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The Tesla Turbine Revisited

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ABSTRACT
This work reviews the physical principles behind the Tesla bladeless turbine, a device invented by the brilliant Croatian engineer Nikola Tesla. Following a quick discussion on the relative motion of rotating surfaces, it sets up the transport equations describing the flow between parallel rotating disks, estimating the boundary layer thickness under laminar and turbulent regimes, leading to expressions yielding the width between consecutive disks. Once the working fluid is defined and its entrance conditions are established, then, if the needed (expected) output power is chosen, this work shows how to calculate the total number of disks required to attain the desired performance. Finally, it is also described the device behavior acting as an air compressor or water pump. These authors are not aware of a comprehensive discussion of the fluid mechanics involved in the design of those devices having been ever done. Besides the usual applications of rotating machinery, Tesla machines are properly fitted when compact unities are required such as in the cases of isolated areas for electric power generation. It should also be noticed that, as a unique source of rotating motion, they can run under a very wide spectrum of fuels and fluids in general.

Keywords: Tesla turbine, boundary layer turbine, rotating machinery

1.0 INTRODUCTION

Nikola Tesla broad field of interest led him to work in nearly every area of technical knowledge, ranging from his well known and valuable contributions in electrical engineering to his interest in developing a flying machine. Indeed he was one of the pillars of scientific endeavors on early twentieth century. In 1910 he was awarded a dual patent (British Office Number 24001) for a turbine and compressor using rotating disks as moving devices. These machines using the same principles, operated in a similar manner. They consisted of an array of parallel thin disks very close to each other, kept apart by spacers (washers) and assembled (mounted) on a shaft, forming a rotor which was fitted in a cylindrical housing (stator) its ends closed by plates properly fitted with bearings to hold the rotor shaft. In the central region of the disks, close to the shaft, exhaust ports were opened, with gaps in the spacers, thus providing an exit to the atmosphere (or to a condenser, depending on its intended use, i.e., if a gas turbine or a steam turbine, part of an otherwise standard thermal engine). A nozzle was located tangentially to the bore of the casing, feeding the working fluid (be it steam or combustion gases), onto the disks, rotating them while proceeding to the exhaust ports.
Since then several of these unities were built [1-3], although none of them of the gas turbine kind, possibly due to the lack of design data. Actually, as time passed by, no more disk turbines were manufactured and interest in the principle lapsed [4].
Further these authors are not aware of a comprehensive discussion of the fluid mechanics involved in the design of those devices. Hence this work, which following the estimates of the laminar and turbulent boundary layers between rotating disks, suggests how to calculate the total number of disks needed for a desired performance to be achieved. It also describes the device behavior as an air compressor or water pump.

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2.0 OPERATION PRINCIPLES

The fluid adherence to a wall (i.e., the "no-slip" condition) is the basic phenomenon behind the Tesla turbine. As the fluid acquires the velocity of the wall over which it flows, therefore a disk has the tendency to acquire the velocity of the fluid imparted over it. If this fluid is injected tangentially to a disk surface then the tangential component of the velocity vector is zero for a reference system attached to the disk surface, moving with it, then the only velocity component "seen" in this system which influences the fluid flow is the velocity component towards the center of the disk which pushes it to that region where it is discharged through the existing exhaust ports around it (so that, for an external observer, the fluid describes a spiral circuit around the disk face). As the disk tends to acquire the velocity of the fluid flowing over it, then, for a more effective momentum transfer to take place, the flow should be laminar. Then knowing the flow mean velocity at the turbine inlet and also that in theory its rotor will start rotating, speeding up until reaching that tangential velocity when the relative velocity between the disks and the flow will be zero. From this moment and on, the only non zero velocity relative to the disks will be that of the fluid penetration velocity between consecutive disks and this is the velocity to be taken into account to calculate the design Reynolds Number. Well established theory [5] yields the laminar Darcy friction factor, f_{lam}, for laminar flows in ducts:

$$f_{lam} = \frac{64 M}{cVd} \qquad (1)$$

where c is the flow density, V the relative flow velocity and M the flow dynamic coefficient of viscosity. Or,

$$f_{lam} = \frac{64}{Re_d} \qquad (2)$$

where Re_d is the Reynolds number based upon the duct diameter as characteristic length.
The proper characteristic distance for the flow between consecutive disks can be chosen from

$$D_h = \frac{4S}{P} \qquad (3)$$

where D_h is the hydraulic diameter, S is the flow cross section area and P its wet perimeter. Then, D_{eff}, the effective diameter, can be written as

$$D_{eff} = \frac{2}{3} D_h \qquad (4)$$

Theory offers reasonable precision when one makes use of the hydraulic diameter, being quite precise if the effective diameter is used. Using the consecutive disks separation distance, a, (i.e., the gap between them), one may write

$$D_h = \frac{4p D_{ext} a}{2(p D_{ext} + a)} \qquad (5)$$

where D_{ext} is the disk external (outer) diameter. Then, as $a << D_{ext}$, this yield

$$D_h = 2a \qquad (6)$$

and

$$D_{eff} = \frac{4}{3} a \qquad (7)$$

Therefore the Reynolds number can be written as

$$R_d = \frac{4}{3} \frac{cVa}{M} \qquad (8)$$

3.0 ESTIMATING BOUNDARY LAYER THICKNESSES AND TOTAL NUMBER OF DISKS

As it is well known, the boundary layer thicknesses for either turbulent or laminar flows can be estimated as follows:

3.1 Estimating the Turbulent Boundary Layer Thickness

The turbulent boundary layer thickness can be estimated using the following correlation [6],

$$\text{д} = 0.526\, r \left(\frac{\nu}{r^2 \text{щ}} \right)^{1/5} \tag{10}$$

where r is the local disk radius, ν is the working fluid kinematic viscosity coefficient and ω the angular velocity.

3.2 Estimating the Laminar Boundary Layer Thickness

Observe that the flow component in the radial direction in the disk periphery is laminar, the same happening to that flow component in the internal outlet. Due to conservation of mass, this takes place as the gases entered in this periphery between disks and left it through the area including the disk region from the axis radius to the extremity of the outlet center flow, all this taking place as the flow possessed only a component along this radius from the periphery towards the inner region.

From the moment the disks reach a uniform angular velocity and on, the only non-zero relative velocity is the one between each disk and the flow inside the passages between consecutive disks flowing towards their center until reaching their exit ports.

In this case the Reynolds number should be calculated taking into account the flow areas between consecutive disks, i.e., the external area, S_{ext} ,

$$S_{ext} = \pi\, D_{ext}\ (a) \tag{11}$$

and the internal area between them, S_{int} ,

$$S_{int} = \pi\, D_{int}\ (a) \tag{12}$$

where D_{int} is the disk diameter taken at the exit ports stations.

Imposing the flow to be laminar, one should use the following estimate of the laminar boundary layer thickness [6] so that, from its value one can estimate the proper disk separation distance,

$$\text{д} \approx 5 \sqrt{\frac{\nu\, l}{U}} \tag{13}$$

where ν is the working fluid kinematic viscosity coefficient, l is the outer disk radius less its internal radius for both Reynolds Numbers, each to be estimated for laminar regimes, and U is the inlet flow velocity. Notice that one should use a disk separation that should guarantee a laminar regime at the disk tip. The velocity, U_{int}, at the disk tip in the internal flow can be written as:

$$U_{int} = \text{щ} R_{int} = \frac{U}{R_{ext}} R_{int} \tag{14}$$

As the flow occurs between disks, a boundary layer is formed along each one of them so that the width between consecutive disks to guarantee laminar flow in the whole region must possess at most the thickness of both boundary layers.

3.3 Estimating the Total Number of Required Disks

To estimate the total number of disks for the turbine, the following procedure is suggested:

Assume that at a given internal circumference of the disk the flow Reynolds number is less than 2300 (i.e., at that region the flow is laminar) Then, writing the mass flow rate through a gap as

$$\dot{m} = cVpDa \tag{15}$$

or

$$Va = \frac{\dot{m}}{pcD} \tag{16}$$

Then the overall mass flow rate, $\dot{m}_T$, can be written as

$$Vna = \frac{\dot{m}_T}{pcD} \quad \text{or} \quad Va = \frac{\dot{m}_T}{n\, p\, cD} \tag{17a}$$

where n is the number of gaps between consecutive disks

Also:

$$Rey = \frac{4}{3}\frac{cVa}{\text{м}} \quad \text{or} \quad Va = \frac{3}{4}\frac{\text{м}Rey}{c} \tag{17b}$$

Using (17a) and (17b),

$$n = \frac{4}{3}\frac{\dot{m}}{\text{рм}DRey} \tag{18}$$

where n is the number of gaps between consecutive disks (the stator lateral surfaces included), so that the total required number of disks will be (n-1)
Recall that in the laminar regime (with the flow being assumed as nearly incompressible) the torque 2T, delivered by a disk wetted both sides can be estimated as [6]:

$$C_M = \frac{2T}{\frac{1}{2}\text{сщ}^2 R^5} \tag{19}$$

where

$$C_M = 3.87\,Rey^{-\frac{1}{2}} \tag{20}$$

and

$$Rey = \frac{R^2\text{щ}}{\nu} \tag{21}$$

Therefore the total torque can be written as

$$T_0 = 2(n-1)T \tag{22}$$

If the turbine is made to work as water pump or air compressor, then the volume flow rate which is pumped outwards as a result of the centrifuging action on one side of a disk of radius R can be written as

$$Q = 2pR\int_0^\infty u\,dz \tag{28}$$

where u is the speed of the fluid at station z from the axis surface, so that, in a laminar regime,

$$Q = 0.885p\,R^2\sqrt{\text{нщ}} = 0.885p\,R^3\,\text{щ}Rey^{-\frac{1}{2}} \tag{29}$$

which is the volume flow rate pumped by centrifuge action on one side of the disk. Now, for the free disk under turbulent regime, the viscous torque turbulent theory developed by Von Kármán [6] for a disk wetted on both sides was shown to be equal to:

$$T_0 = 0.073\,\text{с}^2R^5\left(\frac{\text{н}}{\text{щ}R^2}\right)^{\frac{1}{5}} \tag{24}$$

the generated Power , P_0 , will be

$$P_0 = \omega T_0 \tag{25}$$

the moment coefficient, C_M , becomes

$$C_M = 0.146\ Rey^{-1/5} \tag{26}$$

and the volume flow rate in the axial direction is given by

$$Q = 0.219\,R^3\text{щ}Rey^{-\frac{1}{5}} \tag{27}$$

in a turbulent regime

4.0 A Brief Discussion on an Encapsulated Disk Performance

The disk in a turbine or compressor rotates in a very narrow capsule in which the gap (i.e., the distance s between the disk surface and the capsule wall, σ being the disk outer radius clearance) is much smaller than the disk radius R. Therefore it is worth to investigate the behavior of a disk rotating inside a capsule.
These relations become very simple in a laminar flow regime, $Rey < 10^5$, with a very small gap. If this gap is smaller than the boundary layer thickness the change of the tangential speed across the gap is linear just like in the well-known Couette flow. Then τ, the shear stress at a distance r from the axis is given by

$$\text{ф} = \frac{r\,\text{щм}}{s} \tag{30}$$

and M, the viscous force torque acting on one side of the disk will be

$$M = 2\pi \int_0^R \tau r^2\, dr = \frac{\pi}{2}\frac{\omega\mu R^4}{s} \tag{31}$$

so that considering both sides, this yields

$$2M = \frac{\pi\omega\mu R^4}{s} \tag{32}$$

and the moment coefficient becomes

$$C_M = 2\pi \frac{R}{s}\frac{1}{Rey} \tag{33}$$

C. Schimieden, apud Schlichting [6], investigated the influence of the width of a lateral spacing of a disk in a cylindrical capsule, as shown in figure 1, assuming a very small Reynolds Number (i.e., in a creeping like motion). Then the Navier-Stokes Equations can be simplified and the solution for the moment coefficient can be written as

$$C_M = \frac{K}{R} \tag{34}$$

where the constant K depends on the dimensionless ratios s/R and σ/R.

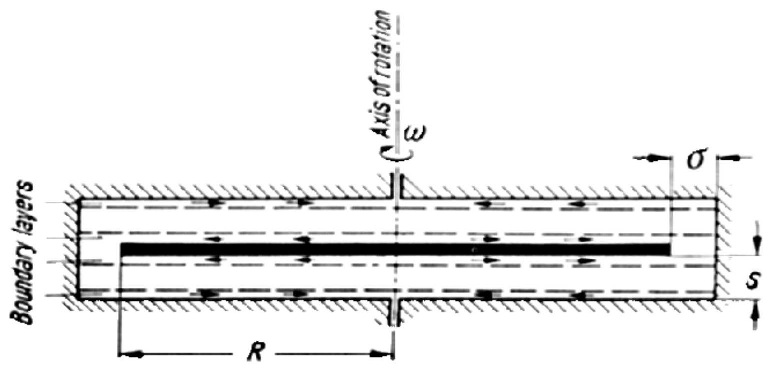

Figure 1 – Sketch of an encapsulated disk for symbols description
(taken from Reference [6])

6.0 CONCLUSIONS

This paper presented a very simple and straightforward technique, using basic fluid mechanics only, to estimate the needed number of disks required for a Tesla turbine, compressor or pump, to accomplish a prescribed job. It also reviews the encapsulated rotating disk performance. Notice that only well known and tried results and assumptions have been used here. However, the authors are not aware of a similar work having been done since the original Tesla wonderful invention. It is worth mentioning that these Tesla machines (i.e., the turbine, the compressor or the pump) are specially fitted to those instances where compact unities for generating electrical power or other equipment are required as in the case of isolated areas. Finally it should be noticed that, as a unique source of rotating motion, these Tesla machines can run under a very wide spectrum of not only fuels but also fluids in general, being useful, for example, as high speed drilling machines among many other applications.

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REFERENCES

[1] Cairns, W.M.J., *The Tesla Turbine*, Published by Camdem Miniature Steam services, GB, 2nd Edition, 2003
[2] Hedin, R.S. *The Tesla Turbine (Construction Plans)*, Live Steam, Nov. 1994
[3] Rice, W., *Tesla Turbomachinery*, Proceedings of the IV International Nikola Tesla Symposium, Sept. 1991
[4] Logan, E. Jr., Handbook of Turbomachinery, Arizona State U. Marcel Dekker Inc., Phoenix, 1995
[5] White, F.M., Fluid Mechanics, McGraw-Hill Kogacucha, Ltd. Kosaido Publishing Co., Tokyo, Japan, 1979, p-325
[6] Schlichting, H., Boundary Layer Theory, McGraw-Hill Book Co., Inc., New York, NY, fourth Ed.,1962, p-547.