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THE DESIGN AND CONSTRUCTION OF A MULTIPLE DISK TURBINE

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THEORY

Design data for the turbine came from the extensive work of Professor Warren Rice of Arizona State University, Tempe, Arizona. The mathematical analysis of the fluid flow, and computer analysis, have been used to develop the pertinent data for a pump, water turbine and steam turbine. The fluid for our turbine is considered Newtonian, incompressible and the laminar flow of most practical interest.

The essential part of the fluid flow is between a pair of co-rotating disks. The momentum of the moving fluid is transferred to disks due to the effects of viscosity and adhesion. The friction force of the fluid transfers this momentum. Low-pressure, low-velocity fluid is exited out of the center (Fig. 1).

The turbine constructed consists of 30 disks and spacers. Each disk being 12 inches in diameter with an exit port of 3.6 inches. The metal disks are of galvanized steel .020 inches thick and the spacers are made of steel shim stock .010 inches thick. The turbine also consists of a fluid supply nozzle, exit ports, rotor housing, casing and shaft bearings.

The high pressure fluid is converted to high velocity fluid at the nozzle. The moving fluid flows between a pair of co-rotating disks. Due to the effects of viscosity and adhesion, there is a force component in the direction of the direction of the fluid, tangentially, and an outward centripical force due to the rotation of the disks. The pressure gradient between the inlet and outlet ports forces the fluid to spiral toward the center exit port. The combination of these forces creates the net torque on the rotor.

INTRODUCTION

The conception and development of multiple-disk turbomachinery is credited to Nikola Tesla. Patents covering the multiple disk turbine and pump were granted to him in May of 1913. These devices were widely discussed in the semi-technical press of that time.

In 1972 Walter Baumgartner built an experimental model turbine which ran on compressed air and produced 30 horsepower at 18,000 rpm.

Presently the Tesla turbine is under development by Sun Wind, Ltd., of Sebastopol, California. Sun Wind plans to use the turbine burning hydrogen in a three wheel car. The turbine can also burn propane, vegehol, and gasoline. Another California company, General Enertech of San Diego, is building and selling the Tesla pump which has been improved and modernized.

Extensive theoretical analysis of the Tesla turbine has been carried out over the past 20 years by Prof. Warren Rice of Arizona State University in Tempe, Arizona. His paper "Calculated Design Data for the Multiple Disk Turbine Using Incompressible Fluid" was used extensively in our turbine model.

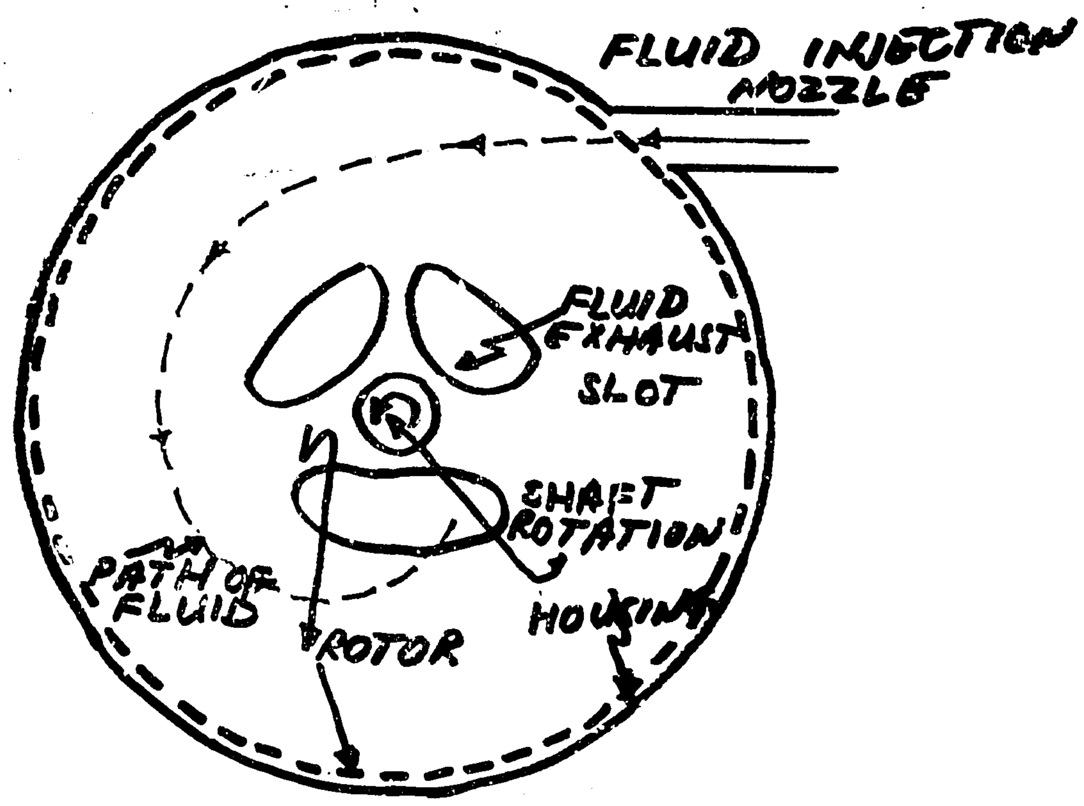
RESULTS

The turbine was tested using a supply pressure which varied from 0 - 50 psi. The maximum rpm's obtained was 220. The power output was calculated to be approximately .05 horsepower. Different values of pressure were tried and at each value rpm readings were taken.

CONCLUSION

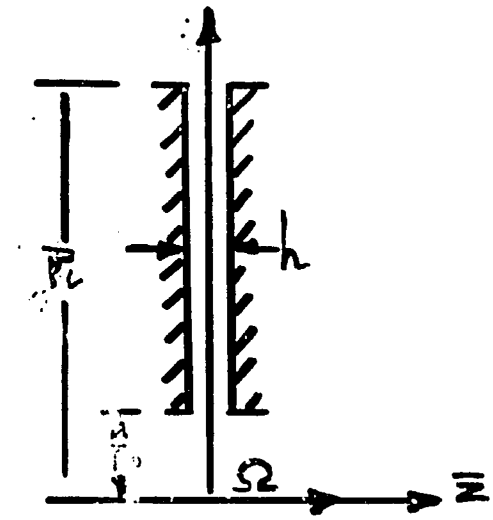
The turbine did not perform as well as expected due to several reasons. The rotor was not perfectly round and rubbed at one spot in the housing. The high perssure ring seals caused too much friction and had to be removed, therefore, a good pressure gradient could not be obtained. The disks were slightly warped in spots closing off some of the passage ways thereby stopping fluid flow bet· the pair of disks.

Despite these problems, this turbine does have advantages in ease of production, low cost, and low maintenance.



Schematic diagram of multiple-disk turbine

(FIG#1)



Coordinate system and nomenclature for turbine flow

The turbine efficiency is given by

$$\eta = \frac{T_1}{2\pi U_0 P_i}$$

T_1 = torque applied by fluid between one pair of disk (F/L)

U_0 = volume flow rate parameter $= \dfrac{Q_1}{2\pi \bar{r}_0 \Omega h}$

Q_1 = volume flow rate between pair of adjacent disk

$\bar{r}_0$ =

Ω = angular velocity of rotor.

h = spacing between adjacent disks

P_i =

Calculated Design Data for the Multiple-Disk Turbine Using Incompressible Fluid

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Earlier analyses of the laminar radially inward throughflow of Newtonian incompressible fluid between parallel corotating disks have been used to calculate the performance of multiple-disk turbines using such flow passages as the rotor. Such turbines are characterized by certain dimensionless parameters and a large number of computerized calculations has enabled preparation of turbine performance maps for turbines idealized as having no losses external to the rotor (except for assumed zero pressure recovery in the turbine exhaust). These maps show the quantitative dependence of turbine efficiency, total pressure and delivered power on the turbine geometry and speed, the turbine nozzle direction and pressure drop, and on the fluid properties; full admission around the periphery of the rotor is assumed. Conventional loss information for the nozzles, and conventional bearing, seal and "disk friction" loss information, must be applied in the design process to provide prediction of actual turbine performance and comparison with conventional turbines.

Introduction

The conception of multiple-disk turbomachinery is credited to N. Tesla early in this century [1-3][1], and the devices were widely discussed in the semitechnical press of that time [4-7]. Subsequently, such turbomachinery received scant attention for many years but the list of references included in this paper indicates the extent of a revival of interest. A number of experimental and analytical feasibility studies have been reported for the turbine configuration of multiple-disk turbomachinery [8-15]. The papers reporting on these studies make the operating principle clear and indicate that laminar flow of fluid in the rotor, rather than turbulent flow, is of the most practical interest.

A recent sequence of analytical contributions has made possible calculation of the performance of multiple-disk turbines, according to an idealized model, provided that the fluid is Newtonian and incompressible and that the flow is laminar. The essential part of the flow in a multiple-disk turbine is that between a pair of corotating disks; the turbine consists of a number of such flows in parallel, together with fluid supply nozzles, an exit port and/or diffuser, and a suitable housing and bearings. The laminar flow of fluid between a pair of disks consists of an entrance region in the outer radial zone near the nozzles and an asymptotic flow region at inner radii remote from the nozzles. Peube and Kreith [16] provided a truncated series type analysis for the asymptotic region of radially outward flow between corotating disks, descriptive of the flow in a multiple-disk pump. The results are easily adapted to the asymptotic region for radially inward flow, corresponding with a multiple-disk turbine, but fail to describe the flow at small radii with sufficient accuracy, particularly with regard to the pressure drop suffered by the flow. An iterative solution for the asymptotic region of radially inward flow between corotating disks, corresponding with a multiple-disk turbine, by Matsch and Rice [17], provided a more definite (but still limited) flow description; the paper referenced details the importance of the concept of the asymptotic flow region and of the progression of the velocity profiles to a final inflection of the radial component of the velocity at a predictable inner radius. Two parameters are necessary to specify an asymptotic flow case. One is a so-called Reynolds number N_{RE} which is dimensionless and depends on fluid properties, angular velocity of the disks, and the spacing between the disks. The second is a flow rate parameter U_e which is dimensionless and depends on the volume flow rate of fluid between a pair of disks, the angular velocity of the disks, and the outer radius of the rotor. It is clear from the referenced papers that knowledge of the asymptotic region alone is not sufficient to determine the performance of a multiple-disk turbine, since large pressure changes and contributions to the torque occur in the entrance region before the asymptotic flow region is reached.

[1]Numbers in brackets designate References at end of paper.

Contributed by the Fluids Engineering Division of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS and presented at the Joint Fluids Engineering & CSME Conference, Montreal, Quebec, Canada, May 13-15, 1974. Manuscript received at ASME Headquarters February 7, 1974. Paper No. 74-FE-9.

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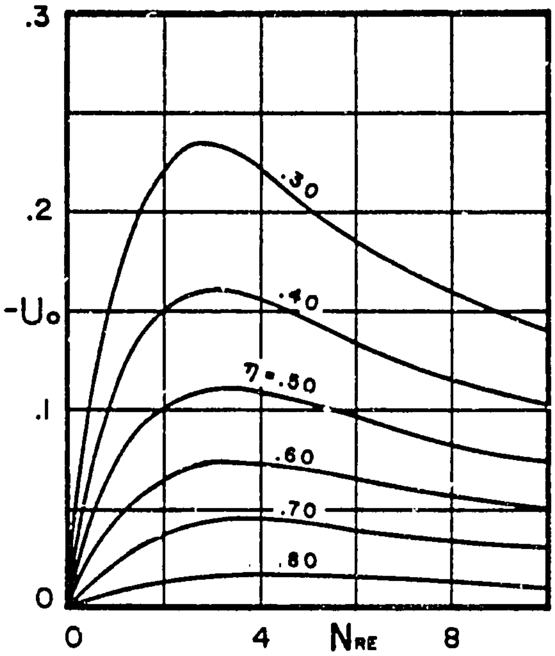

Fig. 1 Constant efficiency lines on N_{RE}, U_o coordinates for $V_o = 1.1$, $r_i = 0.3$ and parabolic inlet velocity distributions

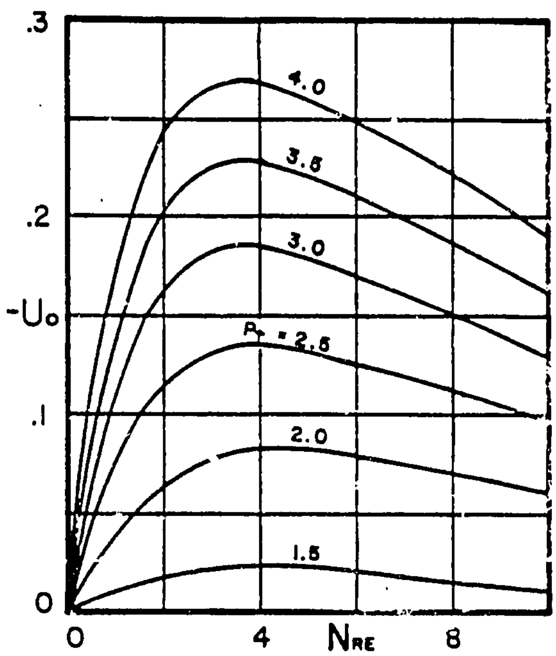

Fig. 2 Constant dimensionless total pressure lines on N_{RE}, U_o coordinates for $V_o = 1.1$, $r_i = 0.3$ and parabolic inlet distributions

A numerical (finite-difference) scheme for calculating the entrance flow between corotating disks, for radially outward flow corresponding with a multiple-disk pump was presented by Breiter and Pohlhausen [18]. A similar method of calculation was applied for radially inward flow, corresponding with the multiple-disk turbine, by Boyd and Rice [19]; the fluid was considered admitted uniformly around the outer periphery of the disks through nozzles providing tangential, radial, and axial components of velocity. In [19], the disk-to-disk spatial distributions of the velocity components at the outer radius $\bar{r}_o$ (provided by the nozzles to the rotor) are considered arbitrary but specified, and these are additional parameters for the specification of a case of flow in the entrance region between the disks.

Furthermore, [19] introduces a simplification in the parameterization; with the general shape of the disk-to-disk velocity component profiles specified (uniform, parabolic, etc.), then the three parameters N_{RE}, U_o and V_o are sufficient to characterize a case of entrance region flow. Here V_o is the average value of

Nomenclature

Dimensional Quantities

H = operating head for turbine, length

h = spacing between adjacent disks, length

$\bar{p}_n$ = pressure change in nozzles, force/(length)^2

$\bar{p}_r$ = pressure change through rotor, force/(length)^2

$\bar{p}_t$ = stagnation pressure change through turbine, force/(length)^2

P = power delivered to turbine rotor, force-length/time

Q = volume flow through entire turbine, (length)^3/time

Q_1 = volume flow between pair of adjacent disks, (length)^3/time

$\bar{r}$ = radial coordinate, length

$\bar{T}_1$ = torque applied by fluid between one pair of disks, force-length

$\bar{u}$ = $\bar{u}(\bar{r}, \bar{z})$ = radial component of velocity distribution, length/time

$\bar{U}_o = \frac{2}{h}\int_0^{h/2} \bar{u}_o(\bar{z})d\bar{z} = \frac{Q_1}{2\pi\bar{r}_o h}$, average value of radial component of velocity at rotor entrance (nozzle exit), length/time

$\bar{v}$ = $\bar{v}(\bar{r}, \bar{z})$ = tangential component of velocity distribution, length/time

$\bar{V}_o = \frac{2}{h}\int_0^{h/2} \bar{v}_o(\bar{z})d\bar{z}$ = average value of tangential component of velocity at rotor entrance (nozzle exit), length/time

$\bar{w}(\bar{r}, \bar{z})$ = axial component of velocity distribution, length/time

$\bar{z}$ axial space coordinate, length

ρ = density of fluid, mass/(length)^3

μ = viscosity of fluid, force-time/(length)^2

Ω = angular velocity of rotor, rad/(time)

Dimensionless Quantities

N = number of spaces between disks, in set of disks composing a turbine rotor

$N_{RE} = \frac{\rho\Omega h^2}{\mu}$, Reynolds number

$p_n = \bar{p}_n/\rho\Omega^2\bar{r}_o^2$

$p_r = \bar{p}_r/\rho\Omega^2\bar{r}_o^2$

$p_t = \bar{p}_t/\rho\Omega^2\bar{r}_o^2$

$r = \bar{r}/\bar{r}_o$

$T_1 = \bar{T}_1/\rho\bar{r}_o^4h\Omega^2$

$u = u(r, z) = \bar{u}(\bar{r}, \bar{z})/\Omega\bar{r}_o$

$U_o = \bar{U}_o/\Omega\bar{r}_o = \frac{Q_1}{2\pi\bar{r}_o^2\Omega h}$ = volume flow rate parameter

$v = v(r, z) = \bar{v}(\bar{r}, \bar{z})/\Omega\bar{r}_o$

$w = w(r, z) = \bar{w}(\bar{r}, \bar{z})/\Omega\bar{r}_o$

$V_o = \bar{V}_o/\Omega\bar{r}_o$ = tangential velocity parameter

$z = \bar{z}/h$

$\eta = \frac{T_1}{2\pi U_o p_t}$ = efficiency

Subscripts

i = inner (exhaust) radial station

o = outer (nozzle exit, rotor entrance) radial station

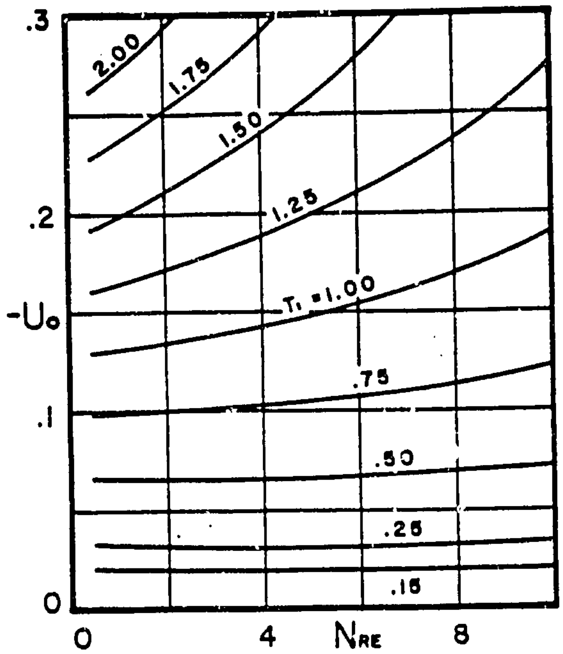

Fig. 3 Constant dimensionless torque lines on N_{RE}, U_o coordinates for V_t=1.1, r_i=0.3 and parabolic inlet velocity distributions

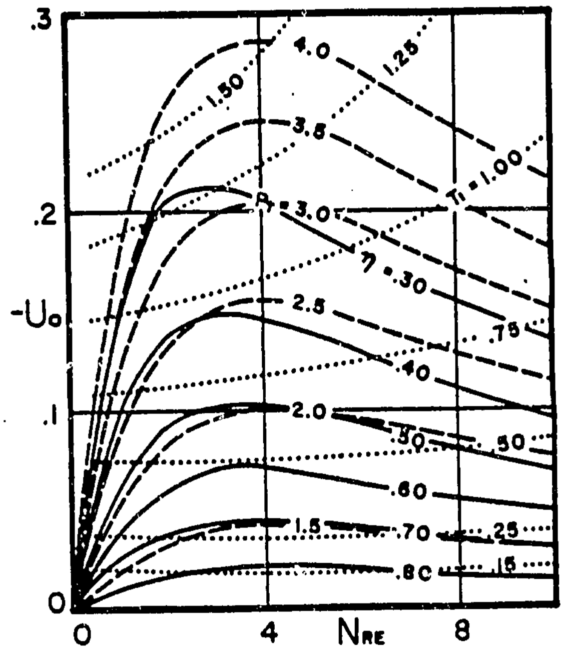

Fig. 5 Performance data for V_t=1.0, r_i=0.3 and parabolic inlet velocity distributions

the dimensionless tangential component of the velocity of the flow supplied by the nozzles to the rotor, corresponding with the "fluid-to-blade velocity ratio" of conventional turbine practice. The calculation procedure detailed in [19] also is applicable for the asymptotic flow region and overcomes the limitation in accuracy in the earlier methods of calculating that region. The disadvantage of the method is the excessively lengthy computation time required by digital computer.

The adequacy of the calculation model and of the numerical computations was confirmed experimentally by Adams and Rice [20].

Boyack and Rice [21] developed an integral method for the flow model used by previous investigators and a computer program which executes extremely rapidly. The method yields results essentially identical with those of the lengthy computations of references [18, 19] and therefore also confirmed experimentally in reference [20]. Use of this computer program enabled calculation of the very large number of flow cases necessary to obtain the design data presented herein. Crawford [22] improved the ability of the method of Boyack and Rice to handle uniform

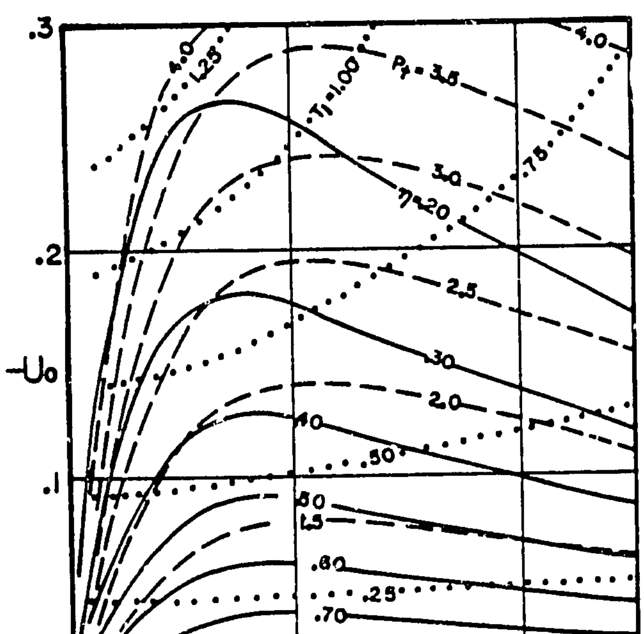

Fig. 4 Performance data for V_t=0.8, r_i=0.3 and parabolic inlet velocity distributions

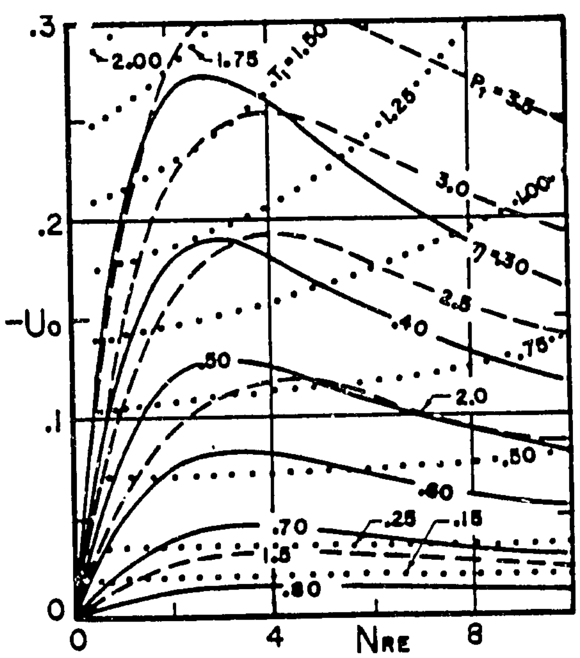

Fig. 6 Performance data for V_t=1.1, r_i=0.4 and parabolic inlet velocity distributions

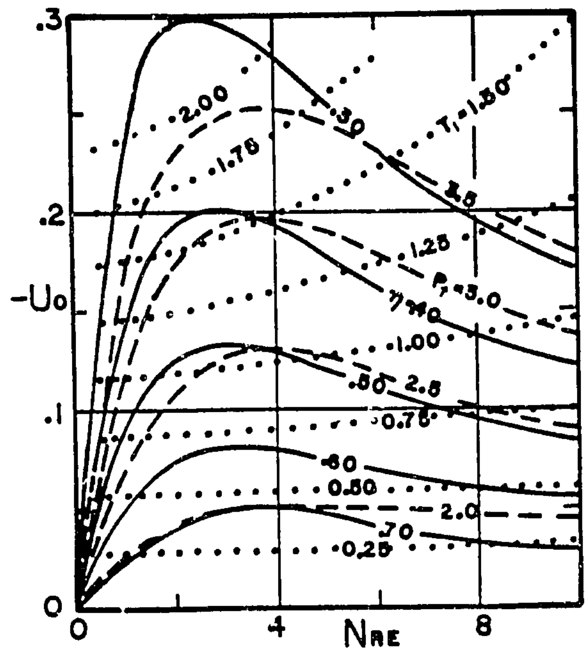

Fig. 7 Performance data for $V_t = 1.3$, $r_i = 0.4$ and parabolic inlet velocity distributions

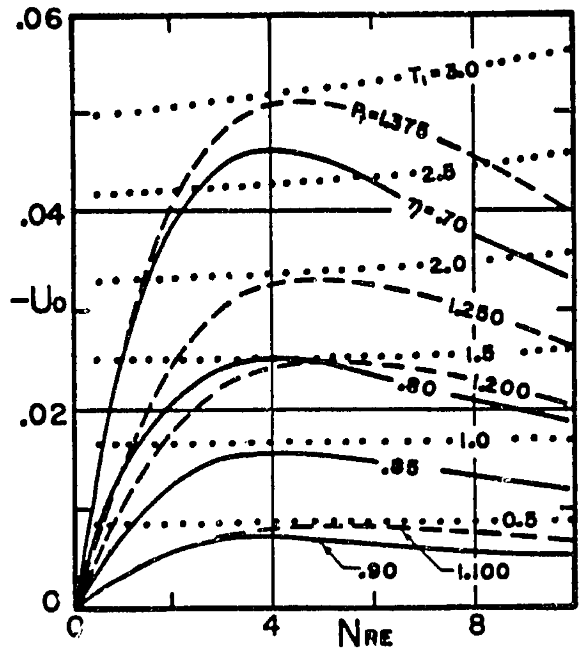

Fig. 8 Performance data for $V_t = 1.1$, $r_i = 0.3$ and uniform inlet velocity profiles

inlet profiles, exhaustively examined the correctness of results in the entrance region, and also further decreased the execution time required for digital computation by improved programming.

Model of Turbine for Calculations

The mathematical model and calculation procedure for the flow between a pair of corotating disks is that used by Boyack and Rice [21] discussed above. The multiple-disk turbine consists of many such corotating disks arranged on a common shaft with the flow passages in parallel. The choice of the number of disks is determined by individual design requirements; the design information presented herein is based on the flow between a single pair of corotating disks. The flow is considered to be a laminar flow of an incompressible Newtonian fluid. The calculations provide realistically for the thermodynamically irreversible flow effects in the space between the disks. The pressure change in the flow between the disks, p_r, is composed of that due to radially-incurred area change (with consequent velocity change) and that due to irreversible (viscous) effects.

The dimensionless parameters N_{RE}, U_e and V_e for the flow between a pair of corotating disks carry over to become also the independent parameters for the turbine model. The turbine is assumed provided with nozzles around the entire periphery of the rotor (full admission). The nozzles provide the specified values of U_e and V_e to the space between a pair of disks from a plenum chamber having the dimensionless stagnation pressure p_t. The stagnation pressure is assumed conserved in the flow in the nozzles in the turbine model, that is, the flow in the nozzles is considered to be thermodynamically reversible. Part of p_t is the dimensionless pressure drop in the reversible nozzles, p_n, given by

$$p_n = \frac{1}{U_e}\int_0^{1/\beta} [u_e^2(z) + v_e^2(z)]u_e(z)dz \qquad (1)$$

where the contribution due to $w_e(z)$ is considered to be negligibly small. It is seen that p_n depends on the disk-to-disk distribution of the velocity components.

The fluid can be supplied to the rotor of a multiple-disk turbine through nozzles arranged in various ways. On the one hand, a separate nozzle ring may be provided for each disk space; in this case the disk-to-disk distribution of the velocity components $u_e(z)$ and $v_e(z)$ will be approximately parabolic. On the other hand, the nozzle ring may consist of slots extending across the rotor or at least across several disk spaces; in this case, disk-to-disk distribution of the velocity components $u_e(z)$ and $v_e(z)$ will be approximately uniform.

As an idealization considered of interest in practice, the distribution of $u_e(z)$ and $v_e(z)$ assumed herein is parabolic generally; only where specifically noted later, the distribution is assumed uniform in order to demonstrate the effect of the distribution on the performance of a multiple-disk turbine. However, in applications the disk spacing frequently is so small that separate nozzles are impractical and a single admission slot must be used. The uniform distribution is then a better idealization of the supply flow.

In the turbine model, the fluid is considered to exhaust from the rotor into a stagnant exit plenum chamber, without the benefit of an exahust diffuser. Thus, with the nozzles considered to be supplied with fluid from a stagnant condition,

$$p_t = p_n + p_r \qquad (2)$$

and p_t is the dimensionless stagnation pressure difference required across the turbine. The dimensional quantity p_t/ρ is, then, the available energy of the fluid per unit mass and is the proper denominator for the expression of turbine efficiency, η.

The torque delivered to the rotor by the fluid can be calculated by two fundamentally different means; both involve use of the calculated velocity field between the disks. As one possibility, the local shear stress over the surface of the disks can be derived from the velocity field and fluid properties and hence the local torque due to the shear force can be integrated over the surface of the disks to obtain the rotor torque for the flow between a pair of disks. As an alternative, a finite control volume can be used between the entrance and exit radii and the torque can be calculated from the change of the moment of momentum of the fluid between those radial stations. The latter procedure is used here since it is more direct and accurate mainly because it avoids differentiation of the fluid velocity profiles. The dimen-

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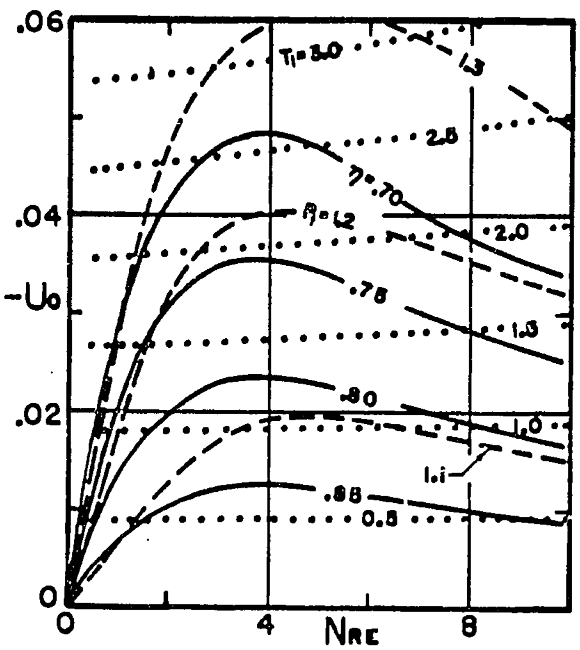

Fig. 9 Performance data for $V_o=1.1$, $r_i=0.4$ and uniform inlet velocity profiles

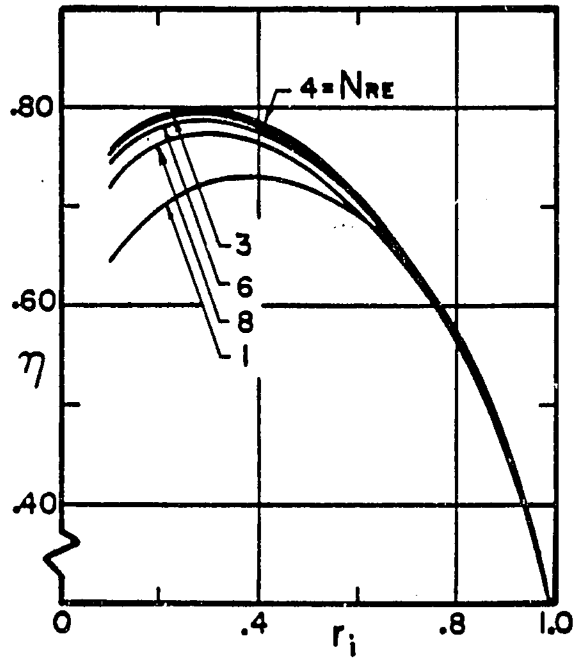

Fig. 10 Turbine efficiency as a function of r_i with N_{RE} as a parameter, for $V_o=1.1$, $U_o=-0.02$, and parabolic inlet velocity distributions

sionless torque T_1 (for one space between a single pair of disks) is also the dimensionless work delivered to the rotor by the fluid and is given by

$$T_1 = 4\pi\left[r_o^2\int_0^{1/2} u(r_o, z)v(r_o, z)dz - \int_0^{1/2} u_i(z)v_i(z)dz\right]. \quad (3)$$

Hence the turbine efficiency is given by

$$\eta = \frac{T_1}{2\pi U_o p_i}. \quad (4)$$

The computer program used to implement the analysis of Boyack and Rice [21] was arranged to accept N_{RE}, U_o and V_o as input parameters with the shape of $u_o(z)$ and $v_o(z)$ specified, and to calculate p_i, T_1 and η as output results for selected dimensionless exit radii r_i such that $1 \geq r_i \geq 0.05$ with $\Delta r_i = 0.05$. Thus each combination of input parameters corresponds with an infinity of dimensionless turbines having different exit radii; furthermore, each combination of N_{RE}, U_o, V_o and r_i correspond with an infinity of dimensional (actual) turbines.

The turbine model does not allow for resisting torques acting on the turbine rotor due to rotor-to-housing "disk friction," bearings, seals, etc. Furthermore, the model ignores pressure losses that will occur in an actual turbine due to interference of the disk edges with the nozzle exit flow and due to irreversibility in the nozzle flow, and due to exit passage area changes and frictional losses. (The latter "loss" may actually be a "gain" if the exit passage can be designed to produce a pressure recovery from the kinetic energy exhausting from the rotor, since the turbine model assumes no such pressure recovery.)

The design information presented thus far, therefore, accounts for losses only in the flow between the disks and leaves to the designer the taks of accounting for the loss of torque and efficiency due to nozzle efficiency, nozzle-rotor interference, rotor-to-housing "disk friction," bearings, and exhaust passage pressure change. The pressure, efficiency, and torque values given herein are therefore upper limits of the performance of actual multiple-disk turbines and must be used with conventional loss information for the turbine parts external to the rotor to calculate the performance of actual turbines. A design example given later herein includes references for such loss information.

Results—Design Information Summary

The computer program was executed for a very large number of combinations of N_{RE}, U_o and V_o, each yielding values of p_i, T_1 and η for values of r_i such that $1 \geq r_i \geq 0.05$. The results were cross-plotted by hand in order to produce data for plotting lines of constant p_i, T_1 and η for a fixed value of r_i and of one input parameter, and using the remaining two input parameters as coordinates. A large amount of judgment was exercised in making additional computer program executions to "fill in"

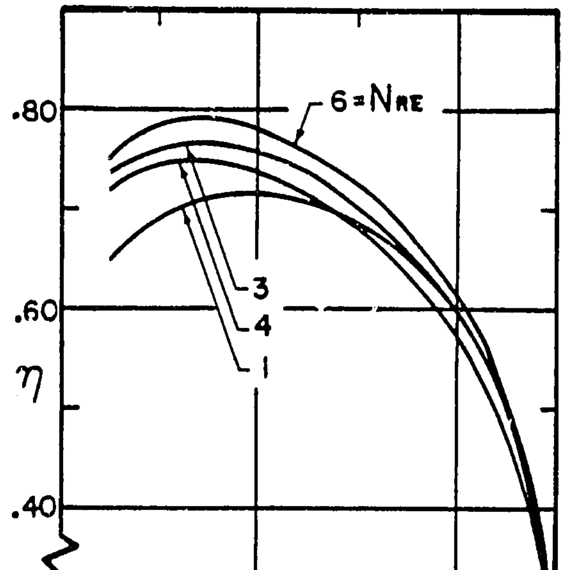

Fig. 11 Turbine efficiency as a function of r_i with N_{RE} as a parameter, for $V_o=1.3$, $U_o=-0.02$ and parabolic inlet velocity distributions

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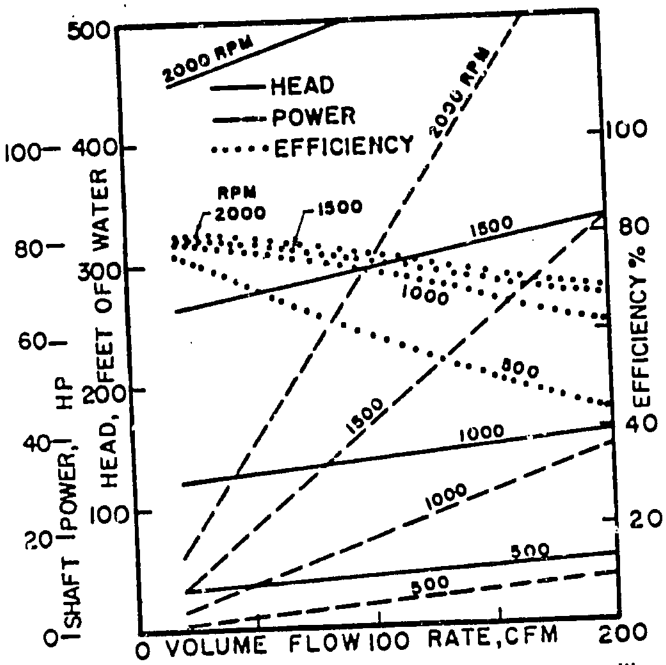

Fig. 12 Calculated performance for an actual water turbine with a rotor dia of 12 in.

the crossplots with definite additional data points wherever rates of change were found to be large during the crossplotting process.

Typical results are shown in Figs. 1, 2 and 3 which give lines of constant p_i, T_1 and η using U_o, N_{RE} coordinates and for $V_o = 1.1$ and $r_i = 0.3$ and for parabolic velocity profiles at the rotor entrance (nozzle exit). From a very large number of plots such as those of Figs. 1, 2 and 3, dimensionless performance maps were produced.

The dimensionless performance maps are voluminous beyond the possibility of journal publication. They are available in the form of a University Report [23]. The results given therein are in four groups, arranged according to four values of the parameter V_o : 0.8, 1.0, 1.1 and 1.3. This is thought to be the useful and interesting range for V_o with the values of most interest being near and just exceeding unity. The data for each value of V_o have r_i as a parameter and are presented in the form of sets of lines for constant p_i, T_1 and η on U_o, N_{RE} coordinates.

Figs. 4–7 present typical samples of the results referenced above, for the case in which the turbines have parabolically shaped disk-to-disk velocity components at the rotor entrance (nozzle exit). The Report [23] also gives voluminous results in the same form for turbines having uniform disk-to-disk velocity components at the rotor entrance. Typical samples of these results are presented in Figs. 8 and 9. Comparison of the various results yields information concerning the effect of the shape of the entrance profiles on turbine performance. The obtaining of the results in the Report [23] and of similar results for other parameter values is expensive in terms of digital computer usage but once done, eliminates the need for further analysis of the flow within the rotor of a multiple-disk turbine; only very modest algebraic calculations are needed to apply the results to the design of multiple-disk turbines.

Discussion

Some insights into multiple-disk turbine design characteristics are afforded through crossplotting of the performance results. Fig. 10 presents the dependency of turbine efficiency on the dimensionless inner (exhaust) radius r_i, with N_{RE} as a parameter, for $V_o = 1.1$ and $U_o = -0.02$. (These are reasonable and interesting values for these parameters.) The optimum inner radius is approximately 0.3 for $3 \leq N_{RE}$ and is approximately 0.4 for $N_{RE} = 1$; the figures also shows that the greatest efficiency occurs for $N_{RE} = 4$ for this parameter combination. An increased value of the volume flow rate parameter reduces the efficiency at all values of N_{RE} and increases the value of r_i for maximum efficiency.

The effect on efficiency of an increased value of V_o is detailed in Fig. 11, for $U_o = -0.02$. The maximum efficiency is somewhat decreased by the higher value of V_o but the turbine characteristics are generally unchanged.

Further details concerning these matters are contained in the Report [23]. For $U_o = -0.02$ and $N_{RE} = 4$, maximum efficiency occurs near $V_o = 1.0$ and the effect of increasing r_i above 0.3 is to reduce efficiency and to cause the local maximum efficiency to occur at higher values of V_o. Furthermore, both T_1 and p_i are almost-linear functions of V_o, increasing as r_i is decreased. Throughout the parameter range, changing the N_{RE} values changes the turbine performance quantitatively but all trends remain similar to those for $N_{RE} = 4$.

It is necessary to realize that the results pertain to the flow between a single pair of disks; for a turbine with N spaces between disks, $((N + 1)$ disks), the total flow rate, Q, will be

$$Q = NQ_1 = NU_o 2\pi r_o^2 \Omega h \tag{5}$$

and the power delivered to the rotor by the fluid will be

$$P = NT_1 \rho r_o^4 h \Omega^3 \tag{6}$$

which can be found alternatively from the equation

$$P = \eta Q p_i. \tag{7}$$

Both equations (6) and (7) refer to the power delivered to the rotor by the fluid; the shaft power will be less because of rotor-to-housing "disk friction," bearing losses, seal losses, etc. Also, for an actual turbine, the stagnation pressure required will be more than p_i becasue of irreversible pressure changes in the fluid supply nozzles and between the nozzles and the spaces between the disks. Consequently, the actual turbine efficiency will be somewhat less than that indicated in the figures. The designer will allow for these facts in the use of the design data to predict turbine performance.

There are numerous sources of information of value in estimating the various losses. General design features and the performance of bearings, seals, and nozzles are reviewed in references [24, 25]. Daly and Nece [26] give sufficient detail concerning the rotor-to-housing "disk friction" to allow optimization of the clearance and estimation of the power loss from that source.

An Example of Turbine Design and Performance

Using the reference material earlier cited and the data in the figures, a very simple digital computer program can be used to determine the physical dimensions of turbines and the performance characteristics. As an example, it is assumed that a water turbine is desired which will produce 12 shaft horsepower when operated at 1000 rpm with a total head of 130 ft of water. A suitable design point is chosen as a compromise between efficiency, physical size, and ease of executing design details; the design point is somewhat arbitrary and several should be investigated by the designer. In this case, the point chosen is $U_o = -0.02$, $N_{RE} = 4$ and $r_i = 0.30$. It is further assumed that the turbine will use well-designed nozzles to supply fluid to the rotor with an approximately parabolic disk-to-disk distribution of velocity, carbon ring seals, and conventional bearings, and will have the optimum casing-to-rotor clearance for least "disk friction."

The foregoing assumptions and choices result in a turbine having a rotor dia of 12.0 in., a rotor exhaust dia of 3.6 in., and

479 disks spaced 0.0083 in. apart. If a disk thickness of 0.004 in. is arbitrarily chosen, this results in a rotor with an axial length of 5.89 in.; the total volume of the turbine is approximately 1200 cu in. including the turbine casing and other structure.

The performance characteristics are shown in Fig. 12.

The very small disk thickness used in this design example is realistic; each disk carries very small torsional and radial loading and no axial loading. The disks can be spaced correctly and the rotor made more rigid by dimpling the disks to achieve the required average disk spacing.

In this example, the maximum turbine efficiency is approximately 81 percent and occurs at lowest flow rate and highest speed. This value of efficiency is by no means the maximum achievable in multiple-disk turbines. A different choice of the design point or of speed, power or head at the design point, will result in a different set of performance curves and characteristics.

As an extension of the foregoing example it is considered that the turbine is to be designed to use a light oil (ρ = 54 lb_m/ft^3, ν = 0.0001 ft^2/sec) rather than water (ρ = 62.3 lb_m/ft^3, ν = 0.0000106 ft^2/sec). All other specifications being the same, the oil turbine requires 197 disks and has a spacing between the disks of 0.02345 in. which results in a rotor length of 5.41 in., and an efficiency of 0.76 and volume flow rate of 75 cfm at 1000 rpm and 130 ft head. The computerized evaluation program would easily generate the off-design point characteristics of this turbine also.

In contrast with conventional turbines, the performance of the rotor and hence good estimates of the turbine characteristics can be realistically calculated for multiple-disk turbines for both design point and off-design point operation, with ease using only algebraic calculations.

Conclusion

While the design data presented herein are sufficient to allow calculation of the performance of multiple-disk turbines and comparison with conventional turbines, the designer will finally want more detail of the flow field between the disks. At such point of interest, the designer should use references [21, 22] to obtain a computer-oriented analysis which easily and economically yields the desired detailed flow information.

Further details concerning the generation of the cross-plots and of design procedures are given in a thesis by Lawn [27] as well as the Report [23].

The multiple-disk turbine can be designed with high efficiency for fluid with any viscosity and density; it remains to be established by further investigation and by practice whether or not the turbine has substantial advantages when compared with conventional turbines, in the various extremes of application. For all of the data presented herein, it is assumed that the flow in the rotor is laminar. It is known from investigation by Adams and Rice [20] that laminar flow occurs in the flow between rotating disks over a wide range of the flow parameters. The criteria for prediction of laminar and of turbulent flow has been established for multiple-disk turbines; it is known at this time that laminar flow persists to high values of N_{RE} for low values of U_o, and to high values of U_o for low values of N_{RE}; high values of both parameters promote turbulent flow. Details of the delineation of the laminar region and of the experiments leading to it are available in a University Report [28] and will appear subsequently in a journal publication.

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UNITED STATES PATENT OFFICE.

NIKOLA TESLA, OF NEW YORK, N. Y.

FLUID PROPULSION.

1,061,142. Specification of Letters Patent. Patented May 6, 1913.

Application filed October 21, 1909. Serial No. 523,832.

To all whom it may concern:

Be it known that I, NIKOLA TESLA, a citizen of the United States, residing at New York, in the county and State of New York, have invented certain new and useful Improvements in Fluid Propulsion, of which the following is a full, clear, and exact description.

In the practical application of mechanical power based on the use of a fluid as the vehicle of energy, it has been demonstrated that, in order to attain the highest economy, the changes in velocity and direction of movement of the fluid should be as gradual as possible. In the present forms of such apparatus more or less sudden changes, shocks and vibrations are unavoidable. Besides, the employment of the usual devices for imparting energy to a fluid, as pistons, paddles, vanes and blades, necessarily introduces numerous defects and limitations and adds to the complication, cost of production and maintenance of the machine.

The object of my present invention is to overcome these deficiencies in apparatus designed for the propulsion of fluids and to effect thereby the transmission and transformation of mechanical energy through the agency of fluids in a more perfect manner, and by means simpler and more economical than those heretofore employed. I accomplish this by causing the propelled fluid to move in natural paths or stream lines of least resistance, free from constraint and disturbance such as occasioned by vanes or kindred devices, and to change its velocity and direction of movement by imperceptible degrees, thus avoiding the losses due to sudden variations while the fluid is receiving energy.

It is well known that a fluid possesses, among others, two salient properties: adhesion and viscosity. Owing to these a body propelled through such a medium encounters a peculiar impediment known as "lateral" or "skin resistance," which is twofold: one arising from the shock of the fluid against the asperities of the solid substance, the other from internal forces opposing molecular separation. As an inevitable consequence, a certain amount of the fluid is dragged along by the moving body. Conversely, if the body be placed in a fluid in motion, for the same reasons, it is impelled in the direction of movement. These effects, in themselves, are of daily observation, but I believe that I am the first to apply them in a practical and economical manner for imparting energy to or deriving it from a fluid.

The subject of this application is an invention pertaining to the art of imparting energy to fluids, and I shall now proceed to describe its nature and the principles of construction of the apparatus which I have devised for carrying it out by reference to the accompanying drawings which illustrate an operative and efficient embodiment of the same.

Figure 1 is a partial end view, and Fig. 2 is a vertical cross section of a pump or compressor constructed and adapted to be operated in accordance with my invention.

In these drawings the device illustrated contains a runner composed of a plurality of flat rigid disks 1 of a suitable diameter, keyed to a shaft 2, and held in position by a threaded nut 3, a shoulder 4 and washers 5, of the requisite thickness. Each disk has a number of central openings 6, the solid portions between which form spokes 7, preferably curved, as shown, for the purpose of reducing the loss of energy due to the impact of the fluid. The runner is mounted in a two part volute casing 8, having stuffing boxes 9, and inlets 10 leading to its central portion. In addition a gradually widening and rounding outlet 11 is provided, formed with a flange for connection to a pipe as usual. The casing 8 rests upon a base 12, shown only in part, and supporting the bearings for the shaft 2, which, being of ordinary construction, are omitted from the drawings.

An understanding of the principle embodied in this device will be gained from the following description of its mode of operation. Power being applied to the shaft and the runner set in rotation in the direction of the solid arrow the fluid by reason of its properties of adherence and viscosity, upon entering through the inlets 10 and coming in contact with the disks 1 is taken hold of by the same and subjected to two forces, one acting tangentially in the direction of rotation, and the other radially outward. The combined effect of these tangential and centrifugal forces is to propel the fluid with continuously increasing velocity in a spiral path until it reaches the outlet 11 from which it is ejected. This spiral movement, free and undisturbed and essentially dependent on the properties of the fluid, permitting it to adjust itself to natural paths or stream lines and to change its velocity and direction by insensible degrees, is characteristic of this method of propulsion and advantageous in its application. While traversing the chamber inclosing the runner, the particles of the fluid may complete one or more turns, or but a part of one turn. In any given case their path can be closely calculated and graphically represented, but fairly accurate estimate of turns can be obtained simply by determining the number of revolutions required to renew the fluid passing through the chamber and multiplying it by the ratio between the mean speed of the fluid and that of the disks. I have found that the quantity of fluid propelled in this manner is, other conditions being equal, approximately proportionate to the active surface of the runner and to its effective speed. For this reason, the performance of such machines augments at an exceedingly high rate with the increase of their size and speed of revolution.

The dimensions of the device as a whole, and the spacing of the disks in any given machine will be determined by the conditions and requirements of special cases. It may be stated that the intervening distance should be the greater, the larger the diameter of the disks, the longer the spiral path of the fluid and the greater its viscosity. In general, the spacing should be such that the entire mass of the fluid, before leaving the runner, is accelerated to a nearly uniform velocity, not much below that of the periphery of the disks under normal working conditions and almost equal to it when the outlet is closed and the particles move in concentric circles. It may also be pointed out that such a pump can be made without openings and spokes in the runner, as by using one or more solid disks, each in its own casing, in which form the machine will be eminently adapted for sewage, dredging and the like, when the water is charged with foreign bodies and spokes or vanes especially objectionable.

Another application of this principle which I have discovered to be not only feasible, but thoroughly practicable and efficient, is the utilization of machines such as above described for the compression or rarefaction of air, or gases in general. In such cases it will be found that most of the general considerations obtaining in the case of liquids, properly interpreted, hold true. When, irrespective of the character of the fluid, considerable pressures are desired, staging or compounding may be resorted to in the usual way the individual runners being, preferably, mounted on the same shaft. It should be added that the same end may be attained with one single runner by suitable deflection of the fluid through rotative or stationary passages.

The principles underlying the invention are capable of embodiment also in that field of mechanical engineering which is concerned in the use of fluids as motive agents, for while in some respects the actions in the latter case are directly opposite to those met with in the propulsion of fluids, the fundamental laws applicable in the two cases are the same. In other words, the operation above described is reversible, for if water or air under pressure be admitted to the opening 11 the runner is set in rotation in the direction of the dotted arrow by reason of the peculiar properties of the fluid which traveling in a spiral path and with continuously diminishing velocity, reaches the orifices 6 and 10 through which it is discharged.

When apparatus of the general character above described is employed for the transmission of power, however, certain departures from structural similarity between transmitter and receiver may be necessary for securing the best result. I have, therefore, included that part of my invention which is directly applicable to the use of fluids as motive agents in a separate application filed January 17, 1911, Serial No. 603,049. It may be here pointed out, however, as is evident from the above considerations, that when transmitting power from one shaft to another by such machines, any desired ratio between the speeds of rotation may be obtained by proper selection of the diameters of the disks, or by suitably staging the transmitter, the receiver, or both. But it may be stated that in one respect, at least, the two machines are essentially different. In the pump, the radial or static pressure, due to centrifugal force, is added to the tangential or dynamic, thus increasing the effective head and assisting in the expulsion of the fluid. In the motor, on the contrary, the first named pressure, being opposed to that of supply, reduces the effective head and velocity of radial flow toward the center. Again, in the propelled machine a great torque is always desirable, this calling for an increased number of disks and smaller distance of separation, while in the propelling machine, for numerous economic reasons, the rotary effort should be the smallest and the speed the greatest practicable. Many other considerations, which will naturally suggest themselves, may affect the design and construction, but the preceding is thought to contain all necessary information in this regard.

It will be understood that the principles

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N. TESLA.

FLUID PROPULSION.

APPLICATION FILED OCT. 21, 1909.

1,061,142. Patented May 6, 1913.

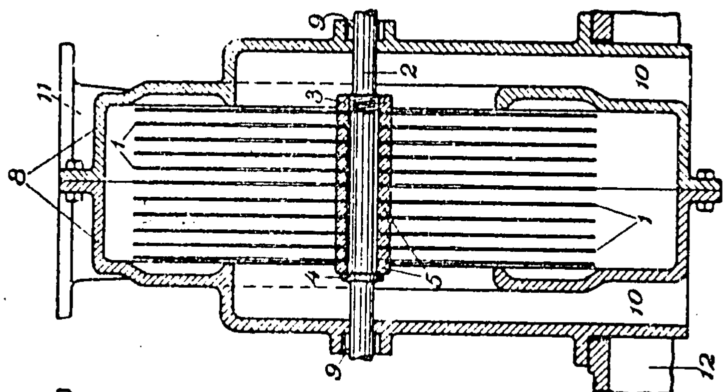

Fig 2

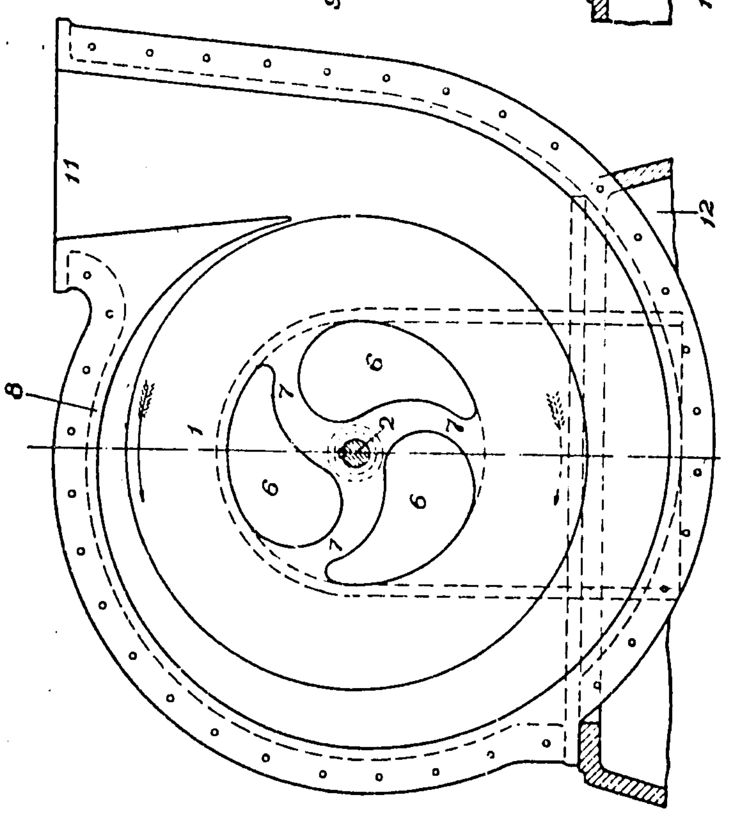

Fig 1

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UNITED STATES PATENT OFFICE.

NIKOLA TESLA, OF NEW YORK, N. Y.

TURBINE.

1,061,206. Specification of Letters Patent. Patented May 6, 1913.

Original application filed October 21, 1909, Serial No. 523,832. Divided and this application filed January 17, 1911. Serial No. 603,049.

To all whom it may concern:

Be it known that I, NIKOLA TESLA, a citizen of the United States, residing at New York, in the county and State of New York, have invented certain new and useful Improvements in Rotary Engines and Turbines, of which the following is a full, clear, and exact description.

In the practical application of mechanical power, based on the use of fluid as the vehicle of energy, it has been demonstrated that, in order to attain the highest economy, the changes in the velocity and direction of movement of the fluid should be as gradual as possible. In the forms of apparatus heretofore devised or proposed, more or less sudden changes, shocks and vibrations are unavoidable. Besides, the employment of the usual devices for imparting to, or deriving energy from a fluid, such as pistons, paddles, vanes and blades, necessarily introduces numerous defects and limitations and adds to the complication, cost of production and maintenance of the machines.

The object of my invention is to overcome these deficiencies and to effect the transmission and transformation of mechanical energy through the agency of fluids in a more perfect manner and by means simpler and more economical than those heretofore employed. I accomplish this by causing the propelling fluid to move in natural paths or stream lines of least resistance, free from constraint and disturbance such as occasioned by vanes or kindred devices, and to change its velocity and direction of movement by imperceptible degrees, thus avoiding the losses due to sudden variations while the fluid is imparting energy.

It is well known that a fluid possesses, among others, two salient properties, adhesion and viscosity. Owing to these a solid body propelled through such a medium encounters a peculiar impediment known as "lateral" or "skin resistance," which is twofold, one arising from the shock of the fluid against the asperities of the solid substance, the other from internal forces opposing molecular separation. As an inevitable consequence a certain amount of the fluid is dragged along by the moving body. Conversely, if the body be placed in a fluid in motion, for the same reasons, it is impelled in the direction of movement. These effects, in themselves, are of daily observation, but I believe that I am the first to apply them in a practical and economical manner in the propulsion of fluids or in their use as motive agents.

In an application filed by me October 21st, 1909, Serial Number 523,832 of which this case is a division, I have illustrated the principles underlying my discovery as embodied in apparatus designed for the propulsion of fluids. The same principles, however, are capable of embodiment also in that field of mechanical engineering which is concerned in the use of fluids as motive agents, for while in certain respects the operations in the latter case are directly opposite to those met with in the propulsion of fluids, and the means employed may differ in some features, the fundamental laws applicable in the two cases are the same. In other words, the operation is reversible, for if water or air under pressure be admitted to the opening constituting the outlet of a pump or blower as described, the runner is set in rotation by reason of the peculiar properties of the fluid which, in its movement through the device, imparts its energy thereto.

The present application, which is a division of that referred to, is specially intended to describe and claim my discovery, above set forth, so far as it bears on the use of fluids as motive agents, as distinguished from the applications of the same to the propulsion or compression of fluids.

In the drawings, therefore, I have illustrated only the form of apparatus designed for the thermo-dynamic conversion of energy, a field in which the applications of the principle have the greatest practical value.

Figure 1 is a partial end view, and Fig. 2 a vertical cross-section of a rotary engine or turbine, constructed and adapted to be operated in accordance with the principles of my invention.

The apparatus comprises a runner composed of a plurality of flat rigid disks 13 of suitable diameter, keyed to a shaft 16, and held in position thereon by a threaded nut 11, a shoulder 12, and intermediate washers 17. The disks have openings 14 adjacent to the shaft and spokes 15, which

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may be substantially straight. For the sake of clearness, but a few disks, with comparatively wide intervening spaces, are illustrated.

The runner is mounted in a casing comprising two end castings 19, which contain the bearings for the shaft 16, indicated but not shown in detail; stuffing boxes 21 and outlets 20. The end castings are united by a central ring 22, which is bored out to a circle of a slightly larger diameter than that of the disks, and has flanged extensions 23, and inlets 24, into which finished ports or nozzles 25 are inserted. Circular grooves 26 and labyrinth packing 27 are provided on the sides of the runner. Supply pipes 28, with valves 29, are connected to the flanged extensions of the central ring, one of the valves being normally closed.

For a more ready and complete understanding of the principle of operation it is of advantage to consider first the actions that take place when the device is used for the propulsion of fluids for which purpose let it be assumed that power is applied to the shaft and the runner set in rotation say in a clockwise direction. Neglecting, for the moment, those features of construction that make for or against the efficiency of the device as a pump, as distinguished from a motor, a fluid, by reason of its properties of adherence and viscosity, upon entering through the inlets 20, and coming in contact with the disks 13, is taken hold of by the latter and subjected to two forces, one acting tangentially in the direction of rotation, and the other radially outward. The combined effect of these tangential and centrifugal forces is to propel the fluid with continuously increasing velocity in a spiral path until it reaches a suitable peripheral outlet from which it is ejected. This spiral movement, free and undisturbed and essentially dependent on the properties of the fluid, permitting it to adjust itself to natural paths or stream lines and to change its velocity and direction by insensible degrees, is a characteristic and essential feature of this principle of operation.

While traversing the chamber inclosing the runner, the particles of the fluid may complete one or more turns, or but a part of one turn, the path followed being capable of close calculation and graphic representation, but fairly accurate estimates of turns can be obtained simply by determining the number of revolutions required to renew the fluid passing through the chamber and multiplying it by the ratio between the mean speed of the fluid and that of the disks. I have found that the quantity of fluid propelled in this manner, is, other conditions being equal, approximately proportionate to the active surface of the runner and to its effective speed. For this reason, the performance of such machines augments at an exceedingly high rate with the increase of their size and speed of revolution.

The dimensions of the device as a whole, and the spacing of the disks in any given machine will be determined by the conditions and requirements of special cases. It may be stated that the intervening distance should should be the greater, the larger the diameter of the disks, the longer the spiral path of the fluid and the greater its viscosity. In general, the spacing should be such that the entire mass of the fluid, before leaving the runner, is accelerated to a nearly uniform velocity, not much below that of the periphery of the disks under normal working conditions, and almost equal to it when the outlet is closed and the particles move in concentric circles.

Considering now the converse of the above described operation and assuming that fluid under pressure be allowed to pass through the valve at the side of the solid arrow, the runner will be set in rotation in a clockwise direction, the fluid traveling in a spiral path and with continuously diminishing velocity until it reaches the orifices 14 and 20, through which it is discharged. If the runner be allowed to turn freely, in nearly frictionless bearings, its rim will attain a speed closely approximating the maximum of that of the adjacent fluid and the spiral path of the particles will be comparatively long, consisting of many almost circular turns. If load is put on and the runner slowed down, the motion of the fluid is retarded, the turns are reduced, and the path is shortened.

Owing to a number of causes affecting the performance, it is difficult to frame a precise rule which would be generally applicable, but it may be stated that within certain limits, and other conditions being the same, the torque is directly proportionate to the square of the velocity of the fluid relatively to the runner and to the effective area of the disks and, inversely, to the distance separating them. The machine will, generally, perform its maximum work when the effective speed of the runner is one-half of that of the fluid; but to attain the highest economy, the relative speed or slip, for any given performance, should be as small as possible. This condition may be to any desired degree approximated by increasing the active area of and reducing the space between the disks.

When apparatus of the kind described is employed for the transmission of power certain departures from similarity between transmitter and receiver are necessary for securing the best results. It is evident that, when transmitting power from one shaft to another by such machines, any desired ratio between the speeds of rotation may be obtained by a proper selection of the diameters of the disks, or by suitably staging the transmitter, the receiver or both. But it may be pointed out that in one respect, at least, the two machines are essentially different. In the pump, the radial or static pressure, due to centrifugal force, is added to the tangential or dynamic, thus increasing the effective head and assisting in the expulsion of the fluid. In the motor, on the contrary, the first named pressure, being opposed to that of supply, reduces the effective head and the velocity of radial flow toward the center. Again, in the propelled machine a great torque is always desirable, this calling for an increased number of disks and smaller distance of separation, while in the propelling machine, for numerous economic reasons, the rotary effort should be the smallest and the speed the greatest practicable. Many other considerations, which will naturally suggest themselves, may affect the design and construction, but the preceding is thought to contain all necessary information in this regard.

In order to bring out a distinctive feature, assume, in the first place, that the motive medium is admitted to the disk chamber through a port, that is a channel which it traverses with nearly uniform velocity. In this case, the machine will operate as a rotary engine, the fluid continuously expanding on its tortuous path to the central outlet. The expansion takes place chiefly along the spiral path, for the spread inward is opposed by the centrifugal force due to the velocity of whirl and by the great resistance to radial exhaust. It is to be observed that the resistance to the passage of the fluid between the plates is, approximately, proportionate to the square of the relative speed, which is maximum in the direction toward the center and equal to the full tangential velocity of the fluid. The path of least resistance, necessarily taken in obedience to a universal law of motion is, virtually, also that of least relative velocity. Next, assume that the fluid is admitted to the disk chamber not through a port, but a diverging nozzle, a device converting wholly or in part, the expansive into velocity-energy. The machine will then work rather like a turbine, absorbing the energy of kinetic momentum of the particles as they whirl, with continuously decreasing speed, to the exhaust.

The above description of the operation, I may add, is suggested by experience and observation, and is advanced merely for the purpose of explanation. The undeniable fact is that the machine does operate, both expansively and impulsively. When the expansion in the nozzles is complete, or nearly so, the fluid pressure in the peripheral clearance space is small; as the nozzle is made less divergent and its section enlarged, the pressure rises, finally approximating that of the supply. But the transition from purely impulsive to expansive action may not be continuous throughout, on account of critical states and conditions and comparatively great variations of pressure may be caused by small changes of nozzle velocity.

In the preceding it has been assumed that the pressure of supply is constant or continuous, but it will be understood that the operation will be, essentially the same if the pressure be fluctuating or intermittent, as that due to explosions occurring in more or less rapid succession.

A very desirable feature, characteristic of machines constructed and operated in accordance with this invention, is their capability of reversal of rotation. Fig. 1, while illustrative of a special case, may be regarded as typical in this respect. If the right hand valve be shut off and the fluid supplied through the second pipe, the runner is rotated in the direction of the dotted arrow, the operation, and also the performance remaining the same as before, the central ring being bored to a circle with this purpose in view. The same result may be obtained in many other ways by specially designed valves, ports or nozzles for reversing the flow, the description of which is omitted here in the interest of simplicity and clearness. For the same reasons but one operative port or nozzle is illustrated which might be adapted to a volute but does not fit best a circular bore. It will be understood that a number of suitable inlets may be provided around the periphery of the runner to improve the action and that the construction of the machine may be modified in many ways.

Still another valuable and probably unique quality of such motors or prime movers may be described. By proper construction and observance of working conditions the centrifugal pressure, opposing the passage of the fluid, may, as already indicated, be made nearly equal to the pressure of supply when the machine is running idle. If the inlet section be large, small changes in the speed of revolution will produce great differences in flow which are further enhanced by the concomitant variations in the length of the spiral path. A self-regulating machine is thus obtained bearing a striking resemblance to a direct-current electric motor in this respect that, with great differences of impressed pressure in a wide open channel the flow of the fluid through the same is prevented by virture of rotation. Since the centrifugal head increases as the square of the revolutions, or even more rapidly, and with modern high grade steel great peripheral velocities are practicable, it is possible to attain that condition in a single stage machine, more readily if the runner be of large diameter. Obviously this problem is

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facilitated by compounding, as will be understood by those skilled in the art. Irrespective of its bearing on economy, this tendency which is, to a degree, common to motors of the above description, is of special advantage in the operation of large units, as it affords a safeguard against running away and destruction. Besides these, such a prime mover possesses many other advantages, both constructive and operative. It is simple, light and compact, subject to but little wear, cheap and exceptionally easy to manufacture as small clearances and accurate milling work are not essential to good performance. In operation it is reliable, there being no valves, sliding contacts or troublesome vanes. It is almost free of windage, largely independent of nozzle efficiency and suitable for high as well as for low fluid velocities and speeds of revolution.

It will be understood that the principles of construction and operation above generally set forth, are capable of embodiment in machines of the most widely different forms, and adapted for the greatest variety of purposes. In my present specification I have sought to describe and explain only the general and typical applications of the principle which I believe I am the first to realize and turn to useful account.

What I claim is:

1. A machine adapted to be propelled by a fluid consisting in the combination with a casing having inlet and outlet ports at the peripheral and central portions, respectively, of a rotor having plane spaced surfaces between which the fluid may flow in natural spirals and by adhesive and viscous action impart its energy of movement to the rotor, as described.

2. A machine adapted to be propelled by a fluid, comprising a rotor composed of a plurality of plane spaced disks mounted on a shaft and open at or near the same, an inclosing casing with a peripheral inlet or inlets, in the plane of the disks, and an outlet or outlets in its central portion, as described.

3. A rotary engine adapted to be propelled by adhesive and viscous action of a continuously expanding fluid comprising in combination a casing forming a chamber, an inlet or inlets tangential to the periphery of the same, and an outlet or outlets in its central portion, with a rotor composed of spaced disks mounted on a shaft, and open at or near the same, as described.

4. A machine adapted to be propelled by fluid, consisting in the combination of a plurality of disks mounted on a shaft and open at or near the same, and an inclosing casing with ports or passages of inlet and outlet at the peripheral and central portions, respectively, the disks being spaced to form passages through which the fluid may flow, under the combined influence of radial and tangential forces, in a natural spiral path from the periphery toward the axis of the disks, and impart its energy of movement to the same by its adhesive and viscous action thereon, as set forth.

5. A machine adapted to be propelled by a fluid comprising in combination a plurality of spaced disks rotatably mounted and having plane surfaces, an inclosing casing and ports or passages of inlet and outlet adjacent to the periphery and center of the disks, respectively, as set forth.

6. A machine adapted to be propelled by a fluid comprising in combination a runner composed of a plurality of disks having plane surfaces and mounted at intervals on a central shaft, and formed with openings near their centers, and means for admitting the propelling fluid into the spaces between the disks at the periphery and discharging it at the center of the same, as set forth.

7. A thermo-dynamic converter, comprising in combination a series of rotatably mounted spaced disks with plane surfaces, an inclosing casing, inlet ports at the peripheral portion and outlet ports leading from the central portion of the same, as set forth.

8. A thermo-dynamic converter, comprising in combination a series of rotatably mounted spaced disks with plane surfaces and having openings adjacent to their central portions, an inclosing casing, inlet ports in the peripheral portion, and outlet ports leading from the central portion of the same, as set forth.

In testimony whereof I affix my signature in the presence of two subscribing witnesses.

NIKOLA TESLA.

Witnesses:
M. LAWSON DYER,
WM. BOHLEDER.

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N. TESLA.

TURBINE.

APPLICATION FILED JAN. 17, 1911.

1,061,206. Patented May 6, 1913.

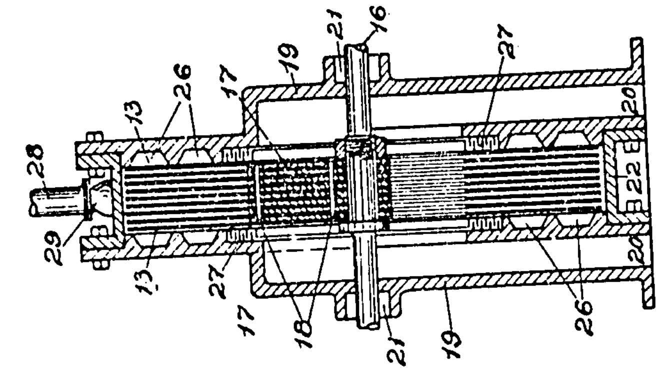

Fig. 2.

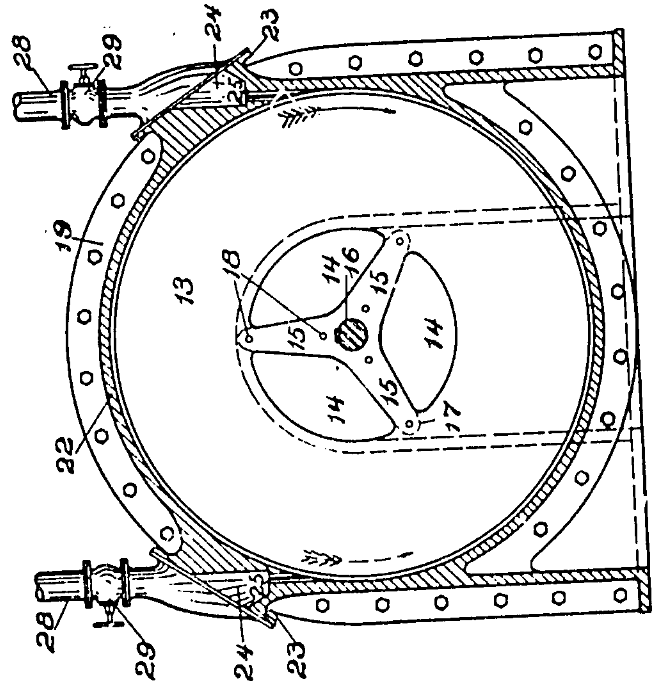

Fig. 1.

Witnesses:

R. Diaz Buitrago

Wm Bohleber

Nikola Tesla, Inventor

By his Attorneys

Kerr Page Cooper & Hayward

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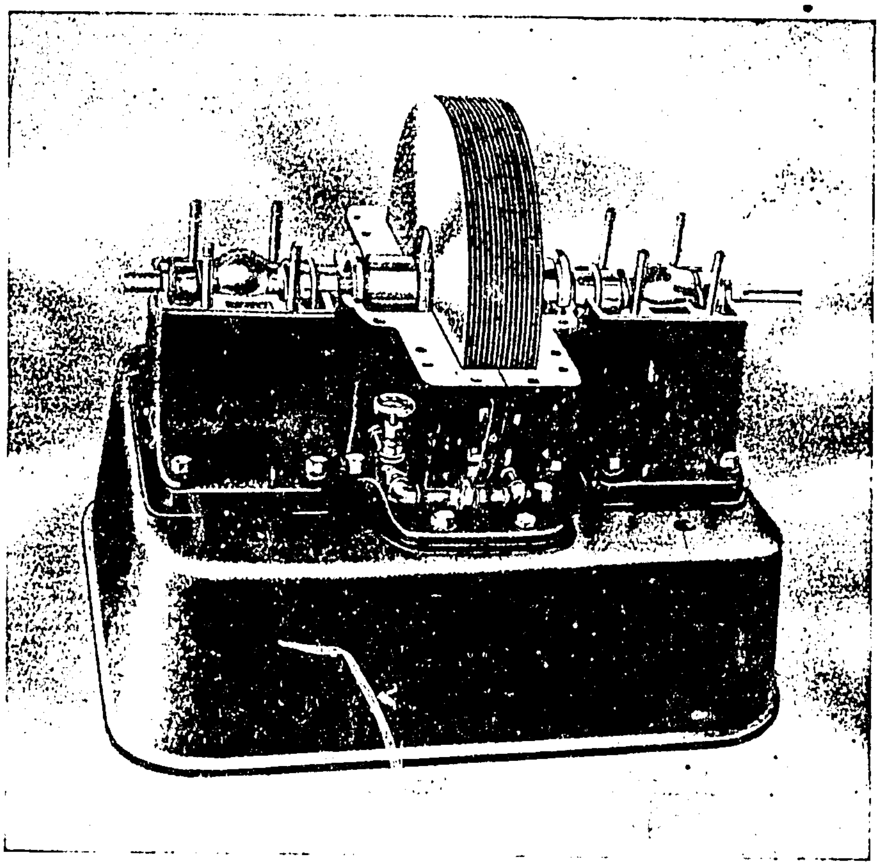
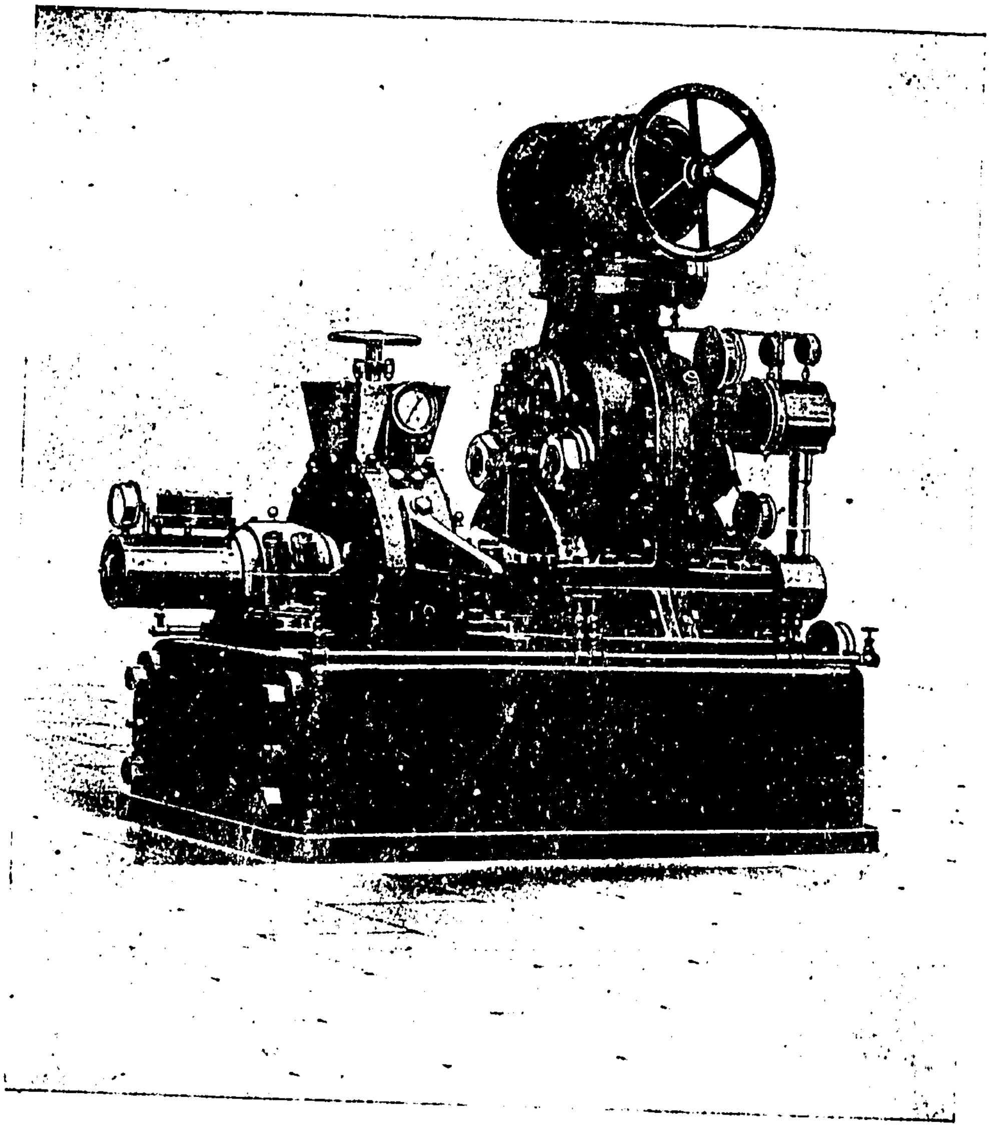


Lower part with rotor of Tesla's steam turbine



Tesla's 10000 HP steam turbine

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