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Structural Design and Analysis of Torque and Power Characteristics of a Multistage Rotary Jet-Driven Rotor

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Abstract

This paper proposes a rotary jet-driven structure designed to address flow-path configuration challenges in multi-stage rotating machinery during gas expansion; the structure comprises a conical rotor, curved internal flow channels, and tangential exhaust ports. By varying the inlet-to-outlet radius ratio and progressively increasing the total exhaust area, the effects of jet reaction and fluid angular-momentum flux on shaft torque are investigated. A five-stage steam rotor is considered as an example, with a specified mass flow rate of 0.1 kg/s, inlet temperature of 140 °C, first-stage inlet pressure of 320 kPa, final-stage back pressure of 101.3 kPa, and idealized lossless conditions. Predicted relative jet velocities and shaft powers are presented for a 25% stage-by-stage increase in total exhaust area. Assuming operation at 50% of the no-load speed, the calculated total shaft powers are 10.77 kW and 11.80 kW for inlet-to-outlet radius ratios of 0.8 and 0.7, respectively; further changes in this ratio would significantly alter shaft power. This paper summarizes the structural design, preliminary calculations, and requirements for subsequent validation. The rotary jet-driven structure is simple, has low manufacturing cost and high efficiency, and can substantially improve the efficiency of various devices that use gas to generate torque. The drive structure may be used individually or in multiple coaxially connected stages. It is not limited to high-speed gases such as combustion gas, steam, compressed air, and carbon dioxide.

Keywords: Rotary Jet Propulsion; Tangential Nozzle; Reaction Torque; Multistage Expansion; Steam Power; Conservation of Angular Momentum

Introduction

Conventional rotating machinery, such as steam turbines and gas turbines with bladed impellers, converts energy through angular-momentum exchange between the fluid and the rotor. The impeller diameter must be increased from stage to stage according to the gas expansion ratio to accommodate the increase in gas volume caused by expansion and to prevent the resulting additional increase in gas velocity. This serves, first, to prevent shock waves, turbulence, and blockage in high-speed flow and, second, to maintain the same rotational speed in upstream and downstream impellers. For example, conventional steam turbines must increase impeller diameter to provide additional space for the increased steam

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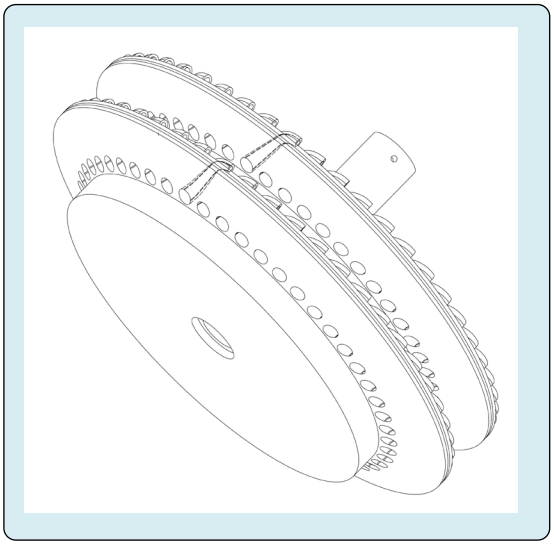
volume during expansion. Since steam expansion performs work, allowing expansion by providing space itself implies an energy loss; nevertheless, this is a measure that must be taken when bladed impellers are used for kinetic-energy conversion. At the same time, bladed impellers remain subject to the Betz-limit conditions; impellers operating at different tip-speed ratios rotate together on a common shaft, significantly reducing overall impeller efficiency.

This paper examines a rotary jet-driven rotor that differs from conventional blade cascades: gas enters a curved internal passage at a smaller radius and is discharged approximately tangentially at a larger radius. The focus is on the possible effects of exhaust-area configuration, inlet-to-outlet radius ratio, and rotational speed on torque and power. This paper does not apply the Betz limit to evaluate the efficiency of a steam turbine with enclosed flow passages, because that limit concerns a particular idealized model of energy extraction from open flow.

Structure and Operating Principle

The main body of the drive wheel is a conical rotor with a smaller-diameter end and a larger-diameter end. Multiple gas inlet ports are arranged circumferentially on the conical surface near the smaller-diameter end, while a corresponding number of exhaust ports are arranged near the circumferential tangent at the larger-diameter end. The inlet and outlet are connected by curved internal passages. The inlet cross-sectional area is at least 1.2 times the outlet area, the ratio of the diameters at the inlet and outlet locations is 0.6–0.9, and the jet discharge direction is nearly tangential.

Multiple rotors may be connected coaxially in series, with the total exhaust area of each subsequent stage increased by approximately 5%–25% relative to the preceding stage. The actual ratio should be determined from mass conservation, interstage pressure, density variation, compressible flow, and the load on each stage under the operating conditions of the working fluid.



Theoretical Analysis Method

Let the mass flow rate be $\dot{m}$; the inlet and outlet radii be r_1 and r_2; and the absolute tangential fluid velocities be $V\theta 1$ and $V\theta 2$, respectively. The steady-state control-volume angular-momentum equation gives the torque exerted by the fluid on the rotor:

$$T = \dot{m} (r_1 V\theta 1 - r_2 V\theta 2).$$

If the outlet relative velocity is directed along the reverse tangent, the relative jet velocity W_2 must be converted into the absolute tangential velocity $V\theta 2$ using a velocity triangle and the rotor peripheral speed $U_2 = \omega r_2$.

The mass-conservation relation is $\dot{m} = \rho * A * Wn$, where ρ is the density at the flow-passage cross section, A is the corresponding effective area, and Wn is the relative normal velocity through that section. At the outlet of a rotating passage, this expression must account for jet direction, discharge coefficient, and choking conditions. Shaft power is $P = T\omega$.

Five-Stage Steam Example and Results

The steam temperature is set at 140 °C, supply pressure at 362 kPa, density at 1.96 kg/m^3, and mass flow rate at 0.1 kg/s. The first-stage rotor inlet pressure is 320 kPa, and the final-stage outlet pressure is 101.3 kPa. Each of the five rotors has 50 sets of flow passages. The rotor inlet port diameter is 5.8 mm, and the first-stage outlet port diameter is 2.38 mm (total exhaust area: 222 mm^2). The total exhaust

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area of each subsequent stage increases by 25% relative to the previous stage: the second-stage outlet diameter is 2.66 mm (278 mm^2), followed by 347 mm^2 in the third stage, 434 mm^2 in the fourth, and 543 mm^2 in the fifth. The diameters at the inlet and outlet locations are 160 mm and 200 mm, respectively, corresponding to radii of 0.08 m and 0.10 m and r_1/r_2 = 0.8. An adiabatic, leakage-free, ideal lossless model with fully tangential passage outlets is assumed, the isentropic enthalpy drops allocated to each stage are Δh_i = [40.66, 40.37, 39.15, 38.12, 37.01] kJ/kg respectively. With isentropic enthalpy drops allocated among the stages. Based on $\left[\mathrm{Wi} = \frac{\dot{m}}{p_i A_i} \right]$ and the changes in outlet area, the calculated outlet steam densities and velocities for the stages are listed below.

Stage	Total outlet area (mm^2)	Single-port diameter (mm)	Outlet steam density (kg/m^3)	Relative jet velocity (m/s)
1	222.4	2.38	1.645	316.6
2	278.1	2.661	1.314	308.4
3	347.6	2.975	1.052	304.5
4	434.5	3.326	0.841	298.9
5	543.1	3.719	0.673	289.1

Table 1: Predicted Five-Stage Exhaust Geometry and Relative Jet Velocities.

The corresponding densities for the IF97 isentropic states are approximately 1.420, 1.166, 0.945, 0.770, and 0.637; the two sets of values tend toward agreement.

After entering the rotating rotor, steam expands/accelerates within the rotor and is discharged tangentially from the nozzle, generating torque through the reaction force. All five rotor stages simultaneously satisfy m=Pi*Ai*Wi (m: mass; P: pressure; A: area; W: jet velocity), as stated in the source.

Calculation of Angular Velocity and Jet Velocity:

Steam enters at an inlet location with a diameter of 160 mm and exits at a location with a diameter of 200 mm: r_1 = 0.08 m, r_2 = 0.10 m.

Angular-momentum equation: $Ti = (r_1 V_{\theta 1,i} - r_2 V_{\theta 2,i})$. Assuming that the inlet absolute tangential velocity of each stage is zero and that the outlet relative jet is directed along the reverse tangent: $[V_{\theta 2,i} = \omega r_2 - W_i]$; then, $[T_i = M r_2 (W_i - \omega r_2)]$.

Total torque of the five stages, $T_{total} = 0.1 \times 0.1 \sum_{(i=1)}^{5} (W_i - 0.1\omega)$. Substituting the five-stage jet velocities T_{total}= 15.175-0.005ω. Here, ω is in rad/s and torque is in N·m. In the theoretical no-load condition, neglecting bearing friction and other mechanical losses, when the combined torque of the five stages is zero, i.e., T = 0, 15.175 − 0.05 = 0, = 3035 rad/s, corresponding to 28,982 rpm.

Assuming maximum shaft power is obtained at 50% of the maximum no-load speed, the rated speed is 14,491 rpm.

Rotor angular velocity:

$$\left[\omega = \frac{2\pi \times 14491}{60} \approx 1518 rad/s \right]$$

Rotor peripheral speed at the inlet: $[u_1 = 0.08\omega \approx 121.4 m/s]$

Rotor peripheral speed at the outlet: $[u_2 = 0.10\omega \approx 151.8 m/s]$.

The resulting angular momentum $\left[\Delta e_r = \frac{u_2^2 - u_1^2}{2} \approx 4.15 kJ/kg \right]$ indicates that the 160 mm/200 mm difference in inlet and outlet positions gives the rotor 4.15 kJ/kg of angular momentum at the rated speed of 14,491 rpm.

Using the simplified formula $\left[W_i = \sqrt{2000(\Delta h_i + \Delta e_r)} \right]$, the jet velocities for the stages are:

$$W_1 = \sqrt{2000(40.66 + 4.15)} = 299 m/s$$

$$W_2 = \sqrt{2000(40.37 + 4.15)} = 298.4 m/s$$

$$W_3 = \sqrt{2000(39.15 + 4.15)} = 294.3 m/s$$

$$W_4 = \sqrt{2000(38.12 + 4.15)} = 290.8 m/s$$

$$W_5 = \sqrt{2000(37.01 + 4.15)} = 286.9 m/s$$

Calculation of the power of each rotor stage at 14,491 rpm: $F_i = m(W_i - u_2)$; $T_i = F_i \times r_2$; $P_i = T_i \omega$] At an inlet-to-outlet diameter ratio of 0.8 and 14,491 rpm, u_2 = 151.8m/s, ω=1518rad/s.

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Stage	Relative jet velocity W (m/s)	Net tangential force F (N)	Torque T (N·m)	Shaft power P (kW)
Stage 1	299	14.72	1.472	2.235
Stage 2	298.4	14.66	1.466	2.225
Stage 3	294.3	14.25	1.425	2.163
Stage 4	290.8	13.9	1.39	2.095
Stage 5	286.9	13.51	1.351	2.05
Total		71.04	7.104	10.77

Table 2: The shaft powers of the individual stages and their total.

If the inlet position is changed so that its circumferential radius decreases to 0.07 m while the outlet diameter remains unchanged, the ratio r_1/r_2 decreases from 0.8 to 0.7. At 14,491 rpm:

$[u_1 = 0.07\omega \approx 106.3 m/s, u_2 = 0.1\omega \approx 151.8 m/s, \omega = 1518 rad/s,$

$\Delta e_r = \frac{u_2^2 - u_1^2}{2} \approx 5.872 kJ/kg]$

Shaft power calculation using: $F_i = m(W_i - u_2)$; $T_i = F_i \times r_2$; $P_i = T_i \omega$.

Stage	Relative jet velocity W (m/s)	Net tangential force F (N)	Torque T (N·m)	Shaft power P (kW)
Stage 1	312.5	16.07	1.607	2.439
Stage 2	311.6	15.98	1.598	2.426
Stage 3	307.4	15.56	1.556	2.362
Stage 4	304.3	15.25	1.525	2.315
Stage 5	300.6	14.88	1.488	2.259
Total		77.74	7.774	11.8

Table 3: The calculation results.

If maximum shaft power is assumed to occur at 70% of the no-load speed, the rated speed is 20,287 rpm under the same conditions.

Calculation of the rotational energy term:

Angular velocity: $\omega = \frac{2\pi \times 20287}{60} \approx 2124.5 rad/s$

Inlet peripheral speed: $u_1 = 0.08\omega \approx 170 m/s$

Outlet peripheral speed: $u_2 = 0.10\omega \approx 212.5 m/s$

Obtain the angular momentum

$$\Delta e_r = \frac{u_2^2 - u_1^2}{2} \approx 8.18 kJ/kg$$

i.e. the angular momentum obtained under the 160 mm/200 mm difference in inlet and outlet positions at 20,287 rpm is 8.18 kJ/kg; the shaft power would therefore increase correspondingly.

Using the same calculation method, if the inlet/outlet positional difference decreases from 0.7 to zero, the angular momentum also decreases to zero, and the shaft power decreases accordingly. For example, the first-stage jet velocity decreases from 312.5 m/s to 286 m/s, a reduction of approximately 8.5%; the first-stage shaft power decreases from 2.439 kW to 2.03 kW, a reduction of approximately 16.7%.

Further Research and Discussion

- The expansion ratio of exhaust ports in each stage under different gas-source conditions.
- The contribution of inlet-flow preswirl direction to the angular-momentum term.
- The relationship between the rated rotational speed and the no-load rotational speed of the rotor.

Potential Applications

Steam turbines, gas turbines, turbofan engines, and turboshaft engines.

Conclusion

1. This paper proposes a multistage rotary jet-driven structure centered on curved internal passages and

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tangential nozzles in conical rotors.

2. The five-stage idealized example shows that progressively increasing the total exhaust-port area and changing the inlet-to-outlet radius ratio can significantly affect the relative jet velocity, torque, and shaft power predicted by the model.
3. The angular-momentum term and shaft power also change significantly at different no-load rotational speeds.

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