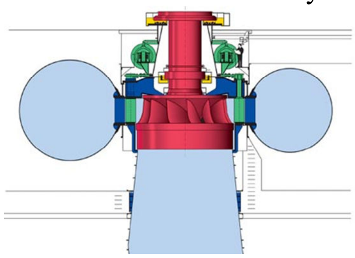
A Century of Turbine Research without Innovation?

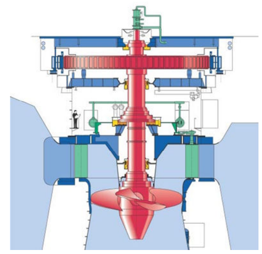
Mads Grahl-Madsen
President, CEDI Sp.zo.o

Introduction

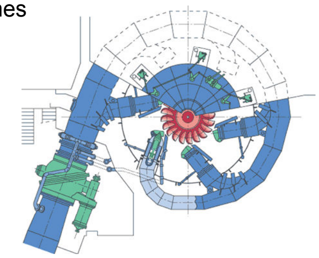
This is a rather provocative title, and clearly the term innovation is essential to the answer of this question. However, the question do not imply a lack of respect for the many brilliant people who has contributed to the development of hydraulic turbines and still contribute substantially to the continuing development of this technology.



Francis Turbine:
Developed in 1848 by James B. Francis(1815 -1892)



Kaplan Turbine:
Developed in 1913 by the Austrian professor Viktor Kaplan (1876 –1934) .



Pelton Turbine: Invented in the 1870s by Lester Allan Pelton (1829-1908),

Figure 1 The most commonly used turbine types

If one with the term of innovation means the release of new products that substantially has changed the business, the answer is obviously: Yes, this has been a century without much innovation. As illustrated in Figure 1we still use the same turbines as before and much of the development of them is mainly due to increased material qualities, better machining tools, introduction of digital controllers, etc. The focus from a research and development perspective has mainly been on increased performance and performance prediction. If a wide definition of the term innovation is applied, one that is more in line with the dictionaries, there has been substantial innovation related to analysis and the understanding of their behaviour. But, and there is a but, the research and development community have far to often neglected developments from other areas of research that could have been useful, and also to the full extend shown that important discoveries is lost between generations. We have reinvented the wheel many times through the last century.

To comment on all the areas of achievements through a century is not possible, only a few areas of importance both to the designer and the power plant operator will be discussed. For details the reader is referred to the references.

Analysis of turbine flows

The ability of a hydraulic turbine to meet with the specified requirements of generated power, efficiency and stability, are essential to the success of the design. Through all times the possibility of analyse the flow through the turbine therefore has been essential to the practising engineer. A lot of work has been done in this area, in the early years until the 1990's this analysis was based upon the assumption of negligible influence of the fluid viscosity, but since then, thanks to CFD (Computerised Fluid Dynamics) analysis, fully viscous, turbulent and even non stationary models has been applied, Brekke (1996) [6], . However, the early work has not lost is significance to a well designed hydraulic turbine, quite a few of the effects experienced through the operation of hydraulic turbines and the relation of these effects to the design parameters of the turbine can still today only be understood by more simplified analysis then available through the modern computerised design methods.

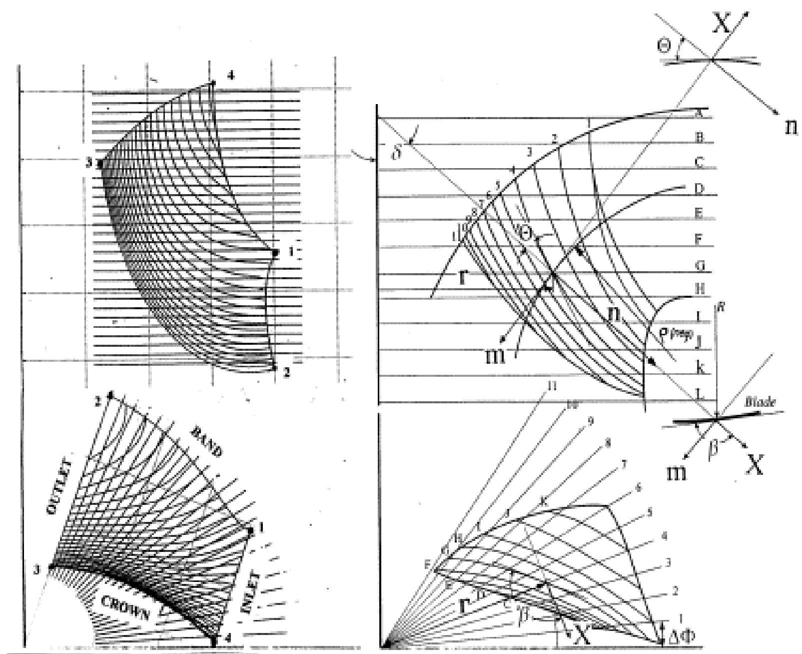


Figure 2 Illustrating the Graphical Solution of the flow through a Francis Turbine Runner

The first comprehensive theoretical treatment of the flow through radial machines was done by Lorenz (1906) [31], and later by Bauersfeld (1912) [3] who applied the theory in Francis Turbines particularly. Dreyfus (1946) [11] published a considerable theoretical work dealing with the internal flow of Francis and propeller turbines, a work that shows a great mathematically capacity, but even today with almost unlimited computer capacity will be hard to carry out on real machines. In Norway Professor Sundby (1937-1938) [35] was a pioneer forming the theoretical basis for the design of Francis turbines, especially turbines with low specific speed. Although his work, assumingly, was heavily influenced by Lorenz (1906) [31], it had a very practical inclination. AS reported by Brekke (1987) [5], this theory is especially valuable in the analysis of the blade design for low specific speed turbines. A general development of this theory can be found in Grahl-Madsen (1985) [15].

Several writers, some of them mentioned above, but also American authors like Csanady (1964) [9] and Wislicenus (1965) [36] wrongly in their theoretical work suggested that the term $d\frac{(u \cdot c_u)}{dn}$ and the term $d\frac{(u \cdot c_u)}{dl}$ between the guide vanes and the runner, both equal zero due to the fact that no torque is transferred prior to the runner inlet edge. u is the peripheral velocity of the runner, c_u the peripheral component of the velocity, n the direction normal to the direction of flow and l the actual direction of the flow. This is due to the flow curvature in this region of the turbine not correct. Only the second of the above terms can be considered equal to zero. As a consequence a lot of turbines where this basis was applied for the mathematical analysis were utilised suffered from major operational problems such as inlet cavitation, the turbines. However, this was not the only reason for operational problems in these turbines but probably a significant contributor taken the nature of some of the problems experienced by these designs.

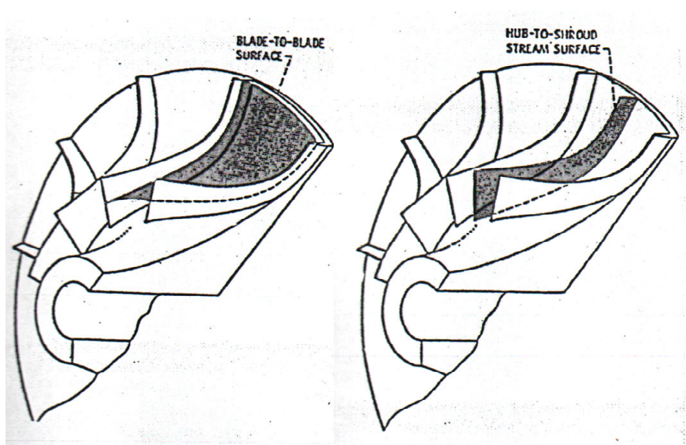


Figure 3 Illustrating Wu (1952) [37] quasi 3D flow surfaces

The Chinese professor Wu (1952) [37] made the most extensive contribution to the theory of turbine flows, and several authors has developed computer algorithms based upon his work. Among these are Hirsch and Warzee (1978) [17], Katsanis and McNally (1969) [22],Hirsch and Warzee (1978) [17], Keck and Haas (1982) [24], Kirsch (1970) [25], Katsanis and McNally (1974) [23], Gjerde (1988) [14] and Chauvin (1977) [8]. The increased capacity of computers changed this quasi 3D approach during the last part of the 1980's into a focus on the non viscous fully 3D Euler equations, Jacobsen, Billdal et al. (1993) [18], and finally into the development of viscous flow solutions based upon the full Navier Stokes equations, Andersson, Gjerde et al. (1988) [2],Jacobsen, Brekke et al. (1990) [19] and Brekke, Jacobsen et al. (1990) [7].

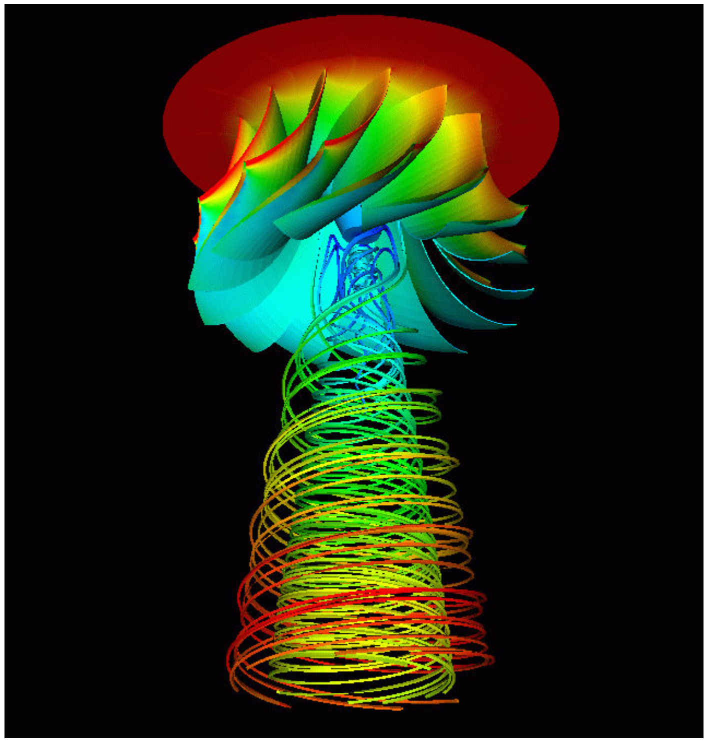


Figure 4 Numerical Results from the GAMM Francis Turbine Runner Davidson: [10]

Real Flow phenomena and their influence on turbine design

Test carried out at the Norwegian University of Science and Technology in the 1960's in order to verify the surface roughness influence on the turbine efficiency showed some interesting results. The test are summarised in Figure 5, and was by no means fully understood at the time.

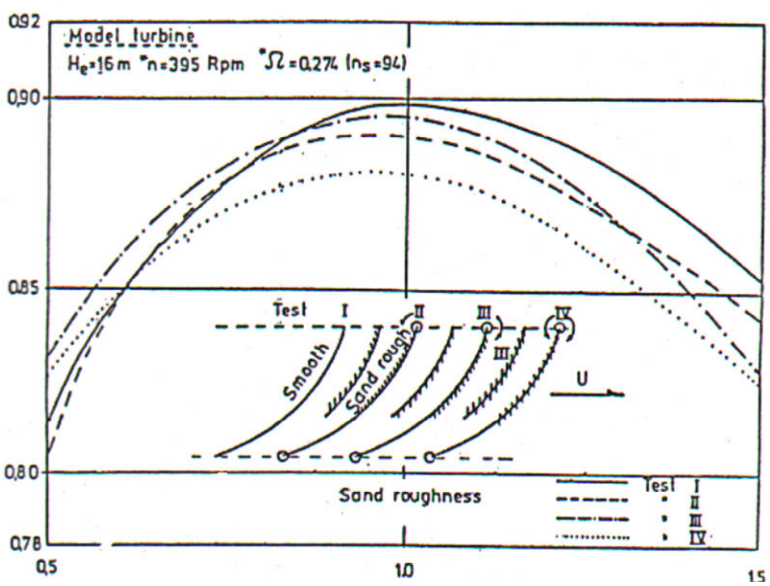


Figure 5 Model turbine tests

In order to understand and their significance to turbine design, both mathematical analysis and extensive research into a variation of sources was necessary, Grahl-Madsen (1985) [16].

The non-dimensional Navier Stokes equation written for the flow through a Turbine Impeller, Grahl-Madsen (1985) [16], can be written as:

$$-\vec{v}\times\left(\nabla\times\vec{v}\right)=-R_o^{-1}\cdot\left(\vec{k}\times\vec{v}\right)-\tfrac{1}{2}\cdot\nabla\cdot\left(1-u\cdot c_u\right)+R_e^{-1}\cdot\nabla^2\cdot\vec{v} \qquad (1.1)$$

And

$$\nabla\cdot\vec{v}=0 \qquad (1.2)$$

The quantities are done non-dimensional by use of the following characteristic expressions:

- $V=\left(2\cdot g\cdot H\right)^{\frac{1}{2}}$ (m/sec)
- D_2 (m)
- $2Y=2\cdot g\cdot H$ (J/kg)

The Rossby Number $R_o=\sqrt{2\cdot g\cdot H}/2\cdot\omega\cdot D$ is a measure of the influence on the flow field from the Coriolis forces, and the Reynolds number $R_e=\sqrt{2\cdot g\cdot H}\cdot D/\nu$ likewise is a measure of influence from viscous forces. The significance of the Rossby number was first (the knowledge of the author) demonstrated in an investigation by Fischer and Thoma (1932) [12]. They visually investigated a centrifugal impeller, and found some very peculiar behaviour. The result of this investigation is shown in Figure 6, where the hatched area indicates separated flow. The speed increase from left to right, while the flow increase vertically. As seen the amount of separated flow along the blades suction side increase with increasing speed and decreasing flow. Separation is also found at overload on the blade pressure side, but it seems to be less dominating then the suction side separation.

Similar results has later been found by a number of other researchers such as Schatzmayr [34], Johnstone, Edge et al. (1991) [21], Lennemann and Howard (1970) [30], Prandtl (1930) [33], Bradshaw (1969) [4; Moore (1973) [32], Furtner and Raabe (1980) [13], and Johnson and Moore (1983) [20].

Figure 6 Flow visualisation in Centrifugal Pump Impeller Fischer and Thoma (1932) [12]

Both Prandtl (1930) [33] and later Prandtl (1930) [33; Bradshaw (1969) [4], found that curvature and Coriolis forces both will influence the turbulent flow by stabilising or destabilising the flow. I the first case the turbulence activity will decrease, while the destabilizing of the flow will increase the turbulence level. As a measure on this effect there can be defined a characteristic number, the Richardson number.

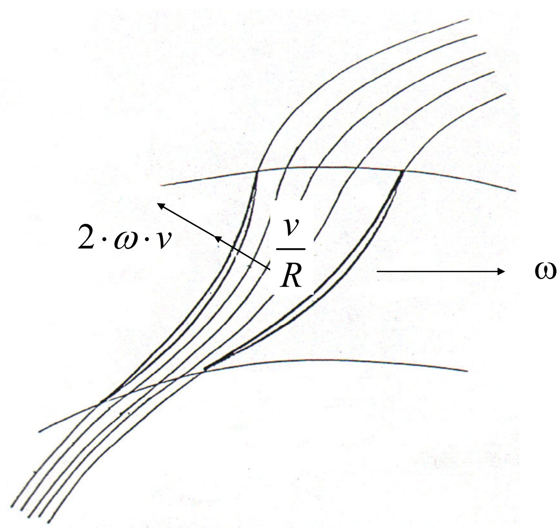


Figure 7 Flow between two turbine blades and the forces acting perpendicular to the flow

$$R_{iC} = \frac{\frac{2}{R^2}\cdot\frac{\partial(c\cdot R)}{\partial n}}{\left(\frac{\partial c}{\partial n}\right)^2} \tag{1.3}$$

And

$$R_{iR} = \frac{-2\cdot\omega\cdot\left(\frac{\partial v}{\partial n}+2\cdot\omega\right)}{\left(\frac{\partial v}{\partial n}\right)^2} \tag{1.4}$$

If Ri > 0 the flow will stabilise, i.e turbulence will decrease, and with Ri <0 the turbulence will increase due to the destabilisation of the flow. The importance Ri can easily be explained by studying the blade to blade flow of a turbine. AS illustrated in Figure 7 both the effect of the curvature and the Coriolis force will stabilise the flow on the suction side of the blade, while the flow on the pressure side will be destabilised. As a consequence the turbulence will be suppressed on the suction side, while intensified on the pressure side.

Returning to the results from the tests in Figure 5 and assuming that the inlet flow angle at the best efficiency point coincidence with the inlet blade angle several interesting observations can be made. If the blade surface is rough the inner share layer will be disturbed and the local turbulence intensity increases. A rough surface therefore leads to a local flow structure with increased resistance against separation. When the roughness is increased at the blade suction side, the efficiency at the best efficiency point, drop due to increased friction losses. However, at part load when Ro increase the efficiency soon becomes higher then the initial curve where the blade surface is smooth. This is even found when both blade surfaces have an increased roughness. If only the pressure side of the blade is given an increased roughness, there is no change in the part load performance. At overload all results show that the friction losses are dominant.

All these results shows the importance of a focus on details when designing a turbine, and as seen in Figure 5 the effect can be dramatic on the turbine performance. Unfortunately this effect can not be calculated to any accurate quantitative level, but the turbine designer should be well aware of them when designing the turbine.

Performance verification Measurement

Even if the individual location of a turbine, set local operational and performance criteria's, the plant owner should always make sure that the performance of the turbine is in accordance with the specification. For low head turbines, one has to depend on model tests performed in accordance with international standards. On high head turbines, above 100 to 150 meters, an alternative is found in the Thermodynamic method for efficiency measurements, Alming and Vinnogg (1986) [1], Kjölle (1978) [26], Kjölle (1983) [27], Kjölle (1983) [28] and Kjölle (1983) [29]. The principle is shown in Figure 8.

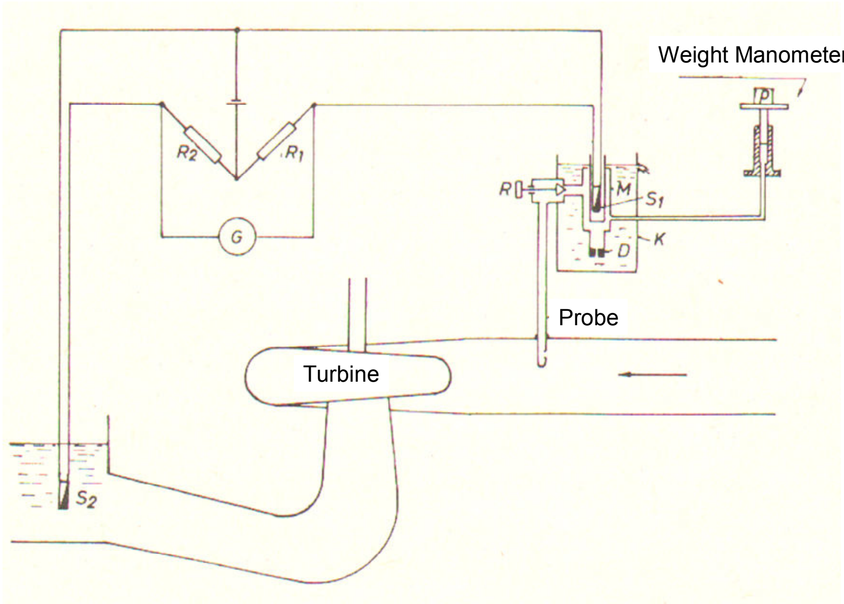


Figure 8 Thermodynamic measurement of turbine efficiency

Conclusion

A lot of good and valuable research has been carried out, but the power plants including the turbines still look like they did 100 years ago. The main research focus has been on optimization of existing turbine solutions, not on innovative new technology.

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