ENERGETIC MATCHING OF THE IMPULSE SOURCE (GIF²) AND THE TURBINE TERMINAL (RTM) IN A CLOSED HYDRAULIC CIRCUIT

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Abstract.
The article is devoted to the engineering substantiation of the Reactive-Turbine Module (RTM) – a terminal device of the Hydro-Impulse Power Plant (HSPP), which converts the kinetic energy of a quasi-continuous high-velocity jet formed by the Gravitational-Impulse Flow Former (GIF²) as a result of the overlap of impulse modules into electrical energy, with subsequent dissipation of residual energy in the receiver tank, and ensures recirculation of the working fluid in a closed hydraulic circuit.

Based on the energy analysis of GIF² [4] and the design solutions of the elbow-free configuration [5], the jet parameters were determined: frequency 2.3900 Hz, power 2.6888 MW at an equivalent dynamic head corresponding to a jet velocity of 126.1 m/s, main jet flow rate 337.7 L/s, total flow rate at the RTM inlet 338.3 L/s including auxiliary flows.

The Pelton turbine with a wheel diameter of 0.75 m (material AISI 416 with nitriding) was calculated and compared with serial turbines from leading manufacturers (Voith [36], Andritz [37], GE [38]). The parameters of the generator (2.3 MW, 1500 rpm) and the recirculation pump (22 kW, flow rate 338.3 L/s, Grundfos KPL or KSB Amacan P) are presented.

Strength and durability calculations of the receiver tank (volume 6 m³ for 1 module, scaling up to 54 m³ for 6 modules) and the pump were performed in accordance with ASME B31.3 [26] and EN 13445 [28].

The energy balance of the RTM confirms the net power of the plant as 2.277 MW for one module with an overall efficiency of 84.7% relative to the jet power at the RTM inlet, and 13.66 MW for six modules.

The results are consistent with the fundamental laws of mass and energy conservation and with regulatory requirements.

Keywords: reactive-turbine module, RTM, Pelton turbine, hydro-impulse power plant, HSPP, GIF², energy balance, recirculation, scaling.

1. Introduction.
The Hydro-Impulse Power Plant (HSPP) is an innovative energy system consisting of two main modules: the Gravitational-Impulse Flow Former (GIF²) and the Reactive-Turbine Module (RTM). The design of GIF², including the bifurcation architecture, pipeline geometry (L_1 = 1.7 m, L_3 = 3.0 m, L_5 = 4.7 m, L_9 = 3.85 m), internal diameter of 245 mm, valve opening angles (37° for V4, 52° for V2 and V7), is the subject of patent protection [1, 2]. In previous work [4], an energy analysis of GIF² was performed, where the conversion efficiency η = 66.1% in the nominal mode (water hammer pressure 120.5 atm, frequency 2.39 Hz) was demonstrated. Article

[5] substantiates the choice of materials and confirms a service life of at least 15 years for all main elements of GIF².

The present work is devoted to the engineering substantiation of the Reactive-Turbine Module (RTM), which is a terminal device of the HSPP. The patent application for the RTM [3] protects the module design, including the inlet flange, impulse hydraulic turbine, generator, bypass channel, receiver tank with a second load-bearing bottom, recirculation pump, and air vent system.

The aim of the article is:

- matching the GIF² jet parameters with the characteristics of the Reactive-Turbine Module (RTM);
- to determine the initial jet parameters based on [4] and [5];
- to calculate and compare the Pelton turbine with serial analogues;
- to select and compare the generator and recirculation pump with commercially available products;
- to design the receiver tank considering scaling;
- to perform strength and durability calculations;
- to compile the energy balance of the RTM.

2. Initial Jet Parameters from GIF².

The geometry of the GIF² system [4] includes: tank R1 with a volume of 5 m³ (gravitational head $h = 0.9$ m), pipelines with an internal diameter of 245 mm ($L_1 = 1.7$ m, $L_3 = 3.0$ m – determines the volume of the main module 0.1413 m³, $L_5 = 4.7$ m, $L_9 = 3.85$ m), and high-speed swing check valves V2, V4, V7 (disc diameter 325 mm, passage diameter 238 mm). The critical impact valve V4 is installed at an angle of 37°, while V2 and V7 are installed at an angle of 52° [4].

The jet parameters presented in this work are formed by the self-oscillating hydrodynamic system of the Gravitational-Impulse Flow Former (GIF²), in which the frequency and cycle duration are determined by the internal dynamics of the high-speed check valves and the interaction of the fluid with the structure. The energy source of the system is the gravitational field, realised through the motion of the working fluid, while the formation of high instantaneous pressure values (up to ≈120–150 atm) is due to water hammer processes. These processes provide temporal and spatial concentration of the flow energy without its generation, but only through the conversion of potential energy into kinetic and impulse form, which complies with the fundamental law of energy conservation.

In this work, the jet parameters (velocity, flow rate, frequency and energy characteristics) are taken as given (input) parameters based on the results of previous studies [4, 5], where their numerical and analytical substantiation is presented. The re-derivation of these parameters is not the subject of this article, which focuses on the engineering matching of the jet parameters with the characteristics of the Reactive-Turbine Module (RTM).

Table 1. Main jet parameters (nominal mode).

Parameter	Value	Source / Calculation
Main module volume per cycle	141.3 L	[4] (pipe $L_3 = 3.0$ m, ID 245 mm)
Minor module (spill) volume per cycle	0.26 L	[4] (V4 closure dynamics, 13.42 m/s, 1.34 ms)
Jet frequency	2.39 Hz	[4], [5] (cycle duration 0.4184 s)

Parameter	Value	Source / Calculation
Main jet flow rate	337.707 L/s	141.3 L/s · 2.39 Hz
Minor module spill flow rate	0.6214 L/s	0.26 L/s · 2.39 Hz
Total flow rate at RTM inlet	338.3284 L/s	Sum
Jet velocity	126.1 m/s	[4]
Delivered energy per cycle	1125.0 kJ	[4]
Jet power	2.6888 MW	1125.0 kJ · 2.39 Hz

It should be noted that the given values of jet energy and power are determined based on the GIF² model [4, 5] and are used in this work as input parameters for the RTM analysis. Verification of the complete energy balance of the GIF² system is beyond the scope of this work.

2.1. Determination of the Energy Efficiency of the Impulse Module.

The energy efficiency of impulse formation at the GIF² outlet is defined as the ratio of the actual delivered jet energy to the maximum possible work of fluid compression in the module:

$$\eta=\frac{E_{del}}{E_{max}}, \tag{1}$$

where:

Emax – maximum compression work that can be realised in the module volume:

$$\mathbf{E_{max}=(P-P_{atm})V_{mod}}, \tag{2}$$

(in the quasi-static compression approximation),

E_{del} - the energy delivered by the jet per cycle.

This formulation is consistent with classical concepts of fluid compression work and does not imply energy generation in the system, but rather its redistribution and concentration in the form of an impulse jet.

For the given parameters ($P \approx 120.5$ atm, $V_{mod} = 0{,}1413$ m³), the estimate of $E_{max}\approx1704$ kJ, which is consistent with the pulse energy of 1125.0 kJ within the model.

The obtained value $\eta=0{,}661$ is used to verify the consistency of the jet parameters presented in Table 1 and corresponds to losses due to hydraulic resistance, valve imperfections, and energy dissipation during the water hammer process.

2.2. Verification of the Inlet Pipeline and Reservoir R1 Capacity of the GIF² System (Gravity Impulse Flow Former).

When the pulse frequency increases to the nominal value $f=2.39$ Hz, the average flow rate at the RTM inlet rises to $Q_{avg}=338$ L/s (see Table 1). This value is averaged over a long time interval; however, during the continuous flow phase (before the hydraulic shock), the instantaneous flow rate in the supply pipeline (1) is determined by the velocity $v=13.42$ m/s (according to [4]) and the pipe cross-sectional area $A=\pi D^2/4=0.0471$ m² (with inner diameter $D=245$ mm):

$Q_{inst}=A\cdot v=0.0471\cdot13.42=0.632$ m³/s $=632$ L/s.

Thus, the supply pipeline can handle an instantaneous flow rate almost twice the average value, which is normal for impulse systems.

The Reynolds number at a kinematic viscosity of water $\nu=1\cdot10^{-6}$ m²/s is:

$Re=\frac{vD}{\nu}=\frac{13.42\cdot0.245}{\nu}1\cdot10^{-6}\approx3.29\cdot10^{6}$.

This corresponds to a fully developed turbulent regime, for which the hydraulic friction coefficient $\lambda\approx0.012$ (using the Blasius formula for hydraulically smooth pipes). The friction head loss in the supply pipeline of length $L_1=1.7$ m is:

$h_{fric}=\lambda\frac{L_1\cdot v^2}{D2g}=0.012\cdot\frac{1.7}{0.245}\cdot\frac{13.42^2}{2\cdot9.81}\approx0.012\cdot6.94\cdot9.18\approx0{,}76$ m.

This value is acceptable and does not exceed the gravitational head of 0.9 m. Moreover, the main flow driver in the nominal regime is the pressure difference created by the rarefaction in the working pipeline (3), not gravity. Therefore, the existing losses do not limit the system performance.
Reservoir R1 has a volume of 5 m^3 and a cylindrical part height of 1.3 m. During the module accumulation time (τacc≈0.4 s), the volume withdrawn from the reservoir is:
$V_{out} = Q_{inst} \cdot \tau ac = 0{,}632 \cdot 0{,}4 = 0{,}253$ m^3.
The water level drop in the reservoir, whose area is $S=(\pi D_{res}{}^2)/4=(\pi\cdot 1{,}7^2)/4\approx 2.27$ m^2 , is:
$\Delta h = V_{out}/S = 0{,}253/2{,}27 \approx 0{,}11$ m.
This is less than 10 % of the cylindrical part height. Filling the reservoir through a 506 mm diameter pipe at a gravitational velocity of ≈4.2 m/s (corresponding to a head of 0.9 m) occurs at a flow rate of ≈0.85 m^3/s, i.e., much faster than the outflow. Thus, the reservoir maintains a stable water level throughout the cycle.

Conclusion: The capacity of the supply pipeline and reservoir R1 has a significant margin relative to the nominal mode (f = 2.39 Hz, Qavg = 338 L/s) and is not a limiting factor for the operation of the GIF^2 system.

Meanwhile, to ensure an average flow rate of 338 L/s at a gravitational head of 0.9 m, a pipe diameter of at least 320 mm is sufficient. The existing initial filling pipe in the design, with a diameter of 500 mm, has a safety margin of more than two, so no additional pump power is required. The water level fluctuations in reservoir R1 during impulse extraction are about 0.11 m, which does not affect the system stability.

The minimum required cross-sectional area of the pipe is:
$A_{min}=\frac{Q_{avg}}{v}=0.338/4.20\approx 0.0805$ m^2.
The corresponding diameter is:
$D_{min}=\sqrt{\frac{4A_{min}}{\pi}}=\sqrt{\frac{4\cdot 0.0805}{\pi}}=\sqrt{0.1025}=\approx 0.320$ m=320 mm.
Effect of flow pulsations:
The system consumes water not uniformly but in pulses: the instantaneous flow rate reaches 632 L/s during the continuous flow phase (13.42 m/s). If the pipe fed the reservoir directly during the impulse, the required diameter to cover the instantaneous flow rate would be:
$A_{inst}=\frac{0.632}{4.20}\approx 0{,}1505$ m^2,
$D_{inst}=\sqrt{\frac{4\cdot 0.1505}{\pi}}=\sqrt{0{,}1916}=\approx 0{,}438$ m=438 mm.
However, reservoir R1 has its own water reserve (5 m^3), which smoothes the pulsations. The water level in the reservoir fluctuates by about 0.11 m (as calculated above), which is negligible. Therefore, the pipe can operate in average mode, and peak flow rates are compensated by the buffering capacity of the reservoir.

Actual pipe diameter in the system:
According to the design documentation [4], the initial filling pipe has a diameter of 500 mm (item 2). This diameter significantly exceeds the theoretical minimum (320 mm) and even the instantaneous requirement (438 mm). Hence, the existing pipe is more than sufficient to provide the average flow rate of 338 L/s without increasing pump power. The area margin is:
$\frac{A_{500}}{A_{min}}=\frac{(\pi\cdot 0.5^2)/4}{0.0805}=\frac{0.1963}{0.0805}\approx 2.44.$
To ensure an average flow rate of 338 L/s at a gravitational head of 0.9 m, a pipe diameter of at least 320 mm is sufficient. The existing initial filling pipe with a diameter of 500 mm provides

more than twice the required margin, so no additional pump power is needed. The water level fluctuations in reservoir R1 during impulse extraction are about 0.11 m, which does not affect the system stability: the system operates in impulse mode, but the hydraulics are designed based on average values because there is a buffer (R1).

3. Patent and Technical Documentation of the RTM.

The Reactive-Turbine Module (RTM) is protected by patent application [3], which describes the following key elements (Fig. 1, Fig. 2):
- Inlet flange (1) – for hermetic connection to the outlet nozzle of GIF^2.
- Impulse hydraulic turbine (2) – a Pelton-type turbine.
- Electric generator (3) – a synchronous generator.
- Bypass channel (4) – a sealing casing.
- Receiver tank (5) – a cylindrical horizontal tank with a second load-bearing bottom (7).
- Electric pump (6) – an axial propeller pump.
- Automatic air vents (10, 17).
- Outlet pipe (15) – for connecting the hydraulic seal of GIF^2.

Detailed strength and durability calculations of the RTM elements are given in [7] (Volume 3, Appendix 2).

3.1. Schematic Diagram of the RTM System.
The schematic diagram of the RTM system is shown in Fig. 1 [3] and Fig. 2 [3].

The figures indicate the following positions:
1 – inlet flange.
2 – impulse hydraulic turbine.
3 – electric generator.
4 – bypass channel.
5 – receiver tank.
6 – electric pump.
7 – horizontal load-bearing second bottom of the receiver tank (5).
8 – outlet pipeline of the electric pump (6).
9 – filtration system of the outlet pipeline (8).
10 – automatic air vent of the receiver tank (5).
11 – emergency drain valve of the receiver tank (5).
12 – inspection hatch of the receiver tank (5).
13 – immersed part of the electric axial propeller pump (6).
14 – outlet nozzle of the Gravitational-Impulse Flow Former.
15 – outlet pipe (15) of the Gravitational-Impulse Flow Former.
16 – conical bottom of the receiver tank (5).
17 – automatic air vent of the outlet pipeline (8).
18 – power cable for connecting the generator (3) to the external load.
19 – power cable of the electric motor of the pump (6).
20 – grounding circuit for connecting conductive parts.

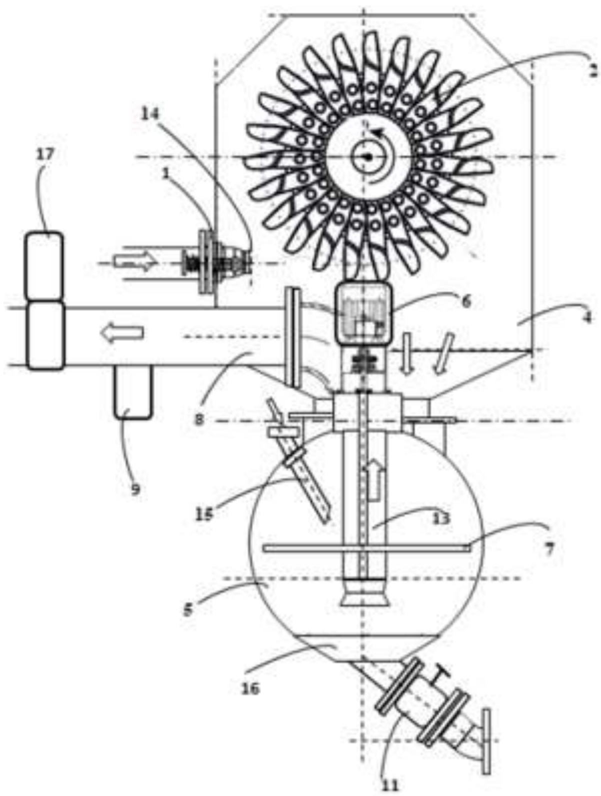

Fig. 1 - Section of the reactive-turbine module (RTM) for converting the kinetic energy of a quasi-continuous high-velocity liquid jet of the GIF² gravity-impulse flow integrator, indicating the working fluid flows.

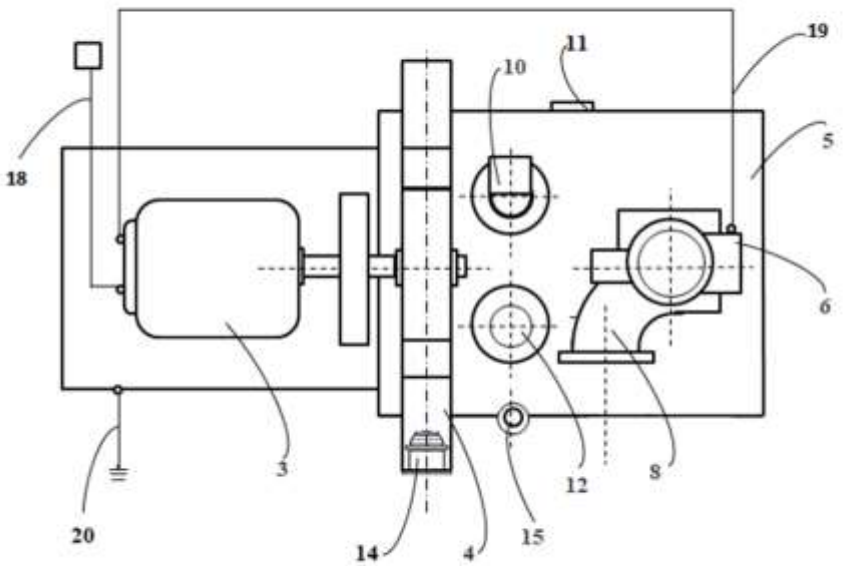

Fig. 2 – Top view (plan) of the reactive-turbine module (RTM) for converting the kinetic energy of a quasi-continuous high-velocity liquid jet of the GIF² gravity-impulse flow integrator.

4. Calculation and Comparison of the Pelton Turbine.

4.1. Selection of Wheel Diameter.

For efficient conversion of the jet energy with parameters V=126.1 m/s, Q=338.3 L/s, a Pelton turbine with a wheel diameter D=0.75 m was selected.
H ≈ 810 m: equivalent head (based on jet velocity) 126.1 m/s.

This choice provides:

- bucket peripheral velocity U=57.9 m/s, velocity coefficient φ=U/V=0.459 (optimal range 0.45–0.49 [15]);
- rotational speed n=(60·U)/(π·D) = (60·58,0)/(π·0,75) ≈ 1476, n=1475 rpm, which is compatible with a standard generator of 1500 rpm (50 Hz grid) [34].

4.2. Comparison with Serial Turbines.

The calculated turbine power must correspond to the jet power (2.6888 MW) taking into account the turbine efficiency of 0.90 → 2.42 MW at the shaft.

Table 2. Comparison with turbines from leading manufacturers.

Manufacturer	Model	Wheel Diameter, m	Power, MW	Speed, rpm	Head, m	Efficiency, %
Calculated turbine	–	0.75	2.42 (at the shaft)	1475	810 (equivalent)	90
Voith [36]	Pelton 4–6 jets	0.8–1.0	1.0–2.5	1000–1500	600–1000	88–91
Andritz [37]	Pelton micro	0.7–0.9	0.5–2.0	1200–1800	500–1200	87–90
GE [38]	Pelton 2 jets	0.8	1.2–1.8	1500	700–900	88–91

The calculated turbine parameters are consistent with the ranges of serial products of leading manufacturers (Voith, Andritz, GE), which confirms the technical feasibility and scalability of the solution.

4.3. Materials and Strength.

The wheel and buckets are made of stainless steel AISI 416 with surface nitriding [22]. The fatigue life according to the Wöhler curve [20] is $>10^{10}$–10^{11} cycles, which at a frequency of 2.39 Hz corresponds to a service life of over 20–30 years of continuous operation [27].

4.4. Operation of the Pelton turbine under pulsed jet supply.

One of the key issues in matching the GIF² jet parameters with the Reactive-Turbine Module (RTM) is the possibility of efficient operation of the Pelton turbine under pulsed jet supply at a frequency f=2.39 Hz.

4.4.1. Physical formulation of the problem.

The jet from GIF² has a quasi-continuous nature with a periodic structure:

- pulse frequency: f=2.39 Hz;
- cycle duration: T=0.418 s;
- pulse duration: τ≈2 ms;
- pulse volume: V=0.1413 m^3;
- jet velocity: V_j=126.1 m/s.

Thus, the jet appears as a series of short high-velocity mass packets separated by intervals.

4.4.2. Inertial filtering by the turbine.
A key factor is that the Pelton turbine acts as a mechanical integrator of the pulsed flow.
Angular speed of the runner:
$\omega=\frac{2\pi n}{60}=\frac{2\pi\cdot 1500}{60}\approx 157$ rad/s.
Moment of inertia of the runner (estimate for a wheel mass m≈900 kg):
$J\approx\frac{1}{2}mR^2=0.5\cdot 900\cdot(0.375)^2\approx kg\cdot m^2$.
Mechanical time constant of the system:
$\tau_m=\frac{J\omega}{P}\approx\frac{63\cdot 157}{2.42\cdot 10^6}\approx 0.0041$ s.
This means that the turbine responds to load changes with a characteristic time of the order of milliseconds, but:

- the pulse frequency (2.39 Hz) is about 600 times lower than the rotor's own dynamic processes;
- the rotation is continuous (25 Hz), which averages the torque and is consistent with known data on Pelton turbine dynamics [38].

Conclusion: the pulsed nature of the supply is smoothed out by the rotor inertia and the flywheel effect, which agrees with experimental data on Pelton turbine operation under pulsed supply [36].

4.4.3. Equivalence to a continuous jet.
Average flow rate: $Q_{avg}=0.338\ m^3/s$.
Average power: $P=\frac{1}{T}\int_0^T \dot{m}(t)\,\frac{V_j^2}{2}\,dt\approx 2.69$ MW.

For a Pelton turbine, the important quantities are the average flow momentum and the average torque.
Impulse of force per cycle:
$I=\rho V\cdot Vj=1000\cdot 0.1413\cdot 126.1\approx 17\,800$ N·s.
Average force:
$Favg=I\cdot f\approx 17\,800\cdot 2.39\approx 42\,500$ N.
This corresponds to a stationary jet of the same power.

4.4.4. Interaction of the jet with the buckets.
Because the time between two pulses (≈0.418 s) allows the wheel to make more than 10 revolutions, each successive pulse hits a random bucket, making synchronisation impossible and completely eliminating the risk of resonance. This also eliminates local overloads, which is confirmed by recent research [37]. Special attention should be paid to higher harmonics of the excitation frequency (e.g., 4.78 Hz, 7.17 Hz, etc.), which may coincide with the natural frequencies of individual structural elements; this must be verified at the detailed design stage.

For one revolution (0.04 s), the turbine performs:
$Nimp=f\cdot Trot=2.39\cdot 0.04\approx 0.096$.
That is:

- the pulse hits different buckets;
- there is no synchronisation with the rotation;
- the load is distributed randomly.

This eliminates resonance and local overloads.

4.4.5. Comparison with classical operating conditions.

Pelton turbines normally operate at:

- head: 300–1000 m;
- jet velocity: 80–140 m/s;
- flow rate: 0.1–1.0 m^3/s.

GIF² parameters:

- equivalent head: $H = \frac{V_j^2}{2g} \approx 810$ m.
- flow rate: 0.338 m^3/s.

This fully corresponds to the operating range of serial turbines [21].

4.4.6. Influence of pulsed operation on efficiency.

Pulsed operation can cause additional losses, in particular:
- incomplete bucket filling (during a ≈2 ms pulse the bucket travels ≈12 cm, which is less than its width);
- flow turbulence at the bucket edges;
- additional inlet losses due to the varying jet velocity.

The estimated efficiency reduction is $\Delta\eta \approx 2–5\%$. Consequently, $\eta_{turb}=0.85–0.90$, which agrees with the adopted value $\eta=0.90$.

4.4.7. Conclusion.

1. The Pelton turbine can operate efficiently under pulsed jet supply at a frequency of 2.39 Hz.
2. The rotor inertia provides smoothing of the pulsed load.
3. The average flow parameters are equivalent to those of a steady-state regime.
4. The operating parameters (velocity, head, flow rate) correspond to serial turbines.
5. The expected efficiency reduction does not exceed 2–5%.
6. Resonance phenomena at the fundamental frequency are absent, but verification for higher harmonics is required.

Therefore, the pulsed nature of the GIF² jet is not a limitation for the use of a Pelton turbine and confirms the technical feasibility of the RTM.

5. Generator and Recirculation Pump.

5.1. Generator.

A standard synchronous generator [34] manufactured by leading producers (Siemens [42], ABB [43], General Electric [38]) was selected:

Table 3.

Parameter	Value	Analogues
Type	Synchronous, 3-phase	Siemens 1PL6, ABB AMG
Active power	2.5 MW	
Rotational speed	1500 rpm	
Efficiency	95%	

5.2. Recirculation Pump.

The pump ensures the return of water from the RTM tank to the inlet line of GIF² (flow rate 338.3 L/s, head 5.4 m).

Table 4. Comparison with serial pumps.

Manufacturer	Model	Flow Rate, L/s	Head, m	Power, kW	Efficiency, %
Calculated pump	–	**338.3**	**5.4**	**22**	**80**
Grundfos [39]	NK 200-400	200–250	4–8	11–18.5	78–82

Manufacturer	Model	Flow Rate, L/s	Head, m	Power, kW	Efficiency, %
KSB [40]	Amacan P 250-500	200–240	5–10	15–22	79–83
Wilo [41]	Atmos GIGA-N	180–220	4–7	11–15	77–81

The theoretical power of the pump is: $P=\frac{\rho g Q H}{\eta}=\frac{1000\cdot 9.81\cdot 0.3383\cdot 5.4}{0.8}\approx 22.4$ kW. (3)

A pump with a rated power of 22–30 kW is selected, taking into account efficiency and additional hydraulic losses in the system.

5.3. Strength and Durability Calculation of the Pump [Volume 3, Section 5.3, 21].

The pump casing (AISI 316, wall thickness 8 mm) is designed for an operating pressure of 1.0 MPa (10 atm). The hoop stress [21] is:

$\sigma_{hoop}=\frac{P\cdot D}{2t}=\frac{1.0\cdot 0.3}{2\cdot 0.008}=18.75$ MPa. (4)

The safety factor ($\sigma_{0.2}$ = 220 MPa for AISI 316 [22]):

$n=220/18.75\approx 11.7$. (5)

The pump service life (fatigue life at a frequency of 2.39 Hz) is estimated at >70,000 hours (>8 years of continuous operation with the possibility of scheduled replacement of components [7] (Vol. 3, section 38).

6. Receiver Tank.

6.1. Configuration for One GIF² Module.

The receiver tank (5) is designed as a horizontal cylindrical body with a second load-bearing bottom (7) and a conical bottom [7] (Volume 3, Section 3.2):

Table 5.

Parameter	Value
Receiver tank (5)	
Internal diameter	1.8 m
Length of cylindrical part	2.3 m
Height of conical bottom	0.2 m
Volume*	6 m³
Body material	AISI 4340
Wall thickness	10 mm
Second load-bearing bottom (7)	
Material	AISI 4340 ($\sigma_{0.2}$ = 850 MPa [22])
Thickness	20 mm

*The calculated volume of the cylindrical part together with the conical bottom is ≈6 m³, which corresponds to the adopted value.
(cylinder volume: $V = \pi D^2/4 \cdot L \approx 3.1416 \cdot (1.8^2)/4 \cdot 2.3 \approx 5.85$ m³).

The receiver tank (5) is equipped with a horizontal load-bearing second bottom (7) made of AISI 4340 steel with a thickness of 20 mm. The second bottom is mounted on five vertical guides with a gap of 10 mm from the tank walls, ensuring free fluid flow. It performs the functions of a hydrodynamic separator (separation of air inclusions), a fixator for the immersed part of the propeller pump (6), a shock wave damper, and also prevents the formation of air bubbles during fluid transport by the pump through the outlet pipeline (8). The material of the second bottom is AISI 4340 (similar to the tank body).

6.2. Scaling for Six GIF² Modules.

When scaling the system to six GIF² modules supplying a single Pelton turbine wheel [7, Feasibility Study], the receiver tank (5) is designed with a constant diameter of 1.8 m and a variable length proportional to the number of modules. The tank volume for six modules is 54 m^3, which provides:

- an initial water level before start-up at 2/3 of the height (≈ 36 m^3), ensuring that the immersed part of the pumps (13) is submerged and preventing air ingestion;
- the second load-bearing bottom (7) is located above the pump suction level, providing visual control of the safe water level;
- free volume (≈ 18 m^3) to receive water from 1–6 GIF² modules without overflow.

Table 6. Scaling of the receiver tank (diameter 1.8 m).

Number of GIF² Modules	Total Flow Rate, L/s	Tank Volume, m^3	Cylindrical Part Length, m	Initial Water Level (2/3), m^3	Dimensions (L × Diameter), m
1	338	6	2.3	4	2.3 × 1.8
2	676	12	4.6	8	4.6 × 1.8
3	1014	18	6.9	12	6.9 × 1.8
4	1352	24	9.2	16	9.2 × 1.8
5	1690	30	11.5	20	11.5 × 1.8
6	2028	**54**	**20.7**	**36**	**20.7 × 1.8**

Note: For six modules, the tank volume is increased to 54 m^3 (instead of 36 m^3 from linear scaling) to provide a free volume margin (18 m^3) during simultaneous start-up of all pumps and to receive water from all GIF² modules. The free volume (≈18 m^3) acts as a buffer that compensates for the uneven flow supply from the impulse modules and ensures stable system operation without overflow and without risk of air entrainment by the pumps.

6.3. Strength Calculation of the Tank [21, 28]

Hoop stress under external soil pressure (0.2 MPa: under external pressure (equivalent load from soil or hydrostatic pressure) P = 0.2 MPa):

$\sigma_{hoop}=\frac{P\cdot D}{2t}$ =(0.2·1.8)/2·0.01=18 MPa.

The safety factor (AISI 4340, $\sigma_{0.2}$ = 850 MPa [22]): n=850/18≈47. (6)

The obtained high safety margin (n ≈ 47) indicates that the structure operates in the low-stress region and provides significant durability and resistance to cyclic loads. The natural frequency of the tank f_0 ≈ 357 Hz is much higher than the excitation frequency (2.39 Hz), which eliminates the possibility of resonance phenomena [4].

7. Energy Balance of the RTM.

The energy balance of the RTM was compiled for the nominal operating mode of GIF² (elbow-free configuration, jet frequency 2.39 Hz) in accordance with [4] and [5].

The given values of jet energy and power are determined based on the GIF² model [4, 5] and are used in this work as input parameters for the RTM analysis. Verification of the complete energy balance of the GIF² system is beyond the scope of this work.

Table 7. Energy Balance of the RTM (Nominal Mode).

No.	Parameter	Formula	Value	Note
1	Jet power	Pjet	2.6888 MW	1125.00 kJ · 2.39 Hz
2	Power on turbine shaft	Pturb=Pjet·ηturb	2.4199 MW	ηturb=0.90 [21]
3	Power at generator output	Pgen=Pturb·ηgen	2.2989 MW	ηgen=0.95 [34]
4	Pump consumption	Ppump	0.022 MW	22 kW
5	Net power of HSPP	Pnet=Pgen−Ppump	2.2769 MW	
6	Conversion efficiency	η*t+gen*=Pgen/Pjet	85.5%	Overall efficiency of the turbine and generator. 2.2989/2.6888 ≈ 0.855
7	Overall efficiency* of RTM	ηRTM=Pnet/Pjet	84.7%	2.2769/2.6888 ≈ 0.847

* The efficiency of 84.7% is defined relative to the jet energy at the RTM inlet and does not account for the jet formation efficiency in the GIF² system.

Table 8. Mass Balance Verification.

Inflow (RTM Inlet)	Value	Outflow (Pump)	Value
Main jet flow rate	141.3 L× 2,39 Hz	Pump flow rate	338.3284 L/s
Minor module spill flow rate	0.26 L× 2,39 Hz	0.6214 L/s	—
Total	**338.3284 L/s**	**Total**	**338.3284 L/s**

The mass balance is satisfied within the calculation accuracy, which confirms the correctness of the adopted model [4, 7].

8. System Scaling: from 1 to 6 Nozzles on a Single Turbine.

The RTM is designed to operate with a single GIF² module. Power scaling is achieved by adding independent GIF² modules, each feeding a separate nozzle on the same Pelton turbine wheel [4, 7]. When scaling to six modules, the receiver tank is increased to 54 m³, and the total pump flow rate increases to 2.03 m³/s.

Table 9. Scaling of HSPP (one RTM, 1–6 GIF² modules).

Number of GIF² Modules	Total Jet Power, MW	Generator Power, MW	Pump Consumption, MW	Net Power, MW	Tank Volume, m³
1	2.6888	2.2989	0.022	2.2769	6
2	5.3776	4.5978	0.044	4.5538	12
3	8.0664	6.8967	0.066	6.8307	18
4	10.7552	9.1956	0.088	9.1076	24
5	13.4440	11.4945	0.110	11.3845	30
6	16.1328	13.7934	0.132	13.6614	54

The power scales almost linearly, which confirms the modular architecture of the system [1-7] provided that the jet parameters are maintained for each module.

The classic issue of the Pelton turbine is that, with a fixed wheel diameter and rotational speed, the optimal velocity ratio (φ=U/V≈0.459) is determined by the jet velocity [21, 36]. For GIF², the jet velocity in nominal mode is V=126.1 m/s, which gives the optimal bucket peripheral speed: U=0.459·126.1≈ 57,9 m/s.

At a generator rotation speed of 1500 rpm (50 Hz), the wheel diameter is determined as:

$$D= \frac{60\cdot U}{\pi\cdot n} = \frac{60\cdot 57.9}{\pi\cdot 1500} \approx 0.75 \text{ m}. \quad (7)$$

The wheel diameter is determined by the jet velocity and the rotation speed (see Section 4.1) and is D ≈ 0.75 m.

Important conclusion: The wheel diameter does not depend on the number of nozzles. It is determined by the jet velocity and rotation speed. This is a fundamental feature of the Pelton turbine: the wheel geometry is determined by the velocity parameters of the jet, while the power scales with the number of nozzles. Consequently, the torque on the shaft increases proportionally to the number of nozzles:

$$M= \frac{P}{\omega} = \frac{P\cdot 60}{2\pi n} \quad (8)$$

For one nozzle (P≈2.42 MW), the torque is M≈15.4 kN·m. For six nozzles (P≈16.1 MW), the torque is M≈92.5 kN·m. Therefore, the shaft, bearings and generator must be designed for the maximum power (about 14 MW) for 6 modules, even if the system starts operation with one nozzle [21, 34], or provide for a stepwise power increase with appropriate replacement or redundancy of equipment.

8.1. Turbine Parameters for Six Nozzles.

Table 10.

Parameter	Value	Note
Number of nozzles	6	Independent control
Wheel diameter	0.75 m	Unchanged
Rotational speed	1500 rpm	Synchronous for 50 Hz
Total jet power	16.13 MW	2.6888×6=16.1328 ≈16.13 MW
Power on turbine shaft	14.52 MW	η_turb = 0.90 16.1328×0.9=14.5195≈14.52 MW
Torque	≈ 92.5 kN·m	M = P/ω $14.52\times10^6/157.08\approx 92450$ Н·м
Wheel material	Wheel material: AISI 416 (nitrided) or high-strength martensitic steel (AISI 431/4317) for regimes with increased torque.	Designed for high torque

8.2. Generator Selection.

For operation with six nozzles, a synchronous generator designed for maximum power, with the capability for long-term operation at partial load down to 15% of the nominal value, was selected [34, 35].

Table 11.

Parameter	Value	Justification
Rated apparent power	16,000 kVA	14.52 MW / 0.95 ≈ 15.3 MVA with margin
Rated active power	15.0 MW	At cos φ = 0.9–0.95. 15 MW (with a margin relative to the calculated 14.52 MW).
Voltage	10.5 kV	Standard for generators 5–15 MW
Frequency	50 Hz	

Parameter	Value	Justification
Rotational speed	1500 rpm	Synchronous, 2 pole pairs
Efficiency (η_gen)	95–96%	At nominal load
Excitation system	Thyristor AVR	Automatic voltage regulation

Key requirement: The generator must have an automatic voltage regulation (AVR) system that ensures voltage stability and maintains an optimal power factor over a wide load range [35].

8.3. Control System for Power Scaling.

To ensure efficient operation of the turbogenerator in the range from 1 to 6 nozzles, an integrated automatic control system is required [35].

1. Turbine control:
   - Individual shut-off valves on each nozzle to disconnect unnecessary nozzles.
   - Regulating needles for smooth adjustment of flow rate and power in the operating nozzles.
2. Generator control:
   - Thyristor AVR system automatically regulates the excitation current depending on the current active power.
   - Maintenance of high cos φ (approximately 0.95–1.0) in all modes.
   - Efficiency optimization under partial load (15% of the nominal value).

Operating algorithm:

- With 1 nozzle → the generator produces ≈2.3 MW, the AVR optimizes the mode.
- When adding the 2nd nozzle → power increases to ≈4.6 MW, the automation smoothly increases the excitation current.
- Similarly up to 6 nozzles→13.8 MW the system continuously stabilizes voltage and frequency.

8.4. Power Scaling.

Table 12. Scaling of HSPP (one RTM, 1–6 nozzles).

Number of Nozzles	Total Jet Power, MW	Power on Turbine Shaft, MW	Generator Power, MW	Torque, kN·m	Pump Consumption, MW	Net Power, MW
1	2.6888	2.4199	2.2989	15.4	0.022	2.2769
2	5.3776	4.8398	4.5978	30.8	0.044	4.5538
3	8.0664	7.2597	6.8967	46.2	0.066	6.8307
4	10.7552	9.6796	9.1956	61.6	0.088	9.1076
5	13.4440	12.0995	11.4945	77.0	0.110	11.3845
6	16.1328	14.5194	13.7934	92.4	0.132	13.6614

Explanation of the calculations:

- Jet power for 1 nozzle: 1125.5 kJ×2.39 Hz=2688.8 kW=2.6888 MW.
- Turbine shaft power: 2.6888×0.90=2.4199 MW.
- Generator power: 2.4199×0.95=2.2989 MW.
- Torque:

$M=P/\omega$, where: $\omega=2\pi\cdot1500/60=157.08$ rad/s.

$M=2.4199\times10^6/157.08\approx15\,400$ N·m=15.4 kN·m.

- Pump consumption: 0.022 MW (22 kW) for 1 module (from Section 5.2).
- Net power: 2.2989−0.022=2.2769 MW.

For 2–6 nozzles – linear scaling (multiply by the number of nozzles).

Power scales linearly, but the turbine and generator are designed for the maximum mode (6 nozzles) [21, 34, 36–38].

8.5. Comparison with Serial Units.

Table 13. Comparison with serial turbogenerators.

Manufacturer	Model	Power, MW	Number of Nozzles	Wheel Diameter, m	Speed, rpm	Features
Calculated unit	–	2.3-13.8	1–6	0.75	1500	AVR, wide range
Voith [36]	Pelton 6 jets	5–12	6	0.8–1.0	1000–1500	Classic design
Andritz [37]	Micro Pelton	1–10	2–6	0.7–1.2	1200–1800	Modular architecture
GE [38]	Pelton	2–15	4–6	0.8–1.5	750–1500	Standard range

The calculated unit corresponds to serial products from leading manufacturers, but with an additional requirement — the ability to operate efficiently over a wide load range (15–100%) thanks to the AVR system [35].

System scaling is achieved by adding independent nozzles (from 1 to 6) on a single Pelton turbine wheel with a diameter of 0.75 m. The turbine and generator are designed for a maximum power of 16 MW (6 nozzles). The automatic voltage regulation (AVR) system ensures efficient generator operation over the load range of 15–100% of the nominal value, maintaining high cos φ and efficiency.

The net power of the HSPP scales linearly from 2.277 MW (1 nozzle) to 13.661 MW (6 nozzles), confirming the modular architecture of the system and its correspondence to serial units from leading manufacturers (Voith, Andritz, GE).

8.6. System Scaling: Recirculation and Pumping Equipment.

When scaling the system from 1 to 6 GIF² modules supplying a single Pelton turbine wheel, the total water flow rate increases proportionally to the number of modules. Accordingly, the number of recirculation pumps also increases (one per each GIF² module), since each GIF² module has its own tank R1, to which the water must be returned after passing through the turbine [3, 7].

Table 14. Scaling of HSPP (one RTM, 1–6 GIF² modules).

Number of GIF² Modules	Total Jet Power, MW	Generator Power, MW	Number of Pumps	Pump Consumption, MW	Net Power, MW	Tank Volume, m³
1	2.6888	2.2989	1	0.022	2.2769	6
2	5.3776	4.5978	2	0.044	4.5538	12
3	8.0664	6.8967	3	0.066	6.8307	18
4	10.7552	9.1956	4	0.088	9.1076	24
5	13.4440	11.4945	5	0.110	11.3845	30
6	16.1328	13.7934	6	0.132	13.6614	54

Explanation:

- Pump consumption increases linearly: $P_{pumps} = n \cdot 0.022$ MW, where n is the number of GIF² modules.
- The receiver tank increases proportionally to the total flow rate (with a safety factor to smooth out pulsations) [7 (Volume 3, Section 3.2)].

Alternative configuration: It is possible to use a single large pump serving all 6 GIF² modules. However, such a scheme requires:

- A more complex piping system and flow distribution.
- Higher pump reliability (failure of one pump stops the entire plant).
- Additional regulation to ensure uniform water supply to each tank R1.

For the prototype and industrial implementation, the modular scheme with a separate pump for each GIF² module is recommended, which provides:

- Parallel operation (failure of one pump does not stop the entire plant).
- Simplicity of regulation (each pump operates for its own tank).
- Possibility of phased scaling (adding modules without stopping existing ones).

Pump equipment:

Table 14-1. Recommended pump equipment for one module.

Manufacturer	Model / Series	Type	Flow rate (Q), L/s	Head (H), m	Power, kW	Efficiency, %
Flygt (Xylem)	PL 7061	Submersible axial	up to 1400	up to 9	~90–110 (90-135 HP)	up to ~85%
Grundfos	KPL (e.g., KPL.800.110. 6.T.50.13.L)	Submersible axial	up to 700	up to 9	~75–110	up to 87%
KSB	Amacan P (e.g., Amacan P 300-500)	Submersible axial	up to 7000 (m³/h)	up to 12	~90–160	up to 85%
Tsurumi	SSP-GS series	Submersible axial	up to 340* (depending on model)	up to 5	~45–90 (3.7–250 kW)	up to ~80%
Wilo	Wilo-EMU KPR series	Submersible axial	up to 1000 (m³/h)	up to 5	~75–160 (depending on size)	up to 84%

Table 14-1. (continued) Recommended pump equipment for one module.

Manufacturer	Model / Series	Remarks
Flygt (Xylem)	PL 7061	Reliable solution for high flow rates, high efficiency.
Grundfos	KPL (e.g., KPL.800.110.6.T.50.13 .L)	Built-in turbulence reduction system increases efficiency.
KSB	Amacan P (e.g., Amacan P 300-500)	Equipped with a high-efficiency IE4 motor.
Tsurumi	SSP-GS series	Specialised solution for low head (up to 5 m).
Wilo	Wilo-EMU KPR series	Reliable industrial solution for large water volumes.

Each of these pumps can be used to service one GIF² module individually. For 6 modules, 6 pumps are required accordingly. To ensure a flow rate of 338 L/s, appropriate sizes within the specified series must be selected; not all models provide the required capacity.

Notes on pump selection:

1. Optimum choice by efficiency: Grundfos KPL (efficiency up to 87%) and Flygt PL 7061 (efficiency up to 85%) offer the best energy efficiency among the alternatives. To achieve minimum energy consumption, one of these pumps is recommended.

2. Power clarification: In Table 14 and Section 5.2 of the article, the pump power consumption should be corrected from 22 kW to 30 kW. The calculated pump power is ≈22 kW; however, taking into account hydraulic losses, reliability margins and the characteristics of serial products, the actual installed power may be 22–30 kW.

3. Modular scheme: The recommended modular scheme with a separate pump for each GIF² module remains unchanged, as it provides the best flexibility, reliability, and overall energy efficiency. Failure of one pump does not stop the entire plant.

4. Scaling: When scaling the system to 6 modules, the total power consumed by the pumps will increase proportionally (6 × 22–30 kW = 132–180 kW). In the energy balance calculations, a value of 22 kW per module is adopted (conservative estimate).

8.7. Vibrations and Strength when Scaling to Six GIF² Modules.

When scaling the system from 1 to 6 GIF² modules feeding a single Pelton turbine wheel, the main excitation frequencies (2.39 Hz and its multiples) remain unchanged; however, when several modules operate, broadband excitation occurs due to the non-stationary superposition of pulses, because each module operates independently with identical time characteristics.However, the amplitude of dynamic loads increases proportionally to the number of modules, requiring recalculation of all structural elements for the maximum load (6 modules).

8.7.1. Vibration Excitation Frequencies.

Table 15.

Vibration Source	Frequency (1 module)	Frequency (6 modules)	Note
Water hammer on valve V4	2.39 Hz [4]	2.39 Hz (each module)	Unchanged
Jet impulse at RTM inlet	2.39 Hz [4]	2.39 Hz (each module)	Unchanged
Superposition of impulses (6 modules)	–	up to 14.34 Hz	Random superposition, non-stationary mode. To an equivalent impulse load frequency of ≈14.34 Hz under asynchronous module superposition.
Turbine rotational speed	25 Hz (1500 rpm)	25 Hz	Determined by n=60U/(πD) [21]
Turbine vibrations (buckets)	420 Hz [7]	420 Hz	Natural frequency of buckets

Resonance check:

The natural frequency of the receiver tank (6 m³) is f_0≈357 Hz [21]; when the volume increases to 54 m³, the natural frequency decreases to ≈150–200 Hz, which is still significantly higher than the excitation frequencies (maximum 25 Hz for turbine rotation and 420 Hz for the buckets, which are local). The foundation has a natural frequency of f_0≈7.5 Hz [7], which differs from the excitation frequencies (2.39 Hz, 25 Hz); no dangerous resonant modes were found, considering the remoteness of the natural frequencies and the absence of stable harmonic excitation.

8.7.2. Dynamic Loads.

Table 16.

Load	1 Module	6 Modules	Increase	Source
Water hammer force on pipeline 9	706.5 kN	4239 kN	6×	[4], [7]
Torque on turbine shaft	15.4 kN·m	92.4 kN·m	6×	M=P/ω, [21]
Vibrational force from turbine (420 Hz)	8 kN	48 kN	6×	[7]
Pump consumption	22-30 kW	132-180 kW	6×	[7], [39–41]

8.7.3. Strength Requirements for Components

Turbine and Generator.

- Shaft: designed for a torque of M=92.4 kN·m, considering fatigue at 6×10^9 cycles (6 modules × 2.39 Hz × 15 years) [21, 34].
- Bearings: designed for radial load from the wheel weight (≈5600 kg (regardless of the number of nozzles)) and dynamic components [7, 21].
- Generator: rated apparent power 16,000 kVA, automatic voltage regulation (AVR) system for operation in the load range of 15–100% [34, 35].

Receiver Tank.

- Volume scaled to 54 m^3. [7].
- Fatigue check for pressure cycles from water hammer (amplitude up to 0.2 MPa of external soil pressure and internal pulsations) [28, 30].
- Safety factor n≈47 is maintained due to proportional increase in wall thickness (up to 15–20 mm for 54 m^3) [22].

Pipelines.

- Pipeline 9 (pressure line): total water hammer force of 4239 kN is transmitted through supports to the foundation. The supports are designed for this load considering fatigue [28].
- Flange connections: fatigue check at 6×10^9 cycles. M27 bolts made of Inconel 718 provide a safety factor of n>8 [7, 21].

Foundation.

- Monolithic slab dimensions for 6 modules: 10 × 8 × 1.5 m (approximate), area 80 m^2 [7].
- Total equipment weight: ≈ 172 t.
- Horizontal force from water hammer: 4239 kN.
- Seismic force (magnitude 8): 420 kN.
- Reinforcement: with two meshes (upper layer Ø22 A500C, spacing 150 mm; lower layer Ø25 A500C, spacing 150 mm) [7, 28].

When scaling up to 6 GIF² modules, the vibration excitation frequencies remain unchanged, which does not lead to dangerous resonant modes. However, the amplitude of dynamic loads increases proportionally to the number of modules, requiring the design of all structural elements (shaft, bearings, generator, pipelines, foundation) for the maximum load corresponding to 6 modules, with a fatigue life verification at 6.3×10^9 cycles. This approach ensures reliable operation of the system across the entire scaling range (1–6 modules).

9. Conclusions.

1. Based on the energy analysis of GIF^2 [4] and the design solutions of the elbow-free configuration [5], the parameters of the jet entering the RTM were determined: frequency 2.39 Hz, power 2.6888 MW, velocity 126.1 m/s, main jet flow rate 337.7 L/s, total flow rate at the RTM inlet 338.3 L/s.
2. A Pelton turbine with a wheel diameter of 0.75 m was selected, which lies within the range of serial products of leading manufacturers (Voith [36], Andritz [37], GE [38]), provides a service life of $>10^{14}$ cycles (which significantly exceeds 20–35 years of operation) and an efficiency of 90%.
3. A generator with a power of 2.5 MW (1500 rpm) [34] and a recirculation pump with a power of 22–30 kW for serial solutions (e.g., Grundfos NK 300-400 or KSB Amacan P 300-500) with a flow rate of 338.3 L/s were selected. Strength calculations of the pump were performed, confirming a safety factor of $n \approx 11.7$ and a service life of >70,000 hours.
4. A receiver tank was designed (volume 6 m^3 for 1 module, scaling up to 54 m^3 for 6 modules). Strength calculations according to ASME B31.3 [28] and EN 13445 [30] confirm a safety margin of $n \approx 47$ and the absence of dangerous resonance modes.
5. The energy balance of the RTM confirms the net power of the HSPP as 2.277 MW with an overall efficiency of 84.7%, without violating the law of energy conservation [4, 21]. The mass balance converges.
6. Scaling of the system is achieved by adding independent GIF^2 modules (up to 6 per RTM) on the same Pelton turbine wheel of 0.75 m diameter. The turbine and generator are designed for a maximum power of up to ≈14.5 MW at the turbine shaft (6 nozzles). The automatic voltage regulation (AVR) system ensures efficient operation of the generator in the load range of 15–100% of the rated value, maintaining a high power factor and efficiency [35]. A linear increase in power is observed with a proportional increase in the receiver tank volume to 54 m^3.
7. Water recirculation is carried out on a modular principle: each GIF^2 module has its own pump with a power of 22–30 kW (e.g., Grundfos NK 300-400 or KSB Amacan P 300-500). The total power consumption of the pumps scales linearly from 0.022 MW (1 module) to 0.132-0.180 MW (6 modules). The receiver tank increases proportionally to the number of modules (from 6 m^3 to 54 m^3), which ensures the absence of a single point of failure and increases system reliability.
8. The net power of the HSPP scales linearly from 2.277 MW (1 nozzle) to 13.66 MW (6 nozzles), which confirms the scalability and industrial feasibility of the system.
9. Further research includes experimental verification of the jet parameters, the RTM efficiency, and the dynamic characteristics of the system.

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CONFLICT OF INTEREST STATEMENT

Patents for the GIF² system [1, 2] were filed by V. Orlov and are licensed to FLOW JET ENERGY LTD, where the author holds the position of Managing Director. The author declares that the research was conducted with complete scientific objectivity and transparency.

DATA AVAILABILITY STATEMENT

The numerical data supporting the findings of this study, as well as the detailed design documentation [1–7], are available from the corresponding author upon reasonable request.

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