55	55
J	-	-	10	10	10	12	12	12
K	-	-	419	489	579	696	856	1056
L	-	-	120	120	120	160	160	160
M	-	-	200	200	200	267	267	267
N	-	-	469	539	629	746	906	1106
O	-	-	175	175	175	233	233	233
P	-	-	10	10	10	10	10	10
Q	-	-	200	200	200	267	267	267
R	-	-	10	10	10	12	12	12
S	-	-	400	400	400	533	533	533
T	-	-	28	28	28	38	38	38
U	-	-	75	75	75	85	90	90
W	-	-	60	65	70	79	84	84
X	-	-	150	150	150	170	180	180
a	-	-	32	32	32	45	45	45
b	-	-	28	28	28	38	38	38
c	-	-	75	75	75	85	90	90
d	-	-	58	62	65	73	77	77
e	-	-	40	40	40	60	60	60



CROSS-SECTION A-A

DESIGN MANUAL ON
MICHELL-BANKI TURBINES

olade

RIGHT-HAND CAM OF REGULATING MECHANISM

DRAWING : TM-06-03	DESIGNED BY : CARLOS HERNANDEZ B.
	DRAWN BY : JULIO PAZMIÑO
SCALE :	DATE : APRIL ___ 82

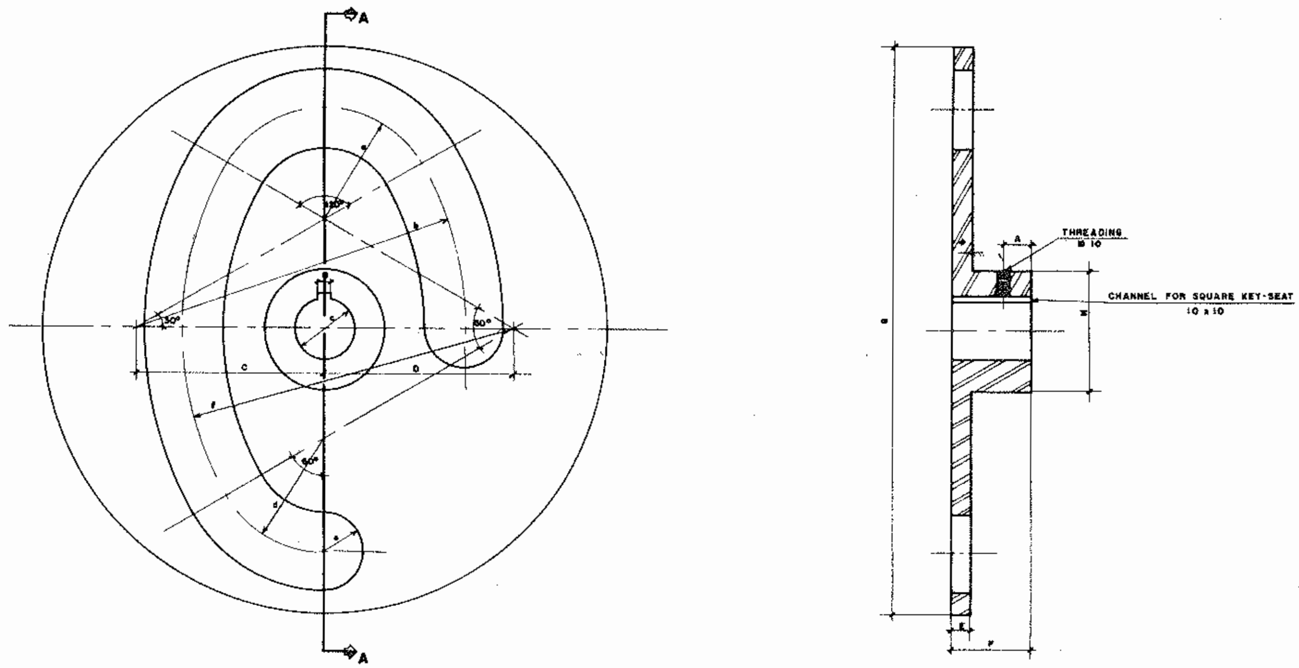
MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-06-03

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A				20				
B				10				
C				15				
D				55				
E				120				
F				360				
G				80				
a				90				
b				210				
c				72				
d				42				
e				26				
f				140				



CROSS-SECTION A-A

DESIGN MANUAL ON
MICHELL-BANKI TURBINES

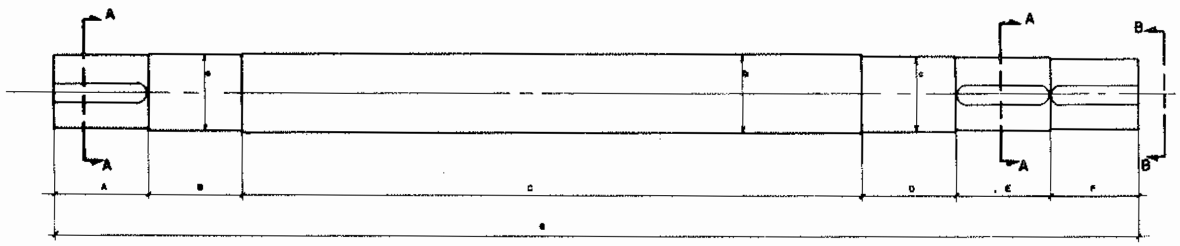
LEFT-HAND CAM OF REGULATING MECHANISM

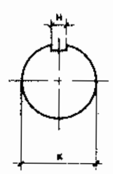
DRAWING : TM-06-04	DESIGNED BY: CARLOS HERNANDEZ B.
	DRAWN BY: JULIO PAZMIÑO
SCALE :	DATE: APRIL 82

TABLE OF DIMENSIONS
(mm)

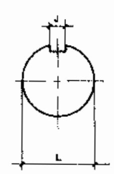
DRAWING Nº TM-06-04

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A				20				
B				10				
C				120				
D				120				
E				15				
F				55				
G				360				
H				80				
a				72				
b				210				
c				42				
d				72				
e				26				
f				210				





CROSS-SECTION A-A



CROSS-SECTION B-B

DESIGN MANUAL ON
MICHELL-BANKI TURBINES

olade

REGULATING CAM SHAFT

DRAWING: TM-06-05	DESIGNED BY: CARLOS HERNANDEZ B.
	DRAWN BY: JULIO PAZMIÑO
SCALE:	DATE: APRIL ______ 82

olade

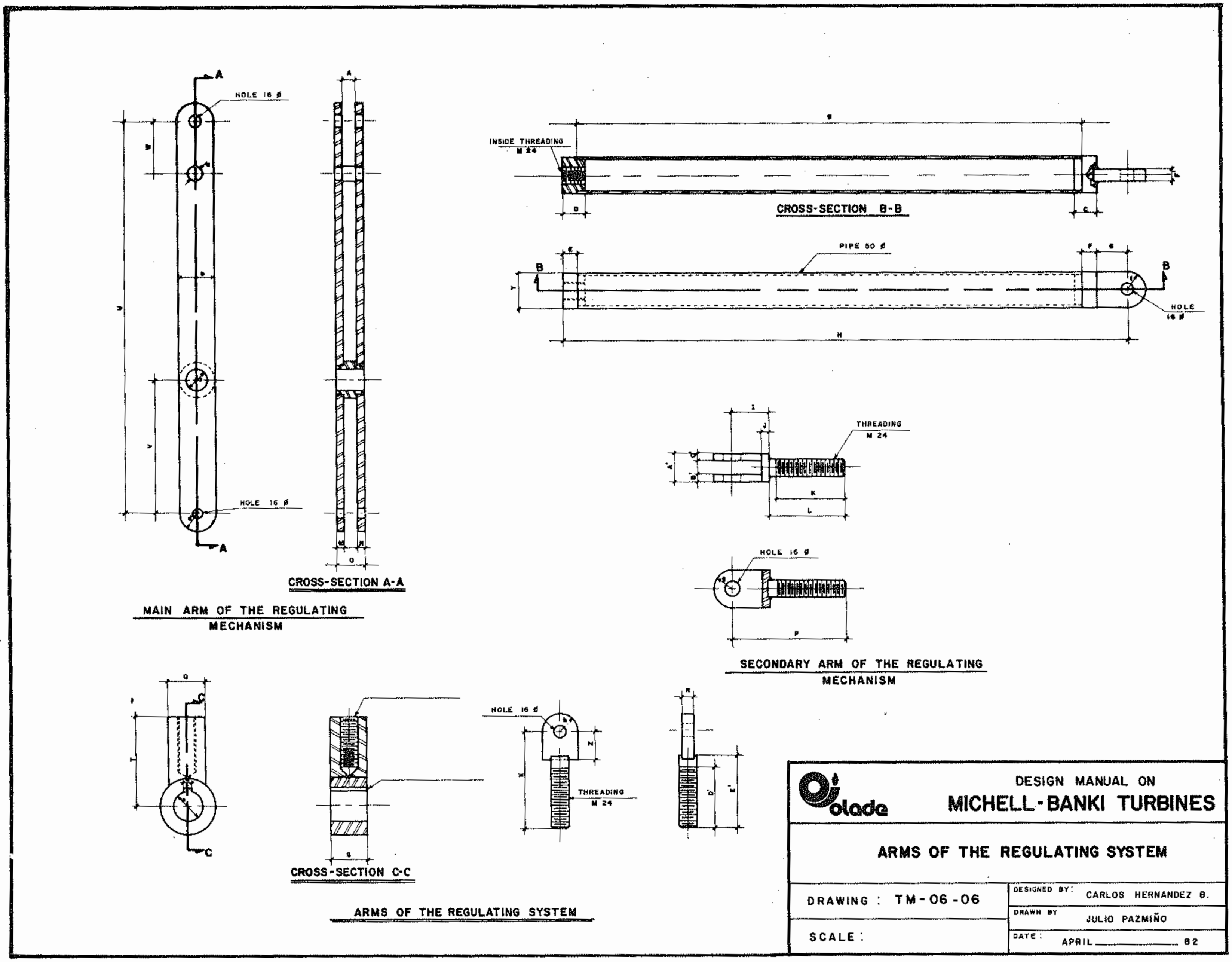
MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-06-05

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A			45	55	60	70	72	75
B			78	78	78	105	105	105
C			289	359	449	526	686	886
D			78	78	78	105	105	105
E			45	55	60	70	72	75
F			45	55	60	70	72	75
G			580	680	785	946	1112	1321
H			10	10	12	12	12	12
J			10	10	12	12	12	12
K			36	42	47	54	56	58
L			34	40	45	52	54	56
á			40	45	50	55	60	60
b			44.5	50.8	63.5	63.5	63.5	63.5
c			40	45	50	55	60	60



HOLE 16 Ø
HOLE 16 Ø
CROSS-SECTION A-A
MAIN ARM OF THE REGULATING MECHANISM
INSIDE THREADING M 24
CROSS-SECTION B-B
PIPE 50 Ø
HOLE 16 Ø
THREADING M 24
HOLE 16 Ø
SECONDARY ARM OF THE REGULATING MECHANISM
CROSS-SECTION C-C
HOLE 16 Ø
THREADING M 24
ARMS OF THE REGULATING SYSTEM
olade
DESIGN MANUAL ON
MICHELL-BANKI TURBINES
ARMS OF THE REGULATING SYSTEM
DRAWING : TM-06-06
SCALE :
DESIGNED BY: CARLOS HERNANDEZ B.
DRAWN BY JULIO PAZMIÑO
DATE: APRIL 82

olade

MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-06-06

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A			21	21	21	25	25	25
B			700	700	700	933	933	933
C			30	30	30	40	40	40
D			30	30	30	40	40	40
E			20	20	20	27	27	27
F			20	20	20	27	27	27
G			40	40	40	54	54	54
H			780	780	780	1040	1040	1040
I			50	50	50	67	67	67
J			10	10	10	12	12	12
K			90	90	90	120	120	120
L			100	100	100	133	133	133
M			10	10	10	12	12	12
N			10	10	10	12	12	12
O			41	41	41	49	49	49
P			150	150	150	200	200	200
Q			50	50	50	50	50	50
R			15	15	15	19	19	19
S			50	50	50	65	65	65
T			120	120	120	160	160	160
U			530	530	530	707	707	707
V			180	180	180	240	240	240
W			70	70	70	93	93	93
X			135	135	135	180	180	180
Y			50	50	50	50	50	50
Z			40	40	40	53	53	53
A'			41	41	41	49	49	49
B'			10	10	10	12	12	12

MANUAL ON DESIGN OF MICHELL BANKI TURBINES

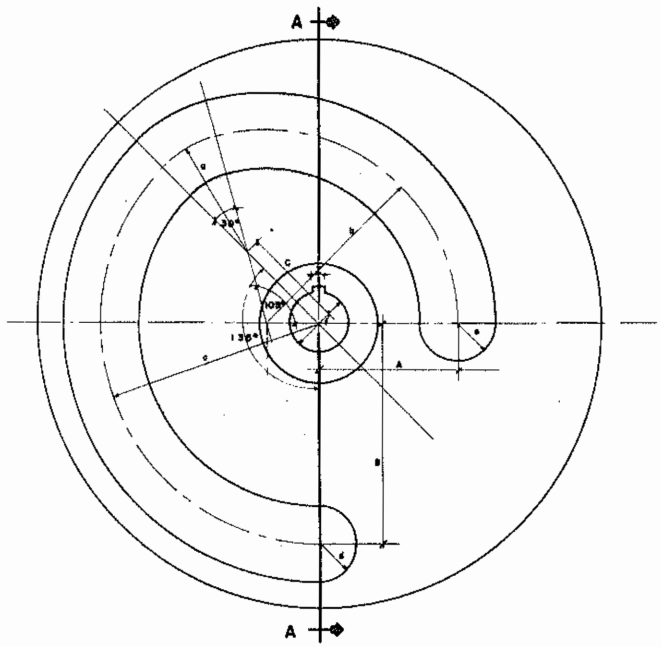
TABLE OF DIMENSIONS

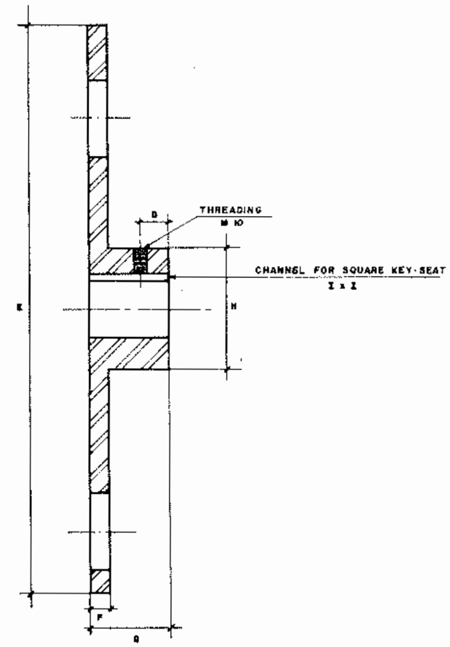
(mm)

(CONTINUATION)

DRAWING Nº TM-06-06

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
C'			10	10	10	12	12	12
D'			10	10	10	12	12	12
E'			100	100	100	133	133	133
F'			15	15	15	19	19	19
G'			10	10	10	12	12	12
a			20	20	20	25	25	25
b			50	50	50	70	70	70
c			28	28	28	38	38	38
d			25	25	25	35	35	35
e			38	38	38	48	48	48
f			25	25	25	25	25	25
g			25	25	25	25	25	25
h			25	25	25	25	25	25





CROSS-SECTION A-A

olade

DESIGN MANUAL ON

MICHELL-BANKI TURBINES

LEFT-HAND CAM OF REGULATING MECHANISM

DRAWING : TM - 06 - 07	DESIGNED BY : CARLOS HERNANDEZ B.
	DRAWN BY : JULIO PAZMIÑO
SCALE :	DATE : APRIL ______ 82

olade

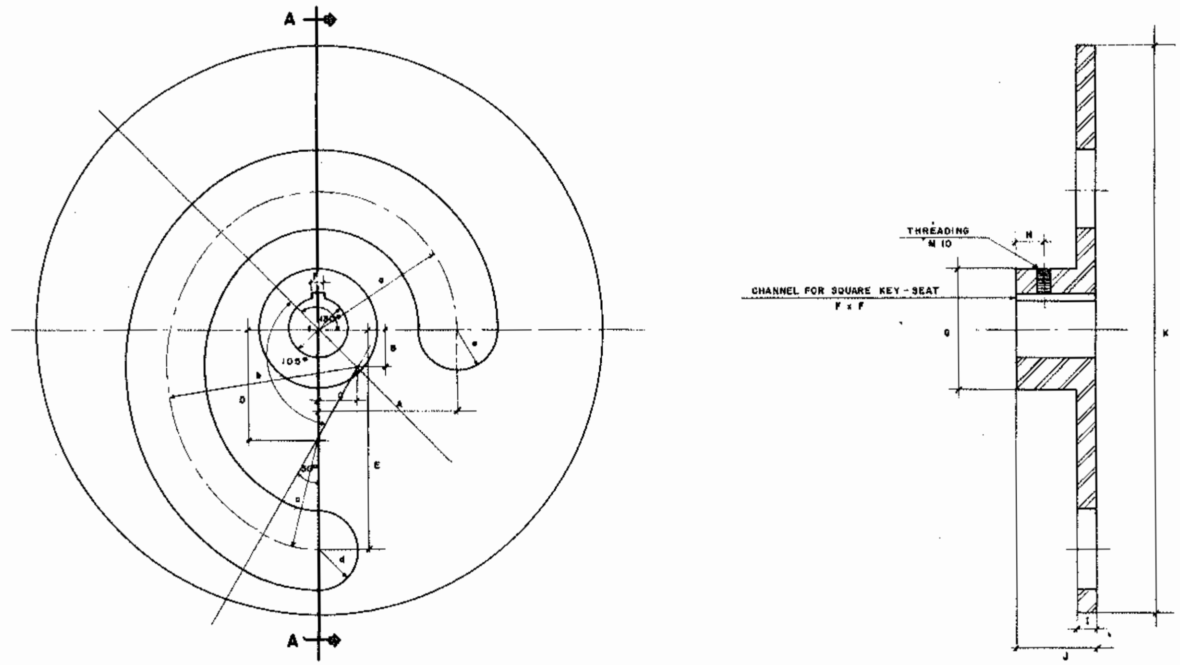
MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-06-07

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	-	-	90	-	90	120	120	120
B	-	-	140	-	140	187	187	187
C	-	-	64	-	64	85	85	85
D	-	-	15	-	20	25	25	25
E	-	-	360	-	360	480	480	480
F	-	-	15	-	15	19	19	19
G	-	-	45	-	60	70	72	75
H	-	-	70	-	100	110	115	120
I	-	-	10	-	12	12	12	12
a	-	-	76	-	76	101	101	101
b	-	-	120	-	120	160	160	160
c	-	-	140	-	140	187	187	187
d	-	-	26	-	26	36	36	36
e	-	-	26	-	26	36	36	36
f	-	-	36	-	47	54	56	58



CROSS-SECTION A-A

DESIGN MANUAL ON

MICHELL-BANKI TURBINES

olade

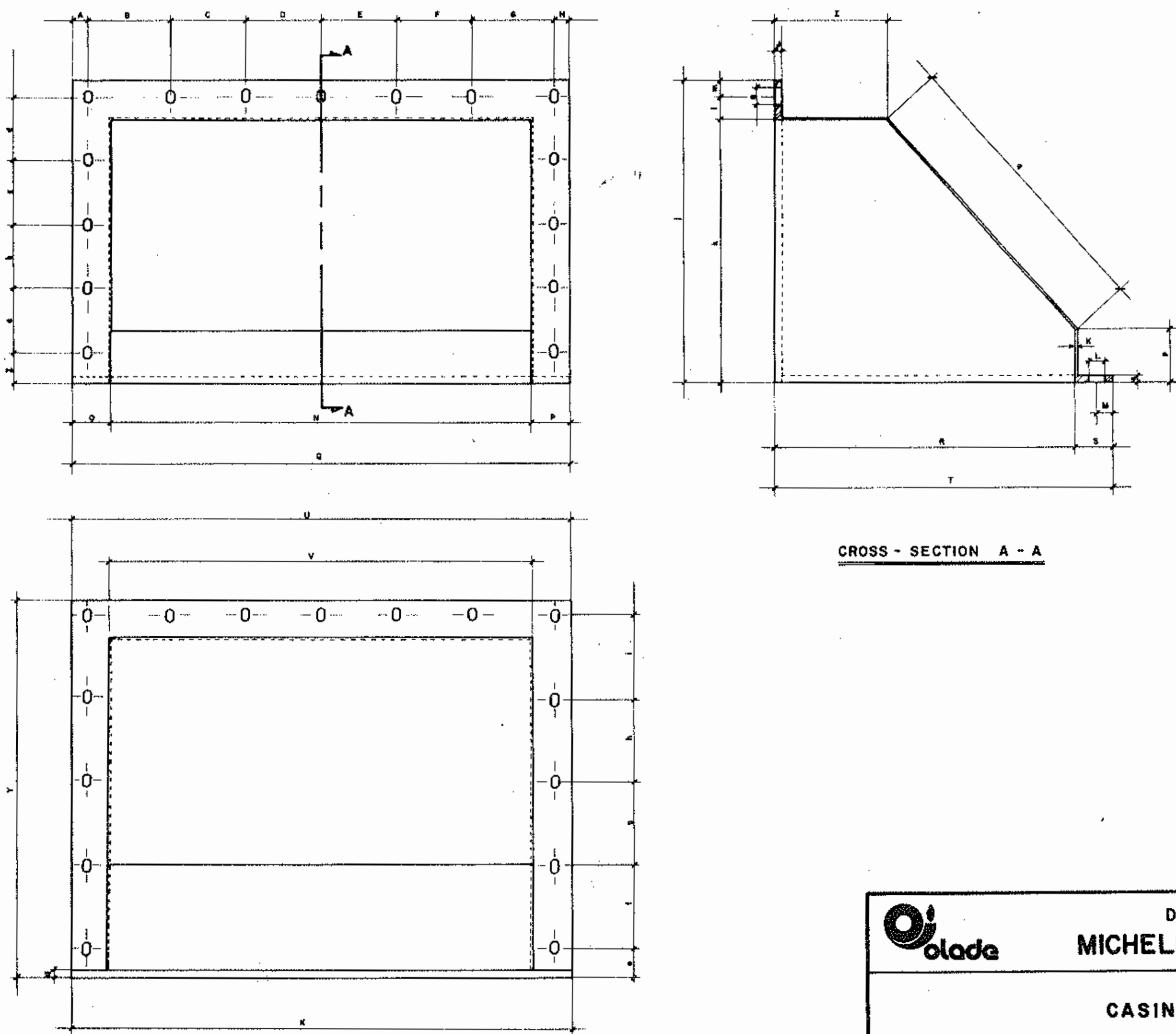
RIGHT-HAND CAM OF REGULATING MECHANISM

DRAWING : TM-06-08	DESIGNED BY:	CARLOS HERNANDEZ B.
	DRAWN BY:	JULIO PAZMIÑO
SCALE :	DATE:	APRIL ______ 82

TABLE OF DIMENSIONS
(mm)

DRAWING Nº TM-06-08

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	-	-	90	-	90	120	120	120
B	-	-	23	-	23	31	31	31
C	-	-	25	-	25	33	33	33
D	-	-	70	-	70	93	93	93
E	-	-	140	-	140	186	186	186
F	-	-	10	-	12	12	12	12
G	-	-	70	-	100	110	115	120
H	-	-	15	-	20	25	25	25
I	-	-	15	-	15	19	19	19
J	-	-	45	-	60	70	72	75
K	-	-	360	-	360	480	480	480
a	-	-	90	-	90	120	120	120
b	-	-	120	-	120	160	160	160
c	-	-	70	-	70	93	93	93
d	-	-	26	-	26	36	36	36
e	-	-	26	-	26	36	36	36
f	-	-	36	-	47	54	56	58



DESIGN MANUAL ON
MICHELL-BANKI TURBINES

CASING

DRAWING : TM - 07 - 01	DESIGNED BY: CARLOS HERNANDEZ B.
	DRAWN BY: JULIO PAZMIÑO
SCALE :	DATE: APRIL ________ 82

MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

DRAWING Nº TM-07-01

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	20	20	20	20	20	25	25	25
B	x	x	x	x	x	x	x	x
C	x	x	x	x	x	x	x	x
D	x	x	x	x	x	x	x	x
E	x	x	x	x	x	x	x	x
F	x	x	x	x	x	x	x	x
G	x	x	x	x	x	x	x	x
H	x	x	x	x	x	x	x	x
I	150	150	150	150	150	200	200	200
J	10	10	10	10	10	10	10	10
K	3	3	3	3	3	3	3	3
L	20	20	20	20	20	20	20	20
M	25	25	25	25	25	25	25	25
N	350	410	550	660	760	822	1032	1232
O	40	40	40	40	40	40	40	40
P	40	40	40	40	40	40	40	40
Q	430	490	630	740	840	902	1112	1312
R	400	400	400	400	400	533	533	533
S	50	50	50	50	50	50	50	50
T	450	450	450	450	450	583	583	583
U	430	490	630	740	840	902	1112	1312
V	356	416	556	666	766	828	1038	1238
W	10	10	10	10	10	10	10	10
X	430	490	630	740	840	902	1112	1312
Y	450	450	450	450	450	583	583	583
Z	40	40	40	40	40	50	50	50
a	x	x	x	x	x	x	x	x
b	x	x	x	x	x	x	x	x

x: space between bolts, to be defined for each case

olade

MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

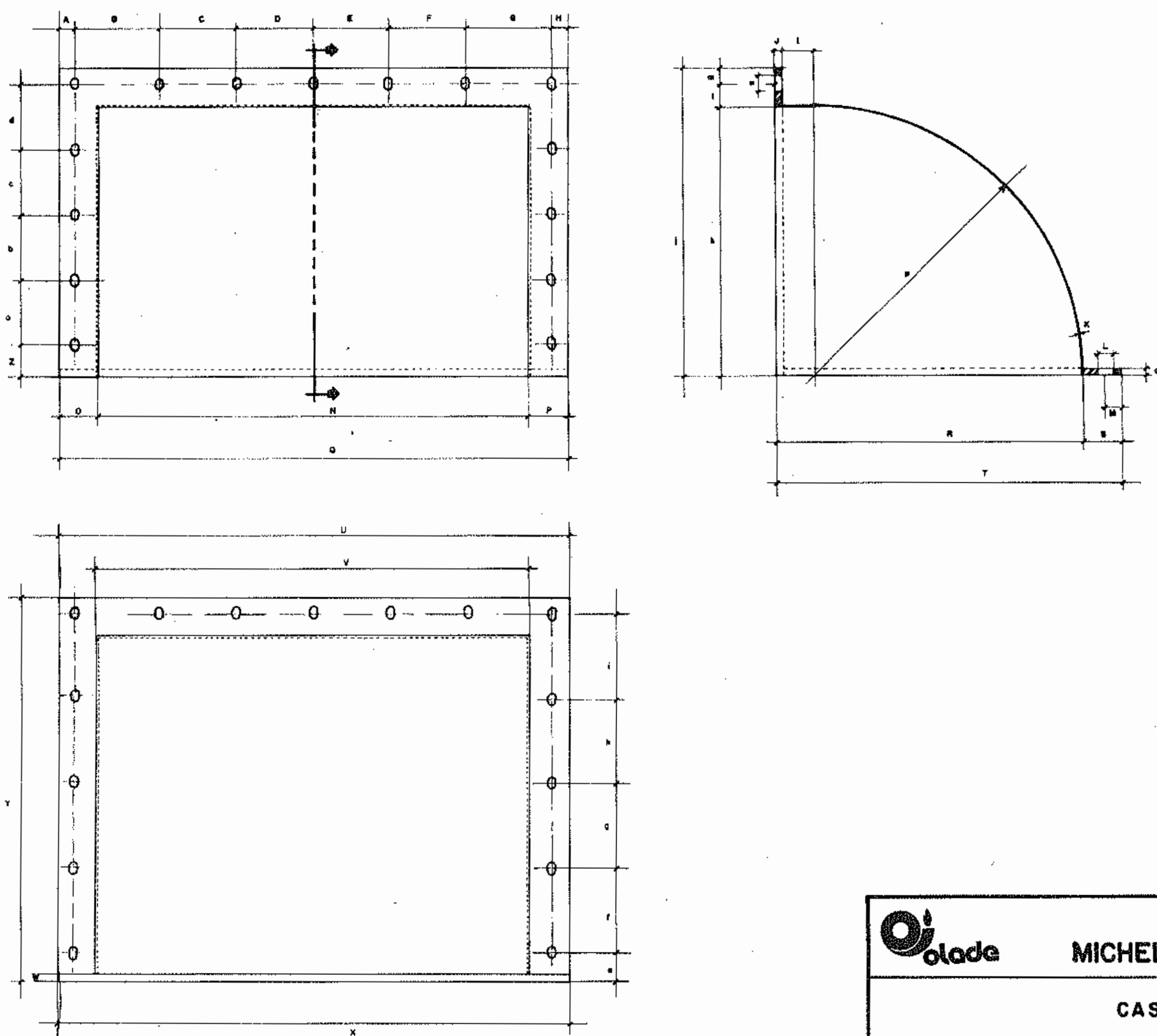
(mm)

(CONTINUATION)

DRAWING Nº TM-07-01

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
c	x	x	x	x	x	x	x	x
d	x	x	x	x	x	x	x	x
e	40	40	40	40	40	50	50	50
f	x	x	x	x	x	x	x	x
g	x	x	x	x	x	x	x	x
h	x	x	x	x	x	x	x	x
i	x	x	x	x	x	x	x	x
j	400	400	400	400	400	533	533	533
k	350	350	350	350	350	466	466	466
l	25	25	25	25	25	25	25	25
m	25	25	25	25	25	25	25	25
n	20	20	20	20	20	20	20	20
o	10	10	10	10	10	10	10	10
p	100	100	100	100	100	130	130	130
q	560	560	560	560	560	747	747	747

x: space between bolts, to be defined for each case



DESIGN MANUAL ON

MICHELL - BANKI TURBINES

olade

CASING

DRAWING : TM - 07 - 02	DESIGNED BY : CARLOS HERNANDEZ B.
	DRAWN BY : JULIO PAZMIÑO
SCALE :	DATE : APRIL ____ 82

TABLE OF DIMENSIONS
(mm)

DRAWING Nº TM-07-02

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
A	20	20	20	20	20	25	25	25
B	x	x	x	x	x	x	x	x
C	x	x	x	x	x	x	x	x
D	x	x	x	x	x	x	x	x
E	x	x	x	x	x	x	x	x
F	x	x	x	x	x	x	x	x
G	x	x	x	x	x	x	x	x
H	20	20	20	20	20	25	25	25
I	50	50	50	50	50	67	67	67
J	10	10	10	10	10	10	10	10
K	3	3	3	3	3	3	3	3
L	20	20	20	20	20	20	20	20
M	25	25	25	25	25	25	25	25
N	350	410	550	660	760	822	1032	1232
O	40	40	40	40	40	40	40	40
P	40	40	40	40	40	40	40	40
Q	430	490	630	740	840	902	1112	1312
R	400	400	400	400	400	533	533	533
S	50	50	50	50	50	50	50	50
T	450	450	450	450	450	583	583	583
U	430	490	630	740	840	902	1112	1312
V	356	416	556	666	766	828	1038	1238
W	10	10	10	10	10	10	10	10
X	430	490	630	740	840	902	1112	1312
Y	450	450	450	450	450	853	583	583
Z	40	40	40	40	40	50	50	50
a	x	x	x	x	x	x	x	x
b	x	x	x	x	x	x	x	x

x: space between bolts, to be defined for each case

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MANUAL ON DESIGN OF MICHELL BANKI TURBINES

TABLE OF DIMENSIONS

(mm)

(CONTINUATION)

DRAWING Nº TM-07-02

SYMBOL	STANDARDIZED TURBINE							
	31 A	32 A	33 B	34 C	35 B	41 B	42 B	43 B
c	x	x	x	x	x	x	x	x
d	x	x	x	x	x	x	x	x
e	40	40	40	40	40	50	50	50
f	x	x	x	x	x	x	x	x
g	x	x	x	x	x	x	x	x
h	x	x	x	x	x	x	x	x
i	x	x	x	x	x	x	x	x
j	400	400	400	400	400	533	533	533
k	350	350	350	350	350	466	466	466
l	25	25	25	25	25	25	25	25
m	25	25	25	25	25	25	25	25
n	20	20	20	20	20	20	20	20
o	10	10	10	10	10	10	10	10
p	350	350	350	350	350	466	466	466

x: space between bolts, to be defined for each case

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