Kaplan turbines: design trends in the last decade

By A. Lugaresi* and A. Massa*

This article provides an update to the results published in 1977-78[1]. As previously, the approach has been essentially statistical, based on data supplied by various manufacturers. The investigation took into account 72 units, designed, with few exceptions, after the year 1976. The research has been limited to the main parameters, such as specific speed and cavitation coefficient, and dimensions that allow for the basic unit to be selected and overall unit size to be determined. The various relationships presented here have been calculated by regression, and the results are accurate enough for a comparison of options for preliminary design and layout.

Main selection criteria

Research shows that, in spite of increasing competition from Francis and various propeller turbines, the vertical Kaplan turbine (with adjustable wicket gates and blades) is still widely adopted in the head range from 10 to 40 m.

The units included in this study have outputs from 5 to 120 MW, and operating heads from 3 to 62 m.

The main advantages of Kaplan turbines are the wide ranges of gate openings and heads which can be guaranteed and, because of the vertical arrangement, the possibility of installing high capacity units under low heads.

This makes Kaplan turbines competitive with Francis machines when: the plant is run-of-river; the head range is very wide; and/or, when the electric system requires units with wide flexibility of operation. In the low-head range, below 20-30 m, the competition is with horizontal or inclined-shaft propeller turbines, which normally require less expensive civil works. If the installed capacity is large, the possibility to reduce the number of units may favour Kaplan machines.

It is reasonable to expect, in the future, an increase in the number of low-head run-of-river plants, and low-head multipurpose plants with a wide operating head range. This should guarantee the application of Kaplan units, while, because of the increasing competition from other applicable turbines, the adoption of vertical shaft fixed-blade propellers units will probably be restricted to special cases. It should be noted that the mechanical complexity of adjustable-blade turbines is counter-balanced by flat efficiency diagrams and stable hydraulic behaviour in a wide range of gate openings and heads; in a weak electrical system, this last factor may prevail over simple economical considerations, being the best remedy against power swings produced by draft tube surges. In the near future, the application of Francis turbines could increase with the extended use of units running at double or variable speed.

Various studies show that, in the 20-40 m head range, where both Francis and Kaplan turbines are suitable, the cost of the units (turbine plus generator and auxiliaries) is often similar, because the higher cost of the Kaplan turbine is partially or totally compensated for by the lower cost of the generator, in particular when no high acceleration constant (flywheel effect) is required. This is because the Kaplan rotational speed is higher than that of a Francis machine anyway; however, the higher runaway speed of the Kaplan units reduces this advantage.

Additional savings can be obtained with the Kaplan solution when no penstock is provided and a concrete spiral case is used, in particular when the cost of the steel forms is divided between a large number of units. The cost of the civil works is usually higher with the Kaplan solution; this is mainly because of the lower setting, but in the case of poor quality foundation materials, a low setting may lead to a saving in civil costs.

In addition to costs, efficiencies, regulation and stability, the following factors should be considered when comparing Francis and Kaplan alternatives[1]:

- *The operation requirements and the need to provide an inlet valve.* When considering this factor, the time for unit start-up, the penstock filling time and procedure, and leakage through the turbine with the unit at standstill, should all be taken into account.
- *The power guaranteed at minimum head (plant dependable capacity).* At equal flow, this is usually higher with a Kaplan, in particular when the head range is wide. This factor may be important in a system having a seasonal or continuous power deficit.

It is worth mentioning that, in a Francis turbine, shifting the design head towards the minimum head to increase the plant's dependable capacity may cause severe problems during operation near the maximum head.

- *The maintenance requirements.* The critical parts of the Kaplan turbine are usually the throat ring and the periphery of runner blades, which are subject to erosion and cavitation; this can be difficult to repair.

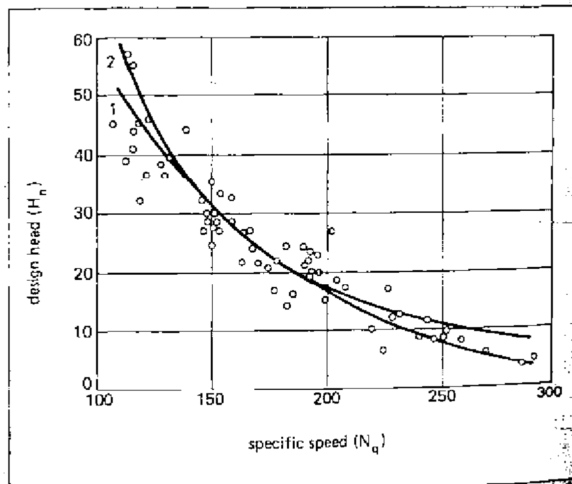

Fig. 1. Design head versus specific speed. Curve 1 is from Eq. (4), curve 2 is from the 1977 paper[1].

Notations

r	=	correlation coefficient
s	=	standard deviation
N_q	=	specific speed
n	=	rotational speed (rev/min)
K_u	=	peripheral speed coefficient (p.u.)
n_f	=	maximum steady runaway speed (rev/min)
Q	=	discharge at full opening under design head (m^3/s)
H_n	=	design head (maximum efficiency head) (m)
σ	=	cavitation coefficient
H_s	=	vertical distance between minimum normal tailrace level and runner blades centreline (negative when tube is below water level) (m)
Z	=	elevation of the runner
V	=	water speed at draft tube outlet (m/s)
D	=	runner discharge diameter (m)
g	=	gravity acceleration (m/s^2)
π	=	3.14159

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It is often possible to provide, at a reasonable cost, a removable section in the throat ring for inspection purposes, and sometimes also complete blade removal. Inspection and repair of the blade periphery and of the throat ring can thus be carried out without dismantling the unit completely.

In any case, the complete dismantling of the Kaplan runner requires the removal of the generator rotor, bearings and turbine cover. It is often necessary to dismantle simultaneously a substantial amount of the turbine's stationary components and the rotating parts. If two cranes are provided for lifting the generator rotor, the possibility of using both cranes for the turbine removal as well should be investigated unless the capacity of a single crane is sufficient for this operation.

Specific speed

As in the paper on Francis turbines[2], the dimensioned specific speed in the metric system is used:

$$N_q = n\,Q^{0.5}/H_n^{0.75} \qquad \ldots (1)$$

where Q is in m³/s, H_n in m and n in 1/min

The relationship:

$$N_q = N_q\,(H_n) \qquad \ldots (2)$$

resulting from the linear regression analysis is:

$$N_q = \frac{564.6}{H_n^{0.3873}} \qquad \ldots (3)$$

where the correlation coefficient and the standard deviation are respectively:

$$r = 0.925 \quad s = 0.26$$

This curve does not show a good fit in the high-head range, therefore some adjustments have been made, and the following regression curve was found to have a better fit with the available points over the whole head range:

$$N_q = \frac{7149}{H_n^{0.5} + 10} - 308 \qquad \ldots (4)$$

$$r = 0.953 \quad s = 46.23$$

This is shown in Fig. 1, curve 1.

Also in Fig. 1, the curve for the period 1970-1976, published in December 1977[1], is shown (curve 2). The equation of this curve, using the dimensioned value N_q for specific speed, is:

$$N_q = \frac{2419}{3\,H_n^{0.489}} \qquad \ldots (5)$$

The two curves give similar results for the head range 15-40 m. If a comparison is made of Eq. (4) with the one in the paper by Schweiger and Gregori[3], Eq. (18) it shows similar results in the range 15-40 m; out of this range, Eq. (4) is more conservative.

In conclusion, in the last decade there has been no apparent trend to increase the specific speed of Kaplan turbines.

Cavitation coefficient

The cavitation coefficient calculated previously[1,2], was referred to the guidevane centreline; during the preparation of the article on Francis turbines[2], it was verified that the scattering of data was not reduced when the cavitation coefficient referred to the elevation of the runner discharge.

Table I — Hydro plants considered in the survey

Powerplant	Supplier	Head (m)	Power (MW)	Rotational speed (rev/min)
Abbach	Voith	3.58	3.7	71.43
Alglund	K M W	28.5	6.45	375
Annabrucke	Voith	24.3	46.99	142.9
Arroyito	Escher Wyss	16.5	42.5	90.9
Aswan II	Allis Chalmers	27	75	100
Balbina	Neyrpic	21.7	51.5	105.9
Bersia	Hydroart	26.5	26.55	187.5
Blafalli	Vevey	27	9.2	333.3
Buyo	Neyrpic	33.3	44	166.7
Cachoeira Dourada IV	Neyrpic	32	115	102.9
Castro	Nohab	38.4	120.2	107.1
Curva-Una 3	Nohab	21.7	13.2	200
Djupdal	Nohab	16.2	14.87	136.4
El Tigre	Nohab	21.5	7.917	300
El Vado	Voith	32	8.29	300
Evenstad	K M W	17.2	12.1	200
Finnfors	K M W	20	22.6	166.7
Geisling	Voith	5.95	8.8	78.95
Gosho	Mitsubishi	24.1	6.5	333
Halvfari	Nohab	23	25.4	187.5
Hogfors	K M W	12.6	7.4	200
Jebba	Escher Wyss	27.65	102.7	93.75
Kellerberg	Voith	8.68	12.25	100
Kingsley	Allis Chalmers	35.1	50	180
Kykkelsrud	Nohab	27	83.42	115.4
Laxede 3	K M W	24.5	77.8	107.14
Machicura	Voith	36.7	47.1	187.5
Manavgat	Vevey	20.8	26	150
Marchtrenk	Voith	22.21	20.51	166.67
Masinga dam	Escher Wyss	45	20.6	300
Nangbeto	Escher Wyss	27	32.8	166.7
Nova Avanhandava	Hydroart	30	100.8	94.74
Nussdorf	Voith	10.09	24.98	75
Oz en Oisans	Neyrpic	40.85	11.5	333
Paternion	Voith	8.32	11.67	100
Pejepscot	Allis Chalmers	8.4	12.35	81.8
Piraju	Voith	16.8	10.08	180
Pocinho	Nohab	19.5	63.92	88.2
Porsi 3	K M W	32.5	101.1	115.38
Porto Primavera	Neyrpic	18.8	103	75
Quincinetto	Hydroart	24.35	11.25	250
Ranasfoss	K M W	12.3	44.8	75
Randi	Nohab	23.6	88	100
Regensburg	Voith	3.94	3.77	75
Rosana	Hydroart	17	80.01	75
Sidi Salem	Vevey	39.7	41	214.28
Salakovac	Escher Wyss	44.05	70.99	150
Samuel	Neyrpic	28.5	44.4	150
San Lorenzo	Voith	30	92.4	105.88
Skoltenborg	K M W	57.2	41	272.72
Solbergfoss	K M W	20	104.9	78.95
Steinsfoss	K M W	55.2	62.7	214.28
Stenkullafors	Nohab	22	57	115.38
Stenkullafors	K M W	22	56.8	115.38
Strandfossen	K M W	11.3	23.4	100
Straubing	Voith	4.92	6.88	75
Suatugurlu	Voith	29.6	25.5	200
Taquarucu	Voith	21.9	10	85.71
Torsebro	K M W	9.8	6.25	166.7
Traunpucking	Voith	24.82	22.93	166.67
Ullum	Nohab	36.5	56.4	176.5
Upper Mechanicville	Allis Chalmers	6.6	8.3	72
Vangen	Vevey	50	35.2	250
Villach	Voith	8.8	11.78	100
Villalcampo	Nohab	38.4	120.2	107.1

All units listed were designed between 1975 and 1985.

On the contrary, for the available data on Kaplan turbines which are the object of this study, the scattering is reduced when referring the turbine submergence level to the runner blades centreline. Therefore, assuming H_s to be the vertical distance between the minimum normal tailrace level and the runner blade centreline (a negative value when the runner is below the tailwater level) the cavitation coefficient may be

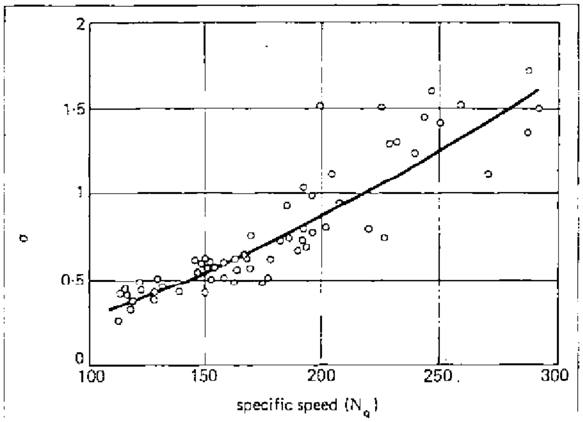

Fig. 2. Cavitation coefficient (σ) versus specific speed.

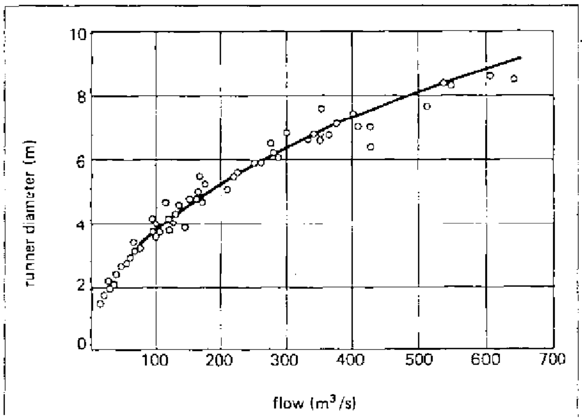

Fig. 3. Runner diameter versus flow.

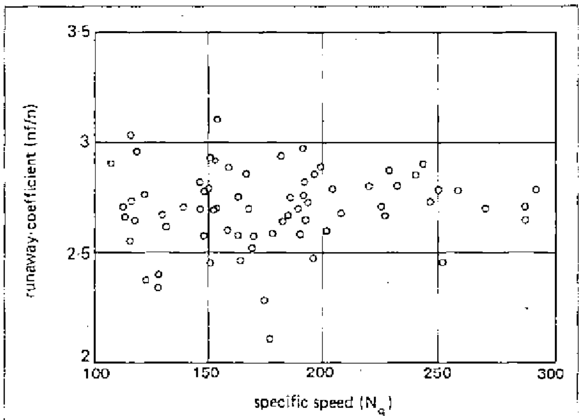

Fig. 4. Runaway coefficient n_f/n versus specific speed.

expressed as:

$$\sigma = (10.33 \times 0.9^{(z/860)} - 0.2383 - H_s)/H_n \qquad \ldots (6)$$

The tailrace level considered here is the minimum necessary value for reliable operation within the specified head range of the plant. For the purpose of this survey, the manufacturers were specifically requested to provide this value, regardless of the actual one selected. The resulting regression curve, indicated in Fig. 2, has the following form:

$$\sigma = 1.6 \times 10^{-4} N_q^{\,1.625}$$

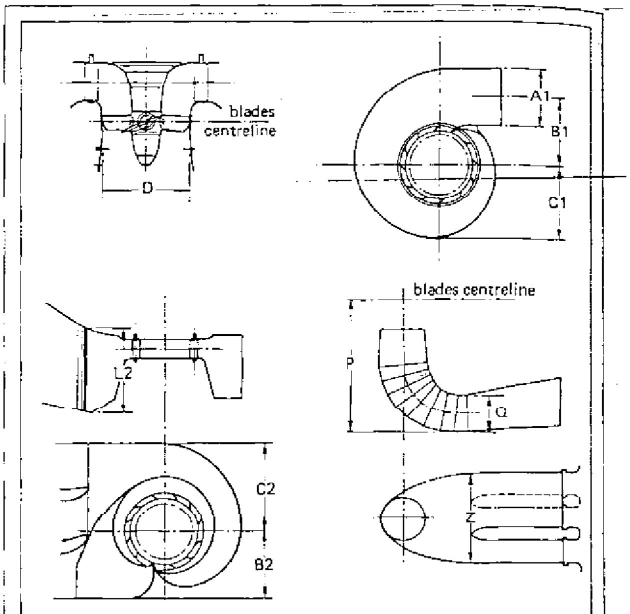

Fig. 5. Main turbine dimensions considered in the survey.

$$r = 0.918 \quad s = 0.44 \qquad \ldots (7)$$

A comparison of Eq. (7) with that of the 1977 paper[1] shows approximately the same results in the case of units with an external runner diameter of 5 m. Eq. (7) is more conservative when the turbine is smaller in size and it is less conservative in the case of large units.

Runner size

The investigation has been limited to the external diameter of the runner, D. Three procedures were used for the calculation; first, through the peripheral velocity coefficient K_u, defined as:

$$K_u = \frac{\pi D n}{60\,(2\,g\,H_n)^{0.5}} \qquad \ldots (8)$$

and the specific speed; second, assuming the square root of the nominal flow to be an independent variable, that is, the flow at full nominal opening under the design head; and, third, using the flow directly.

The third procedure, which is more empirical, leads to the highest correlation coefficient. The results are as follows:

$$K_u = 0.90163 + 0.004242\,N_q$$

$$r = 0.962 \qquad \ldots (9)$$

$$D = Q^{0.5}/2.72$$

$$r = 0.980 \qquad \ldots (10)$$

$$D = Q^{0.4696}/2.289$$

$$r = 0.988 \; s = 0.41 \qquad \ldots (11)$$

Eq. (11) is shown in Fig. 3.

Comparing Eq. (10) with the relative one for the Francis turbine[2], the Kaplan runner is found to be about 7 per cent larger than the equivalent Francis one or, in other words, the average axial component of the absolute water speed at the Kaplan runner outlet is 14 per cent lower than for Francis. Eq.

(10) is practically the same as in the 1977 article[1]. Apparently the discharge water speed is almost constant both in Francis and Kaplan turbines, and the value is about 10 m/s.

Even if the correlation coefficient of Eq. (10) is excellent, the resulting water speed is an average component, and it is calculated by a statistical procedure: when local and actual values are considered, the trend is much less evident.

Runaway speed

As already mentioned in the article on Francis turbines[2], at least two speeds should be considered when designing the rotating parts of a unit: the maximum momentary overspeed, and, the maximum steady-state runaway speed. The possibility of reaching the maximum momentary overspeed under runaway conditions, during simultaneous sudden load rejection of other units fed by the same penstock[2], is extremely rare, particularly with Kaplan units which are usually fed by single penstocks.

Overspeed caused by sudden load rejection in Kaplan turbines is always much lower than the runaway speed.

Kaplan turbines may achieve runaway speed in on-cam conditions and off-cam conditions. If the unit has no special protection and/or design provision, the maximium steady runaway speed, which is obtained under off-cam conditions with guidevanes fully open and a small runner-blade angle, is higher than any maximum momentary overspeed.

On the other hand, the maximum momentary overspeed, defined as "the momentary overspeed attained under the most unfavourable transient conditions", may constitute the worst condition, if the unit is designed in such a way as to prevent sustained operation in dangerous off-cam conditions.

The large variety of possible conditions is therefore one explanation for the scattering of data in Fig. 4. All these values should be considered as representing the maximum steady-state runaway speed in the most unfavourable condition.

No correlation has been found between the ratio of runaway/rotation speed and any other main parameter. The average value obtained with the available data is 2.7. In cases where bidding is carried out for the turbine and generator jointly, the values of the maximum momentary overspeed and maximum steady runaway speed should be declared and proved by the manufacturers. As suggested by the IEC, no site runaway test should be specified.

If the turbine and generator are bid for separately and simultaneously, the values should be specified based on previous experience with similar machines.

The possibility of reducing the risk for operating the unit under runaway conditions, by providing the unit and/or the plant with protective equipment, should not relieve the manufacturers from the obligation to design the unit to withstand the maximum off-cam runaway speed. A safety coefficient, with respect to the yield point, close to unity, could be accepted, depending on the reliability of the protection.

It is worth mentioning that, as a safety measure, many Kaplan units are designed so that the natural opening tendency of the runner blades is more towards large angles, to avoid the small angles' off-cam conditions. While this limits the speed, rather dangerous vibrations are generated, unless the energy entering the unit is reduced quickly. To this effect, the closure of a downstream gate (gate stroking) could be an effective measure.

Other dimensions

The statistical investigation has been limited to the dimensions shown in Fig. 5.

The missing dimensions, in particular those of the spiral casing, can be obtained either by physically drawing the element, or by using the formulae given in the previous article[1], which are still valid.

The design of Kaplan turbines may allow for two basic alternatives: a steel or concrete spiral casing. The choice mainly depends on the plant layout; size and pressure may also be limiting factors, but available data show that there is a wide overlap in the 20 and 35 m head range. Of all the plants investigated, about 50 per cent have a steel spiral case.

When the design parameter $D\,H_n$ (m^2) is considered, where D is the inlet spiral case diameter (or equivalent diameter in the case of concrete spiral case) and H_n the design pressure, the maximum value for concrete spiral cases is 320 and for steel spiral cases, 424; the difference is not therefore substantial.

The use of concrete spiral cases above 35 m head is probably rare only because, with a higher head, a penstock is required in any case.

The best correlation coefficients were found when the influence of the specific speed was neglected and all dimensions were calculated considering the maximum runner diameter to be an independent variable.

The interpolating functions, shown in Figs. 6 to 14, are as follows:

$$A1 = 1.304\,D \qquad r = 0.987 \qquad \ldots (12)$$

$$B1 = 1.385\,D \qquad r = 0.991 \qquad \ldots (13)$$

$$C1 = 0.428 + 1.498\,D \qquad r = 0.981 \qquad \ldots (14)$$

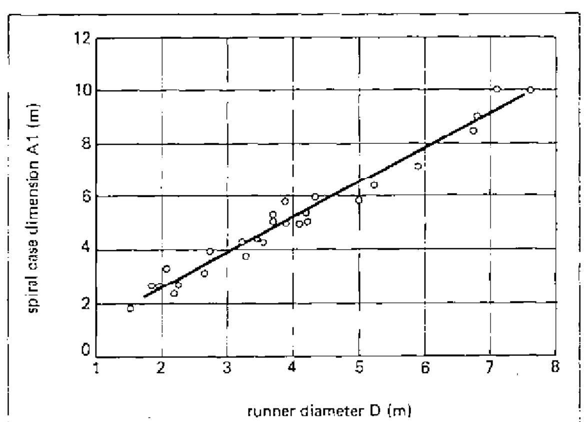

Fig. 6. Spiral case dimension **A1** *versus runner diameter* **D**.

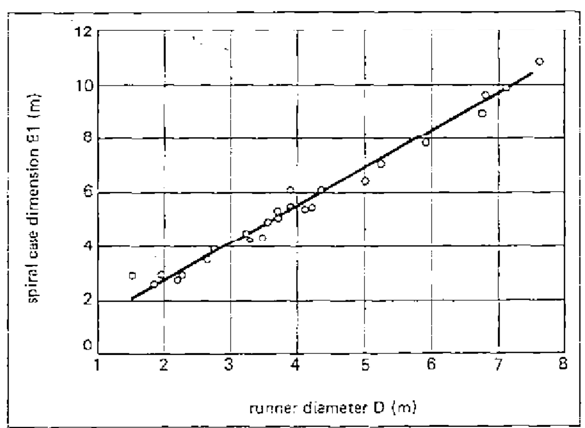

Fig. 7. Spiral case dimension **B1** *versus runner diameter* **D**.

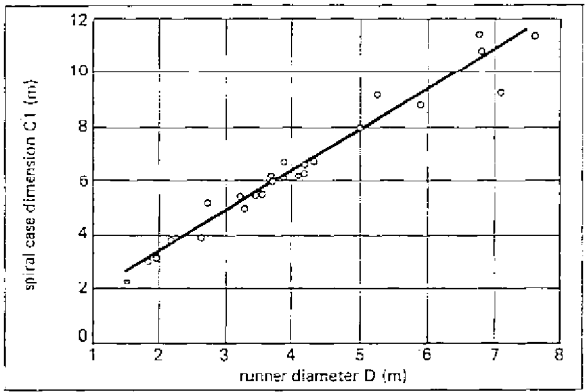

Fig. 8. Spiral case dimension C1 *versus runner diameter* D.

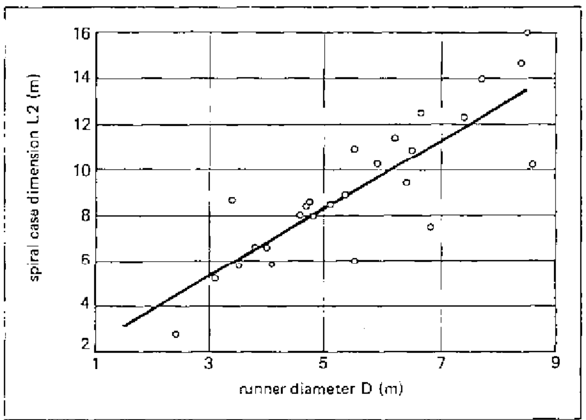

Fig. 9. Spiral case dimension L2 *versus runner diameter* D.

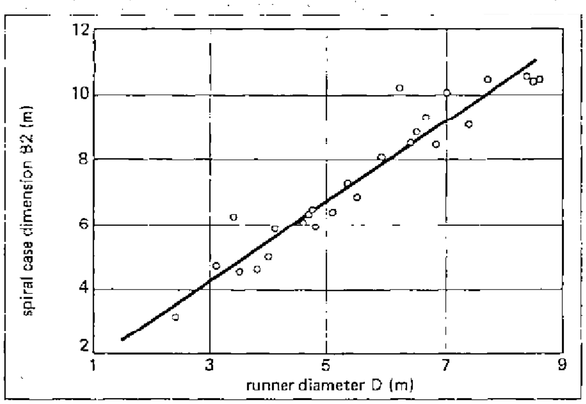

Fig. 10. Spiral case dimension B2 *versus runner diameter* D.

$$L2 = 0.939 + 1.475\,D$$
$$r = 0.848 \qquad \ldots (15)$$

$$B2 = 0.578 + 1.229\,D$$
$$r = 0.953$$

$$C2 = 0.6 + 1.684\,D$$
$$r = 0.964 \qquad \ldots (16)$$

$$P = -1.306 + 2.676\,D$$
$$r = 0.961$$

$$Q = 0.6\,D$$
$$r = 0.9 \qquad \ldots (17)$$

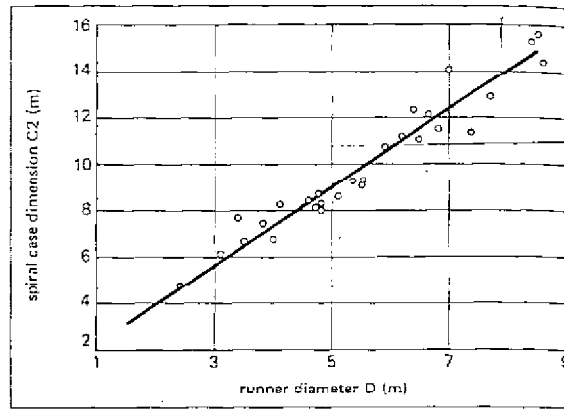

Fig. 11. Spiral case dimension C2 *versus runner diameter* D.

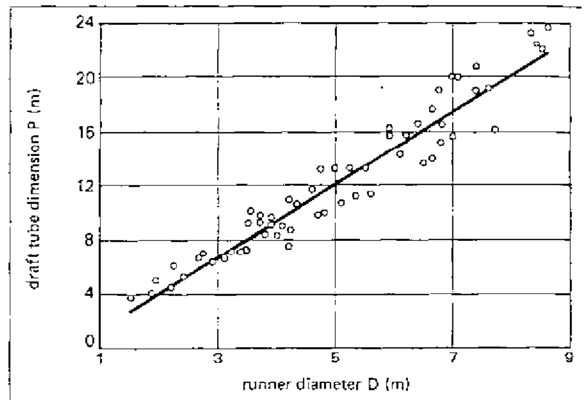

Fig. 12. Draft tube dimension P *versus runner diameter* D.

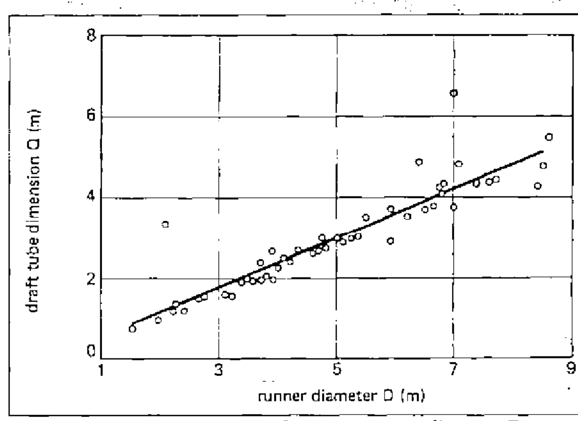

Fig. 13. Draft tube dimension Q *versus runner diameter* D.

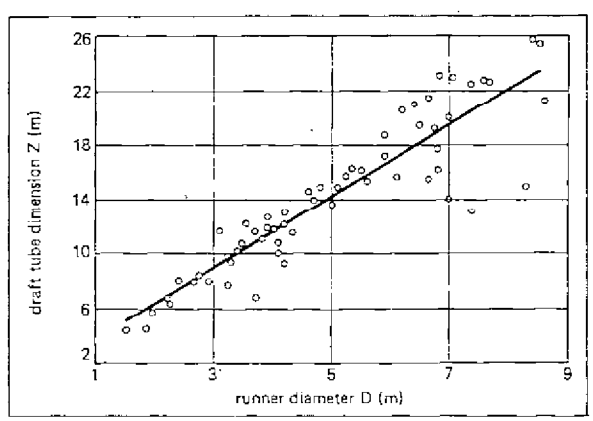

Fig. 14. Draft tube dimension Z *versus runner diameter* D.

$$Z = 1.172 + 2.616\,D$$
$$r = 0.91 \quad \ldots (18)$$

The distance from the machine centreline to the bottom of the draft tube can be obtained summing up the above dimension ~~P and 0.4D, which is the distance from the runner blade~~ centreline to the machine centreline.

As in the previous article for Francis machines[2], no satisfactory interpolating function has been found for the water speed at the draft tube outlet.

Fig. 15 shows the ratio, in per cent, between the discharge velocity head and the design head ($V^2/(2g\ H_n)$) versus the design head. □

Acknowledgement

The authors wish to thank all the manufacturers mentioned in the Table for supplying the data and dimensions of their machines, and thus making this study possible.

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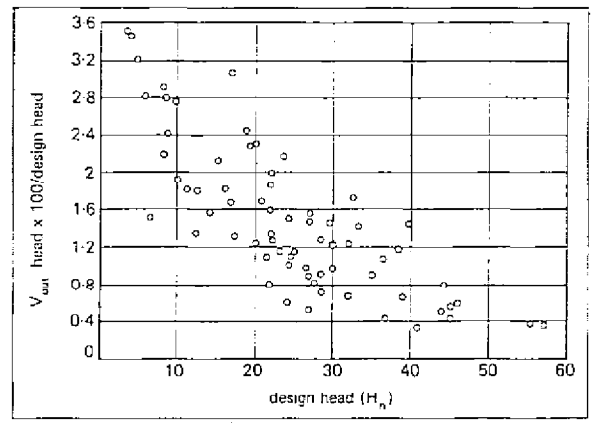

Fig. 15. Ratio between draft tube velocity head and design head versus design head.

Numerical refinement of Palmiet's pump-turbine design

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The efficiency level of the two Palmiet pump-turbines, supplied by J. M. Voith GmbH, FRG, was considerably improved in both modes of operation, at the design stage, by the application of numerical methods. This article describes the integrated numerical system (INS) used for the hydraulic improvement of the low specific speed pump-turbines with a head of 300 m and an output of 250 MW. Some principal results of the preliminary computations, the model test results, and the improvement of the efficiencies related to the original runner design, are shown. The commissioning of the Palmiet powerplant is to take place this month.

In the southwest of South Africa, near Cape Town, the Palmiet pumped-storage plant is under construction. The Electricity Supply Commission, ESCOM, contracted J. M. Voith GmbH of West Germany to supply and deliver single-flow, single-stage reversible pump-turbines, together with governors, spherical valves and auxiliary equipment. Fig. 1 shows the simplified longitudinal section through the pumped-storage plant.

The pump-turbines have vertically arranged shafts (see Table I). The order for the pump-turbines called for the highest efficiencies in both modes of operation, and this was achieved by numerical methods.

For some years the use of numerical methods for the development of hydraulic machinery has been gaining ground. A decisive factor for this success is that the new computational methods, in conjunction with concomitant tests and measure-

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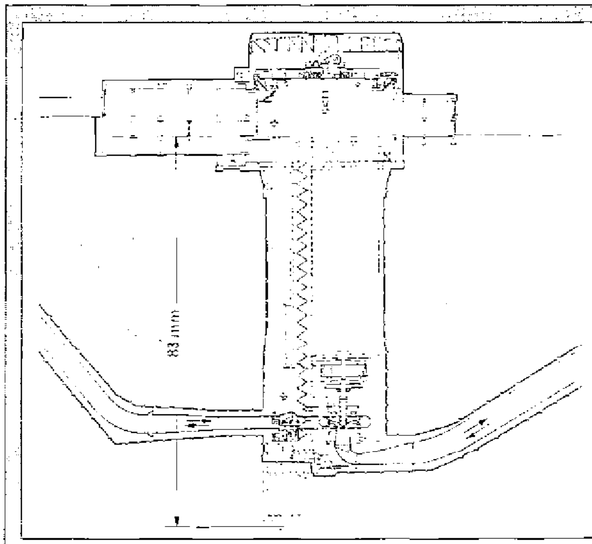

Fig. 1. Simplified section through the Palmiet pumped-storage plant.

Table I — Palmiet plant data

	Turbining operation	*Pumping operation*
Rotational speed	300 rev/min	300 rev/min
Max. output	253 MW	203.3 MW
Head/delivery head	301.4 m	258.8 m
Discharge	96.81 m³/s	74.57 m³/s