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Hybrid energy storage system using post-mining infrastructure (HESS)

HESS

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1 Introduction

This document presents the preliminary design of an underground power-generation unit and pumping system for a pumped-storage power plant located in a decommissioned mining shaft. The work was carried out under the project "Hybrid energy storage system using post-mining infrastructure", co-funded by the RFCS programme. The report summarises the current functional–technical assumptions, preliminary design solutions, and the basic operating parameters of the system in the configuration intended for further analyses and stakeholder alignment.

The project objective is to verify technical feasibility and to develop a preliminary model of the generation unit and pumping system integrated within the shaft, while simultaneously using the shaft as the lower water reservoir. The adopted configuration comprises a Pelton turbine (PV3i-790/160) of about 5.3 MW, supplied by a DN400 penstock divided into pressure zones (PN40/PN63/PN100). The unit operates with a 6.3 kV synchronous generator (parallel grid operation), and flow control is provided by needle nozzles with jet deflectors and a hydraulic power unit (HPU). In pumping mode, staged conveyance of water from intermediate levels to the surface is envisaged, with effective throughput dependent on pump availability and a working storage volume of approximately 12,000 m³.

The scope of this report includes:

- an outline of the mechanical and hydraulic solutions for the turbine-generator set, associated valves/armatures, and the HPU;
- a description and preliminary sizing of the DN400 penstock with pressure-zone division and thermal/structural expansion compensation;
- a concept for space utilisation within the shaft at around −900 m, including service platforms, the machine foundation, and access nodes;
- the use of the shaft as the lower reservoir, including a preliminary volumetric assessment and assumptions for further modelling;
- preliminary operational assumptions (generation/pumping cycles, start-up and shutdown sequences, reliability requirements);
- a characterisation of the water pumping system from the lower reservoir and a description of the upper reservoir.

2 Selection of Turbine Type for Operating Conditions

In selecting the type of turbine for a hydropower plant, two key parameters are of primary importance: the hydraulic head and the water flow rate. Based on these parameters, the optimal type of machine is determined to ensure the highest efficiency, operational stability, and safety (fig. 1). In hydropower practice, four main types of turbines are used: Kaplan, Francis, and Pelton. Each of them features a distinct design and is intended for operation under different hydraulic conditions.

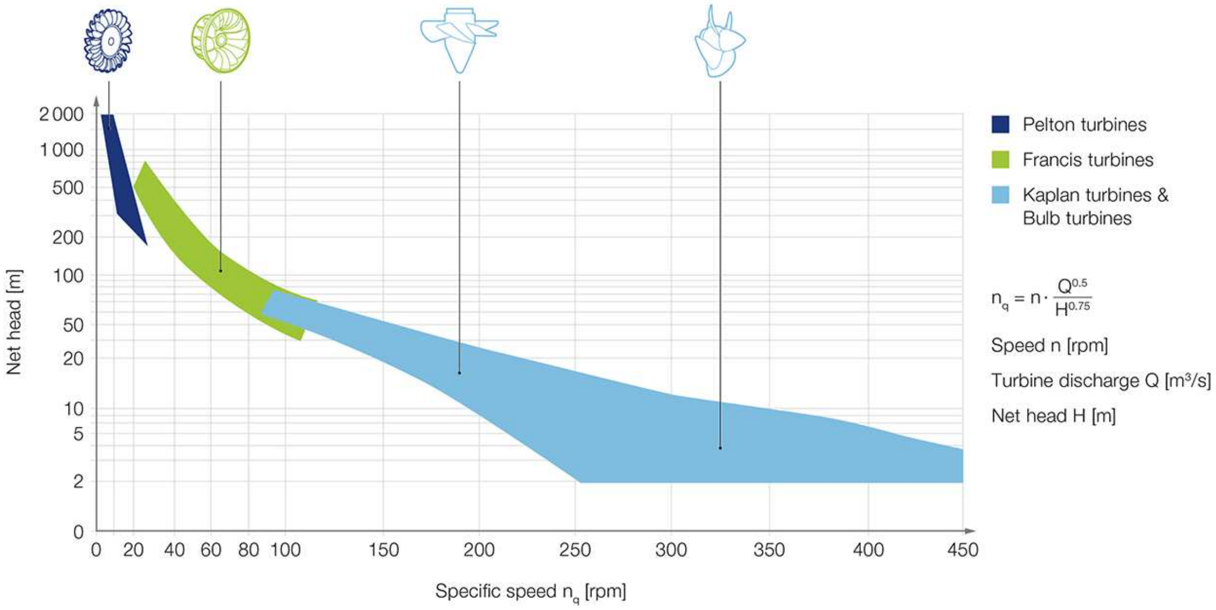

Fig. 1. Application range of different types of turbines [4]

The Kaplan turbine (fig. 2) belongs to the group of axial-flow turbines with adjustable runner blades and guide vanes. Water flows through the runner along the shaft axis, and the ability to change the blade angles allows the turbine to maintain high efficiency over a wide range of flow rates. These turbines are mainly used for low heads (2–70 m) and high discharges (10–1000 m³/s), typical of large rivers or lower dam stages. They are characterized by high efficiency but, due to limited pressure resistance, cannot be used for high heads [3]. Under the conditions of the designed power plant, where the net head is approximately 837.8 m, the Kaplan turbine would be entirely unsuitable because the hydraulic pressure would exceed the structural limits of such a machine.

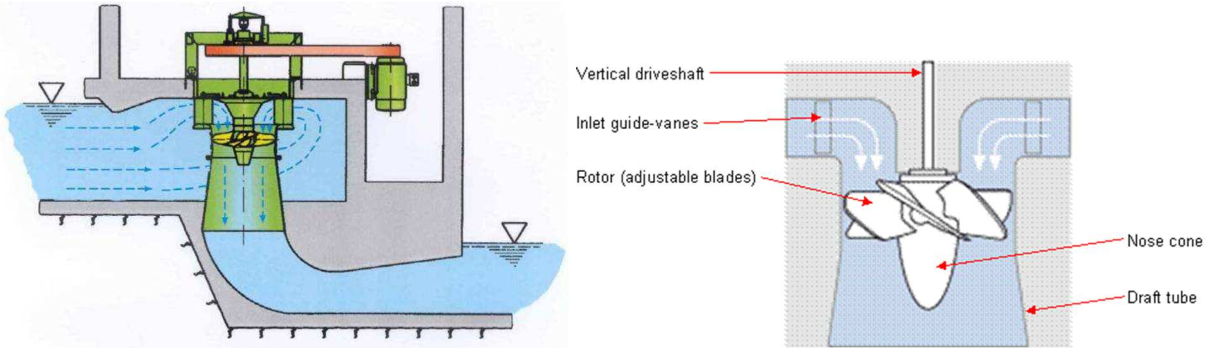

Fig. 2. Cross-section of a typical vertical-axis Kaplan turbine system [3].

The Francis turbine (fig. 3) is a reaction turbine with radial-axial flow, in which water enters the runner through a spiral casing and guide vanes, then flows through the blades in an axial direction. Power regulation is achieved by adjusting the angle of the guide vanes. This type of turbine operates under medium heads (20–300 m) and flow rates ranging from a few to several dozen cubic meters per second, offering very high efficiency and stable operation. However, for heads exceeding 400 m, problems such as cavitation and excessive centrifugal forces occur, which limit their applicability. In the case of a power plant with a head of over 800 m, the use of a Francis turbine would be technically unjustified, as it would require extreme pressure and rotational parameters, leading to structural overload and the risk of mechanical failure.

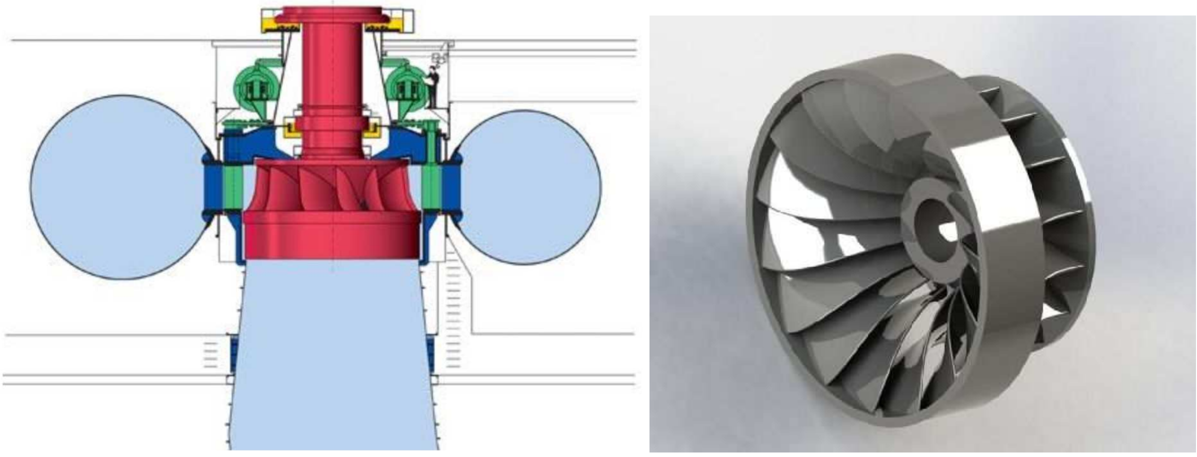

Fig. 3. Francis turbine system [4]

The Pelton turbine, applied in the project, belongs to the group of impulse turbines in which the entire hydraulic energy of water is first converted into kinetic energy in the nozzles and then into mechanical energy on the runner. Water exits the nozzles as narrow, highly pressurized jets that strike the buckets of the runner, transferring energy and setting it into rotational motion. After striking the buckets, the water loses almost all of its energy and flows into the tailrace under atmospheric pressure. The flow is regulated by needle valves in the nozzles, while in emergency situations, the jet can be immediately deflected by deflectors.

Pelton turbines are primarily used in power plants with high heads (200–1500 m) and low flow rates (0.05–2 m³/s), which corresponds precisely to the operating range of the designed unit. Their design enables efficient conversion of the potential energy of high-pressure water into mechanical energy, while maintaining very high efficiency—often exceeding 90%. Since the runner operates in air, the risk of cavitation is eliminated, and the separation of the hydraulic circuit from the mechanical components simplifies the design and facilitates maintenance. High resistance to pressure fluctuations and excellent controllability under variable loads make the Pelton turbine the most reliable solution for high-head pumped-storage power plants.

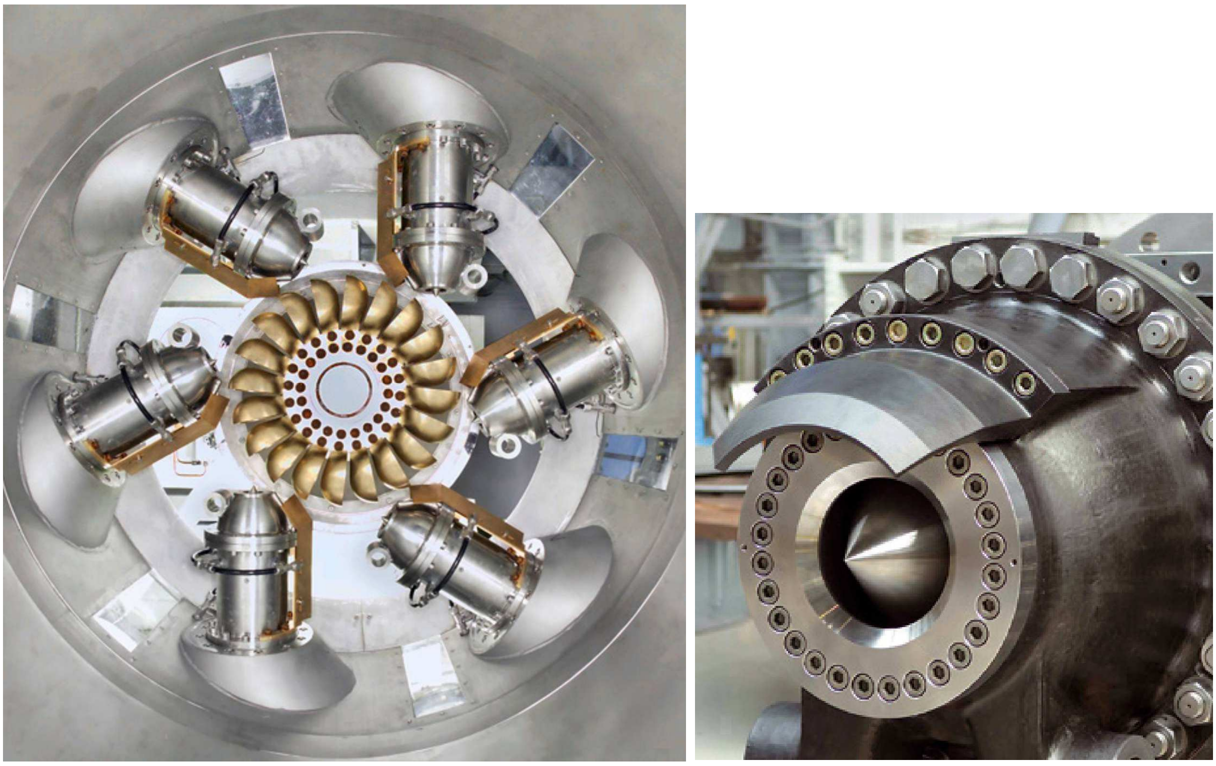

Fig. 4. Pelton turbine and nozzles [6]

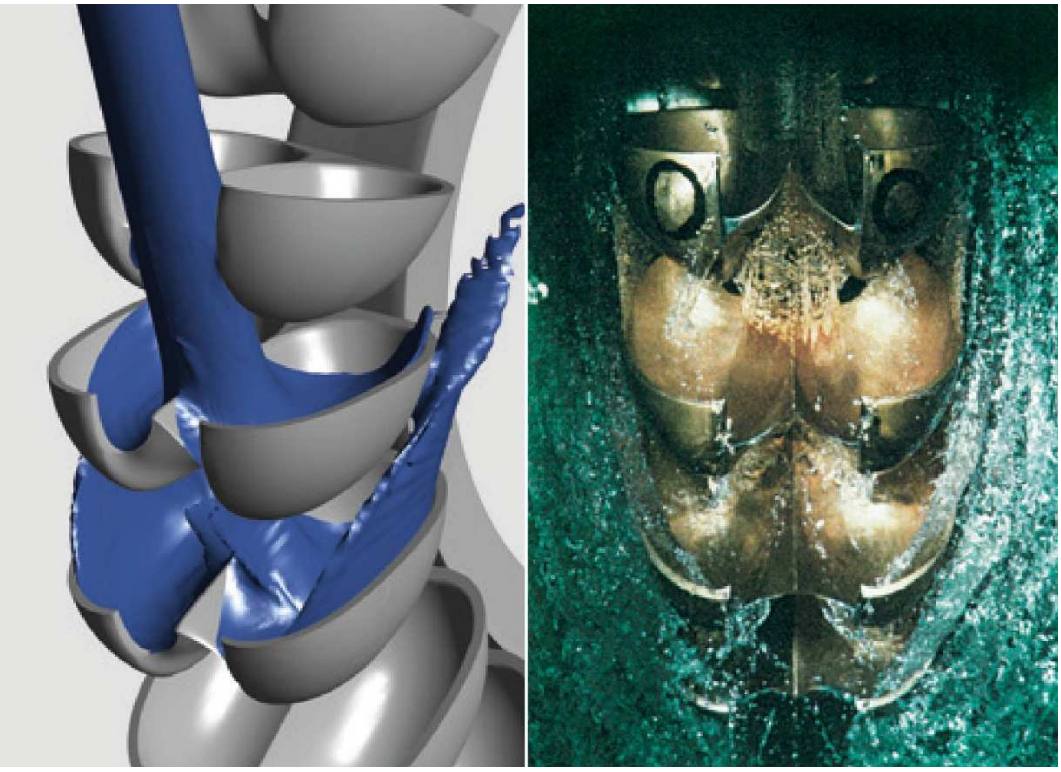

Fig. 5. Simulated flow through Pelton runner and actual flowthrough model [6]

3 System of the hydro turbine and generator with the shaft reservoir.

The key component of the designed pumped-storage power plant located in an inactive mine shaft (fig. 6) is a generating unit consisting of a high-head Pelton turbine coupled with a medium-voltage synchronous generator. The unit operates with a head of approximately 900 m, a flow rate of about 0.71 m³/s, and a rotational speed of 1500 rpm, corresponding to a rated power of around 5.3 MW. The three-nozzle turbine is supplied through a DN 400 pipeline, while the generator, directly coupled to the turbine, delivers power to a 6.3 kV grid, operating in parallel and meeting standard synchronization criteria (frequency, voltage, and phase angle). The structural design of the entire system is presented in the preliminary design prepared by ITG KOMAG, designated as W52.010PW01.

System components:

1. Vertical Pelton turbine,
2. Turbine housing with distribution manifold,
3. Control nozzle with deflector,
4. Synchronous generator,
5. Inlet pipe,
6. Support pipe,
7. Support structure,
8. Pipe section,
9. Gland-type expansion joint,
10. Guiding support,
11. Supply pipe,
12. Ball valve DN400, PN100
13. Hydraulic power unit.

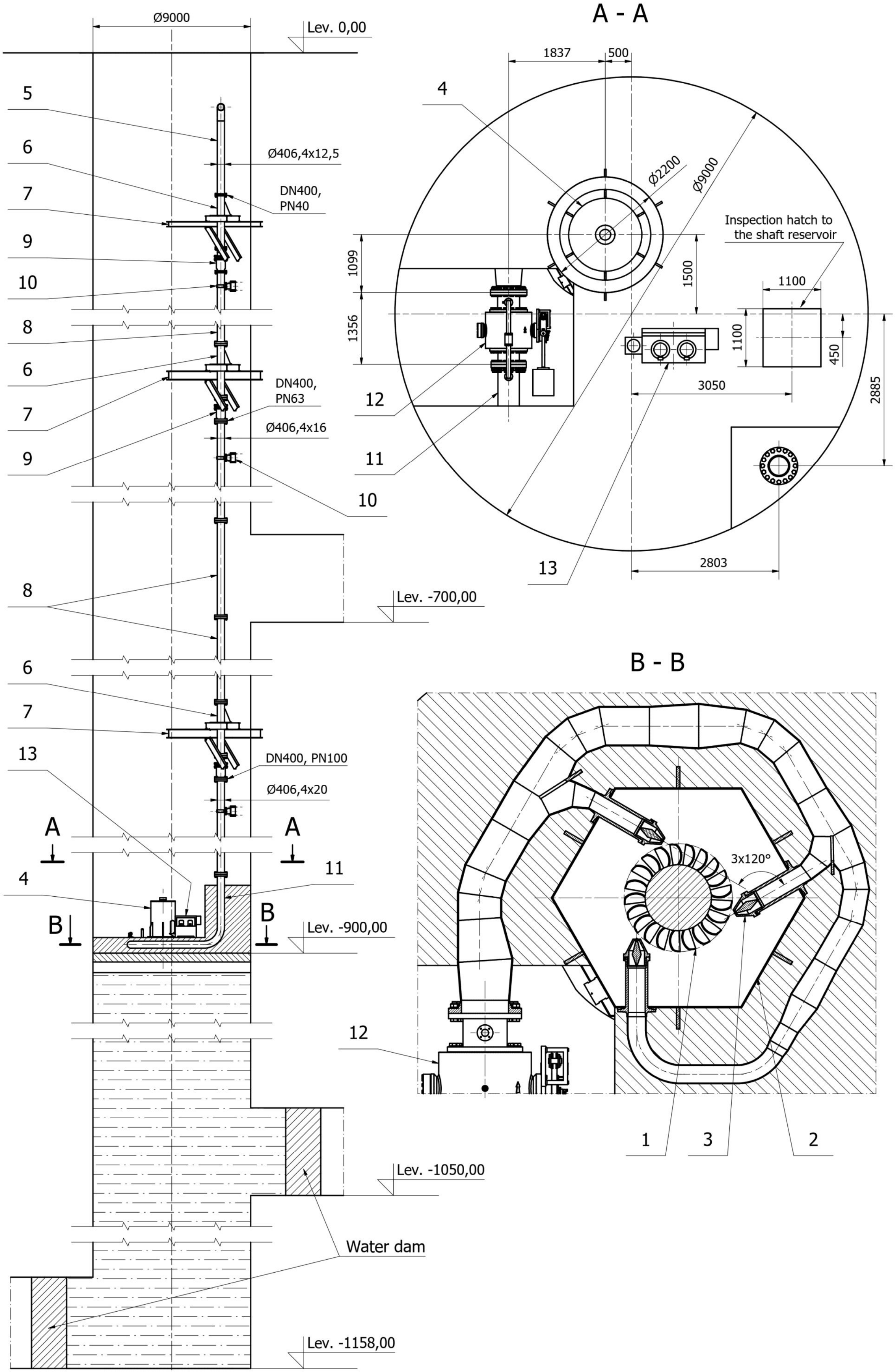

Fig. 6. System of the hydro turbine and generator with the shaft reservoir [2]

Table 1. Technical data [1].

Parameter	**Value**
Rated gross head	900 m
Rated net head	837.81 m
Penstock diameter	DN400 mm
Pressure at turbine inlet	8.22 MPa
Rated discharge	0.71 m³/s
Runner diameter	790 mm
Runner bucket width	160 mm
Number of nozzles	3
Nozzle diameter	175 mm
Generator rated power	5800 kW
Power factor cos φ	0.9
Rated turbine power	5287 kW
Generator rated voltage	6.3 kV
Frequency	50 Hz
Rated turbine efficiency	90.82 % (Q = 0.71 m³/s)
Max turbine efficiency	91.39 % (Q = 0.5 m³/s)
Rated turbine speed	1500 rpm
Turbine overspeed	2786 rpm
Valve design pressure	100 bar
Governor system pressure	max. 12 MPa
Insulation / Temp. rise	F / B

The Pelton turbine is optimal for the designed power plant in the mine shaft, as it operates most efficiently under very high heads and relatively low flow rates ($H \approx 900$ m, $Q \approx 0.71$ m³/s), which precisely corresponds to the project conditions. As an impulse turbine, it is practically immune to cavitation and allows a wide range of power regulation through needle-controlled nozzles and deflectors, maintaining high efficiency even under partial loads. In comparison, the Francis turbine (reaction type) achieves peak efficiency at medium heads but would require extremely high pressures, complex hydraulics, and would be prone to cavitation under such a high head. The Kaplan turbine, on the other hand, is designed for low heads and high flow rates, making it uneconomical and technically unsuitable in this case. Additionally, the multi-nozzle configuration enables fast flow control and reliable operation with the 6.3 kV grid, while the simpler hydraulic layout and the absence of a sensitive reaction runner enhance reliability and operational maintainability.

In the design of hydropower and pumped-storage plants, two fundamental concepts are distinguished: gross head and net head. The gross head (rated gross head) is a geometric value representing the difference in elevation between the water surface in the upper reservoir (or at the pipeline inlet) and the water level in the lower reservoir, where the water is discharged after passing through the turbine. In this project, the value amounts to 900 m and corresponds to the total height of the water column available for utilization, assuming no hydraulic losses. Therefore, the gross head defines the topographic conditions of the site and serves as the basis for sizing the pressure pipeline, selecting fittings, and performing strength calculations for the installation components.

The second parameter is the net head (rated net head), which defines the actual height of the water column available at the turbine runner after subtracting the hydraulic losses that occur during flow through the pipeline and hydraulic equipment. These losses include linear friction losses along the pipeline as well as local losses at bends, valves, nozzles, distributors, and other system components.

For the designed pipeline with a length of approximately 900 m and a nominal diameter of DN400, at a nominal flow rate of Q = 0.71 m³/s, the total head loss amounts to about 62 m. Consequently, the net head determined at the Pelton turbine is approximately 837.81 m. This is the value used for calculating the turbine unit's power, torque, and efficiency parameters.

The distinction between the two values has significant practical importance. The gross head forms the basis for designing the hydraulic infrastructure and defines the boundary conditions for mechanical loads in the pipeline, whereas the net head is the key parameter for selecting and calculating the turbine characteristics. Accounting for the difference between these two heads allows for a realistic assessment of the plant's energy potential and ensures optimal matching of the hydraulic and mechanical systems to the operating conditions.

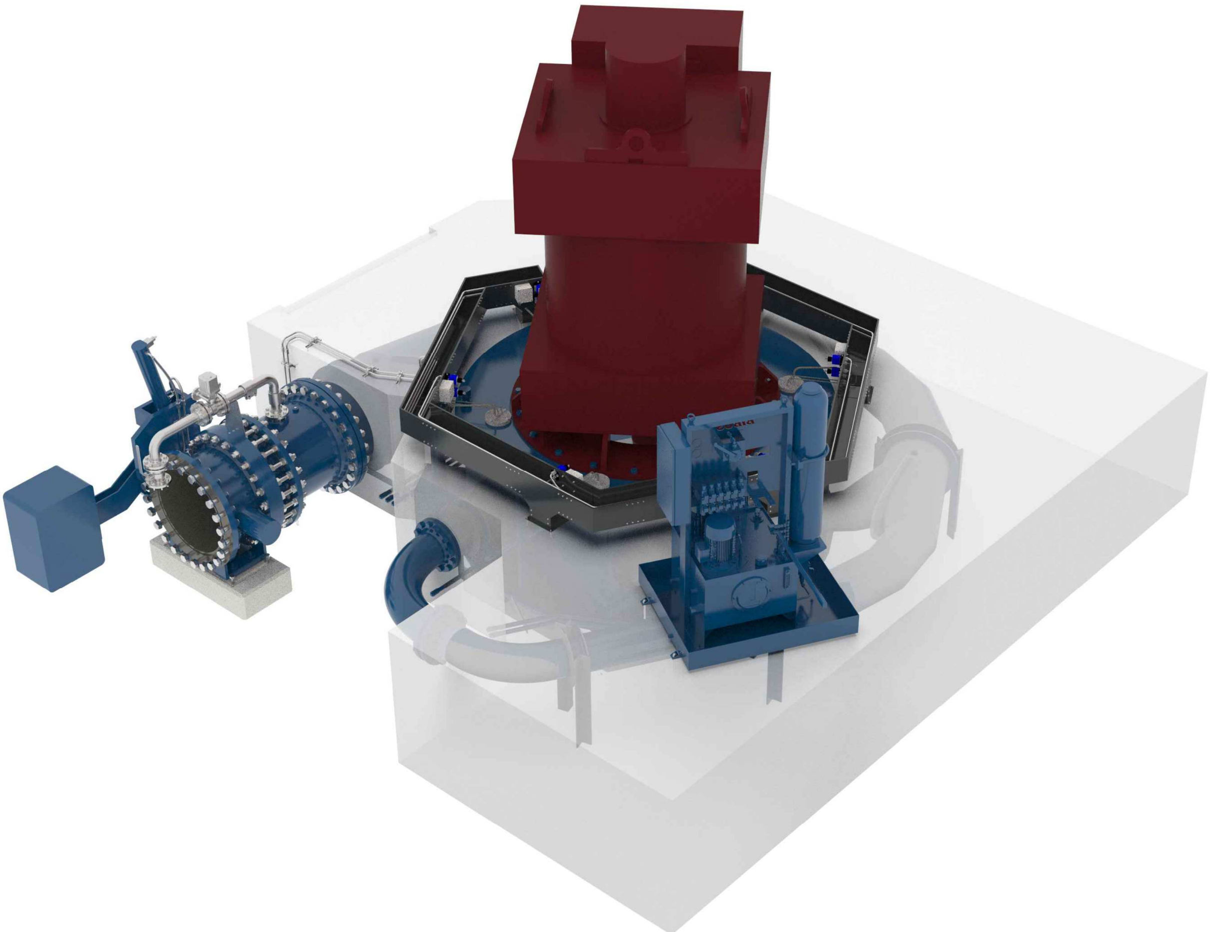

Fig. 7. Turbogenerator unit [1].

The generator (no. 4) together with the Pelton turbine (no. 1) is installed in the mine shaft at a depth of –900 m (fig. 7). Both are mounted on a load-bearing structure in the form of a reinforced concrete slab anchored to the shaft walls. The turbine and generator are enclosed within the turbine casing (no. 2), which is directly mounted on the reinforced concrete slab. The casing includes an integrated collector assembly surrounding the turbine, distributing the working fluid into three jets directed toward the turbine buckets. The collector outlets, arranged evenly around the turbine at 120° intervals, are equipped with needle-type regulating nozzles with deflectors (no. 3). The turbine is supplied with water from the upper (surface) reservoir through a DN 400 penstock with a head of approximately 900 m. A DN 400 spherical shut-off valve (no. 12) is installed directly upstream of the turbine collector. The entire turbine casing with the collector, as well as the terminal section of the penstock, are embedded in concrete to ensure proper anchoring of the generating unit and effective vibration damping. The reinforced concrete foundation slab of the turbine assembly includes an inspection opening providing access to the lower section of the shaft, which serves as the lower reservoir. Its height is approximately 258 m, and with a shaft diameter of 9 m, it provides a total storage volume of about 16.411 m³.

The designed pumped-storage power plant is equipped with two water reservoirs: the upper one, located on the ground surface, and the lower one, formed within the shaft below the load-bearing structure of the turbine assembly. Water flowing from the upper reservoir through the penstock drives the turbine and generator. After passing through the turbine, the water enters the lower (shaft) reservoir. The element that completely shuts off the water flow during the generating unit's downtime is a DN400 spherical valve installed directly upstream of the turbine collector. In the collector distributing the turbine's water stream, the flow is divided into three jets and directed to needle-type regulating nozzles, which deliver a concentrated jet onto the bucket surfaces of the Pelton turbine runner. The momentum exchange on the buckets converts hydraulic energy into torque, which is transmitted via a direct coupling to a 6.3 kV synchronous generator. Power regulation is achieved smoothly by adjusting the needle stroke—from small openings during startup to full opening at rated power. The multi-nozzle configuration (3×120°) additionally allows staged flow control by temporarily limiting the operation of one of the nozzles, which improves efficiency under partial loads and enhances regulation stability.

In dynamic states (sudden load rejection, overspeed signal, or rapid power reduction command), jet deflectors located at the nozzle outlets are activated. The deflector immediately diverts the jet away from the runner, reducing the torque on the shaft to nearly zero without abruptly closing the flow. In the next phase, the needle valves are gradually closed until a new operating point or minimum position is reached. If necessary, the sequence is completed by closing the DN400 spherical valve installed directly upstream of the collector. This sequence—deflectors, needles, valve—limits overpressure and minimizes the risk of water hammer in the penstock.

3.1 Pelton turbine runner

The Pelton turbine runner (fig. 8) used in the designed pumped-storage power plant constitutes the main component of the system responsible for converting hydraulic energy into mechanical energy. In the proposed solution, it has the form of a vertical wheel mounted on a shaft, which is directly coupled to a 6.3 kV synchronous generator. The runner diameter and the number of buckets were selected according to the design parameters—head of approximately 900 meters and flow rate of 0.71 m³/s—in compliance with the characteristics of the Pelton turbine type PV3i-790/160. The buckets (fig. 9), which are the actual working elements of the runner, have the shape of deep cups resembling two joined hemispheres separated by a central splitter ridge. The function of the splitter is to divide the concentrated water jet formed by the needle nozzle into two equal parts, ensuring symmetrical force distribution and reducing transverse loads on the shaft. The working surfaces of the buckets are highly rounded and smooth and, in this project, additionally nickel-plated, which reduces friction, turbulence, and flow separation, thereby maximizing the recovery of the jet's kinetic energy.

Fig. 8. Pelton turbine runner [1].

The buckets are made of high-strength, erosion-resistant stainless steel and are mounted on the runner rim in a manner that allows their replacement during operation. The runner operates by successively capturing the energy of the jets as the buckets enter the nozzle action zone. When the high-velocity water jet, resulting from the large head (on the order of several tens of meters per second), strikes the splitter of a bucket, the momentum exchange process begins. The jet is divided into two parts and guided along the concave surfaces of the bucket. As the jet slides along the surface, it changes direction by approximately 160–170°, which is essential for maximum energy recovery—according to the principle of conservation of momentum, the highest efficiency is achieved when the water leaves the bucket at minimal velocity, directed opposite to the inflow direction.

Fig. 9. Pelton runner bucket [1].

At the end of the contact phase, the kinetic energy of the water is almost entirely converted into torque on the shaft, and the jet exits the bucket without interacting with the subsequent runner elements. The appropriate number and arrangement of buckets ensure continuous torque transmission and smooth turbine operation, reducing vibrations and power fluctuations.

In practice, various structural designs of Pelton runners are distinguished. The buckets may be cast integrally with the runner rim or manufactured as separate, bolted elements, allowing their replacement in case of wear. Stainless steels and special alloys are used, and to improve erosion resistance, nickel plating is applied to the surfaces in contact with water. Runners can be mounted on a vertical axis—as in our design, which facilitates installation in the shaft chamber—or on a horizontal axis, more commonly used in conventional surface systems. Another important criterion is the number of nozzles: typically from one to six, and more rarely up to eight. In our case, a configuration with three nozzles spaced 120° apart has been adopted, allowing flexible power control and high efficiency under partial load conditions.

The Pelton runner is characterized by high hydraulic and mechanical efficiency, reaching 88–91% at the optimal operating point. Combined with a high-efficiency generator, this enables a very favourable overall energy balance of the pumped-storage power plant. Unlike reaction turbines such as the Francis or Kaplan types, the Pelton turbine is not susceptible to cavitation, as it operates under free-jet conditions where the pressure in the jet impact zone does not fall below the vapor pressure of water. Additionally, the bucket geometry and precise jet guidance minimize hydraulic losses, while the materials used enhance the durability of the components most exposed to erosive wear.

The operating parameters of the runner must be selected to satisfy the ratio between the peripheral velocity and the jet velocity, known as the *speed ratio*, which in practice ranges from 0.46 to 0.48. This allows for maximum energy recovery and stable operation at the nominal rotational speed of 1500 rpm. The high precision and strength requirements for the runner result from the fact that it operates under very high dynamic loads generated by three water jets striking the buckets with substantial kinetic energy.

In summary, the Pelton runner in the designed power plant is a vertical wheel equipped with a series of concavely curved, erosion-resistant buckets, whose design and materials have been optimized for operation under extremely high head and moderate flow conditions. The interaction process between the runner and the jet occurs in successive phases—impact on the splitter, jet division, guidance along the concave surfaces, and exit from the bucket—ensuring maximum conversion of kinetic energy into torque. Various structural variants of runners allow the turbine to be adapted to specific operating conditions; however, the fundamental principle of operation remains unchanged, making the Pelton turbine a particularly advantageous solution for high-head pumped-storage power plants such as the present design implemented within a mine shaft.

3.2 Synchronous generator

In the designed pumped-storage power plant, the generator constitutes an integral part of the turbogenerator assembly and is directly coupled with the vertical Pelton turbine. A synchronous generator with a rated voltage of 6.3 kV has been applied, designed to operate at a rotational speed of 1500 rpm, corresponding to the synchronous speed of a four-pole system at a grid frequency of 50 Hz. The generator's rated power has been selected to match the turbine's hydraulic power of approximately 5.3 MW, with an appropriate margin to ensure thermal and dynamic stability.

The unit operates in a vertical configuration, with the rotor mounted on a common shaft with the turbine and supported by journal bearings that transmit axial and radial loads. The rotor is manufactured using a hardened and precisely balanced pole design, enabling stable operation under the high torque generated by the turbine. The stator is wound with high-voltage conductors, impregnated and insulated according to insulation class F/B, ensuring resistance to heating under continuous operation and long service life.

The IC81W cooling system used in the designed synchronous generator belongs to the group of solutions based on a forced-air circulation system, in which the cooling medium removes heat through water-cooled heat exchangers. Air inside the generator is circulated by fans, flows through the stator windings and core components, and is then directed to the water-cooled heat exchangers, where the thermal energy is absorbed. This configuration enables the maintenance of stable temperature conditions within the machine, even under high loads and rapid power fluctuations, which is essential in a pumped-storage power plant characterized by dynamic operating conditions. The closed air circuit reduces the risk of internal contamination by dust or moisture, thereby improving insulation reliability and minimizing maintenance requirements. Compared with conventional air cooling, the IC81W system provides a higher power density for the generator, allowing a reduction in its overall dimensions while maintaining full rated power.

The excitation of the generator is provided by an Automatic Voltage Regulator (AVR) system operating in conjunction with a rectifier and rotor excitation power supply. The regulator maintains a constant voltage level in the 6.3 kV network and enables reactive power control according to the system's operating conditions. The system is equipped with protection against excitation loss and includes a Power System Stabilizer (PSS) for oscillation damping.

The generator specified in the project is designed for parallel operation with the power grid. Synchronization is carried out according to the standard criteria adopted in the design: frequency deviation not exceeding ±0.1 Hz, voltage difference not exceeding ±2%, and phase angle deviation not greater than ±10°. After synchronization, active power control is achieved by regulating the turbine needle nozzles, while reactive power control is performed through the generator excitation system.

To ensure reliable operation, the generator is equipped with a range of electrical protection systems, including stator differential protection, overcurrent protection, undervoltage and overvoltage protection, frequency protection, and protection against asynchronous operation. Mechanical parameters such as shaft vibration, bearing and stator winding temperature, oil level, and lubrication system pressure are also monitored. The measurement and control systems cooperating with the generator are integrated into the plant's central automation system, enabling remote operation and rapid response to grid signals.

The generator and turbine are mounted on a common concrete structure within the shaft chamber, comprising a foundation block and an anchoring plate. The foundation system transfers dynamic forces and torque to the shaft lining, while the use of expansion joints and non-shrink grouting minimizes the impact of vibrations on the surrounding structure. As a result, the turbogenerator unit exhibits high mechanical stability and low levels of vibration and noise emissions.

The synchronous generator coupled with the Pelton turbine in this project plays a key role in converting hydraulic energy into high-quality electrical energy. Its design and construction ensure high efficiency, the ability to operate over a wide load range, and safe integration with the power grid. Furthermore, the implemented protection, cooling, and control systems ensure that the machine meets the operational requirements of pumped-storage power plants, where both availability and rapid response to changing grid demands are essential.

3.3 Annular supply manifold for the turbines

The annular supply manifold of the turbine (fig. 10) is a key component of the hydraulic inlet system in the Pelton turbine. In the designed pumped-storage power plant, it serves as a high-pressure distributor delivering water from the DN400 penstock to three regulating nozzles symmetrically arranged around the turbine runner. Its function is to distribute the water flow evenly under uniform pressure to all nozzles, regardless of their position or instantaneous opening degree. This ensures balanced loading of the runner, stable turbine operation, and minimization of dynamic vibrations of the entire assembly.

The purpose of the annular supply manifold is to ensure uniform water delivery to all Pelton turbine nozzles and to minimize pressure losses and hydraulic imbalance within the inlet chamber. Since each of the three nozzles can operate independently, the manifold must maintain a constant supply pressure regardless of the opening degree of the other nozzles. It also serves as a transition element between the main pipeline and the nozzle connection flanges.

The manifold structure is based on a circular (ring-shaped) design made of sections of high-strength pressure steel pipes welded into a closed loop. In the designed turbine, the manifold is made of high-grade structural steel. The inlet of the manifold is connected to the DN400 inlet flange with a spherical shut-off valve, while the outlets are spaced 120° apart around the turbine axis, where they connect to the regulating nozzles. On the outer surface of the manifold, there are control and measurement ports (for pressure, temperature, and venting) as well as service connections.

Fig. 10. Cross-section of the casing with a view of the turbine supply manifold [1].

The interior of the manifold is structurally divided to ensure uniform pressure distribution under variable flow conditions. Due to its large cross-sectional diameter and smooth flow curvature, hydraulic losses within the manifold are minimized. The external surface of the manifold is protected against corrosion with epoxy or polyurethane coatings resistant to moisture and operating pressure. The component is mounted to the turbine casing using a rigid support frame or brackets welded to the foundation structure.

The operating principle of the annular manifold is based on the even distribution of water flow supplied from the inlet valve to all turbine nozzles. When the DN400 spherical valve is opened, water at a pressure of approximately 8.4 MPa enters the interior of the manifold, which serves as a distribution chamber. Owing to the symmetrical geometric arrangement, the pressure in each branch of the manifold remains nearly identical, ensuring uniform supply to all three nozzles. Each nozzle draws from the manifold the amount of water corresponding to its current opening, controlled by the electrohydraulic system. When only one or two nozzles are partially open, the pressure in the manifold does not fluctuate significantly, as the remaining volume is filled with pressurized water, and water hammer effects are mitigated by the chamber geometry and the structural elasticity.

The use of an annular manifold in the Pelton turbine enables efficient and stable supply to all nozzles with minimal pressure loss and uniform load distribution. Its construction provides high rigidity and resistance to operating pressures, which is crucial for operation under high hydraulic head conditions. Therefore, the manifold serves not only as a hydraulic component but also as an essential structural element of the entire turbine assembly, ensuring operational stability, safety, and optimal utilization of the water's potential energy.

3.4 Regulating nozzles

The regulating nozzles (fig. 11) are essential components of Pelton turbines, as they are responsible for precise control of the amount of hydraulic energy delivered to the runner. In the designed power plant, which employs a Pelton turbine of the PV3i-790/160 type, three nozzles with a diameter of 175 mm are provided, each equipped with an integral double-acting servomechanism. Their purpose is to enable smooth regulation of the water flow and to adjust turbine operation to the generator load and the current conditions of the power system. By controlling the nozzle opening degree, it is possible to maintain the desired active power output to the grid, and in combination with the jet deflectors, rapid power reduction can be achieved without immediately closing the needle, thus effectively preventing water hammer in the penstock.

Fig. 11. Needle-type regulating nozzle [1].

The regulating nozzles are essential components of Pelton turbines, as they are responsible for precise control of the amount of hydraulic energy delivered to the runner. In the designed power plant, which employs a Pelton turbine of the PV3i-790/160 type, three nozzles with a diameter of 175 mm are provided, each equipped with an integral double-acting servomechanism. Their function is to enable smooth regulation of the water flow and to adjust turbine operation to the current generator load and power system conditions. By controlling the nozzle opening degree, the desired active power output to the grid can be maintained, and in combination with jet deflectors, rapid power reduction is possible without the immediate closure of the needle, providing effective protection against water hammer in the penstock.

The construction of the nozzle is based on several main assemblies. The outer casing is made of cast steel and includes an inspection hatch, while all surfaces in contact with water are nickel-plated to enhance resistance to corrosion and cavitation erosion. Inside, there is an integrated body and cover made of chrome steel, forming the passage for the water jet. System tightness is ensured by a replaceable sealing ring, also made of chrome steel, which can be replaced independently without disassembling the entire nozzle. The central regulating element is a needle with a chrome-steel tip that moves axially inside the nozzle, varying the flow cross-section of the water jet.

The nozzle assembly is completed with the nozzle head, also cast from steel and nickel-plated on all surfaces in contact with water, which shapes the jet directed onto the turbine runner buckets. The control mechanism is based on an integral double-acting hydraulic servomechanism equipped with seals and guides, allowing rapid and precise adjustment of the needle position. The needle movement is supported by a spring acting in the closing direction, ensuring that in the event of a hydraulic system failure, the nozzle automatically closes, protecting the turbine and the penstock.

The operating principle of the nozzle is based on the axial movement of the needle, which regulates the opening size and thus the amount and velocity of water directed onto the runner. In the closed position, the needle completely blocks the flow, while in the retracted position, it forms a maximum-velocity water jet striking the Pelton runner buckets. This design provides highly repeatable, linear flow characteristics throughout the full needle stroke range, combined with high kinematic rigidity of the actuator. The ability to smoothly position the needle allows the turbine to operate efficiently over a wide load range while maintaining high energy efficiency. The smoothness and geometric precision of the needle tip, along with the nickel-plated surfaces in the critical zone, minimize turbulence and jet dispersion, resulting in near-ideal momentum exchange with the buckets. The integrated actuator, powered by a hydraulic unit, provides high speed and positioning accuracy while enabling smooth damping of needle movements, which is essential for reducing pressure surges in the DN400 vertical penstock.

The integral hydraulic servomechanism responds to signals from the electrohydraulic controller, ensuring dynamic and stable interaction with the power grid. As a result, the nozzles serve not only a regulating function but also a protective one, since the proper sequence of their opening and closing, combined with the use of jet deflectors, minimizes the risk of water hammer, enhancing the safety and durability of the entire installation. In the designed shaft-based power plant, a three-nozzle Pelton turbine with needle regulation and a jet deflector system is applied, supplied by an annular manifold delivering water from the DN400 penstock. The nozzle outlets are spaced at 120°, ensuring axially symmetrical loading of the runner and stable bearing operation.

3.5 Deflectors

The water jet deflectors (fig. 12) are an integral part of the Pelton turbine control system and are used to ensure rapid and safe power reduction without the need for immediate closure of the needle nozzles. In the designed pumped-storage power plant equipped with a Pelton turbine type PV3i-790/160 with three regulating nozzles, the deflectors are designed as a set of hydraulically operated blades that deflect the water jet away from the runner buckets. Each deflector operates in conjunction with its corresponding nozzle, and their movement is controlled by a single common hydraulic servomechanism, allowing simultaneous and synchronized operation of all components.

The water jet deflectors are an integral part of the Pelton turbine control system and are used to provide rapid and safe power reduction without the need for immediate closure of the needle nozzles. In the designed pumped-storage power plant equipped with a Pelton turbine type PV3i-790/160 with three regulating nozzles, the deflectors are designed as a set of hydraulically operated blades that divert the water jet away from the runner buckets. Each deflector operates together with its corresponding nozzle, and their movement is controlled by a single common hydraulic servomechanism, allowing simultaneous and synchronized operation of all components.

Fig. 12. Jet deflector (symbolic picture of 4-nozzle unit) [1].

The primary purpose of using deflectors is to protect the hydraulic system from the effects of sudden changes in generator load and the need for rapid reduction of the water energy supplied to the turbine. In situations where the power demand in the grid drops sharply or the generator is disconnected from the system, the deflectors enable an almost instantaneous diversion of the water jets without causing a sudden increase in pressure within the penstock. As a result, the kinetic energy of the flow is dissipated, and the occurrence of water hammer is effectively eliminated. Once stable operating conditions are restored, the nozzles can be gradually closed, and the deflectors return to their initial position, directing the jets back onto the runner.

The construction of the deflector is based on a cast structure made of chromium-nickel steel, which ensures high mechanical strength, resistance to wear and hydraulic erosion, and dimensional stability. Each deflector consists of a blade with a precisely shaped front surface, pivotally mounted on the nozzle body so that, in its operating position, it does not interfere with the water jet, while in the deflected position, it effectively changes the flow direction. The blade's rotational movement is driven by a hydraulic system, where the working force is transmitted from the actuator (servomechanism) through a lever or drive shaft. In the project, a single servomechanism controlling all deflectors has been provided, which facilitates movement synchronization and simplifies the hydraulic installation. The entire assembly is made of materials resistant to high-pressure water exposure, and the working surfaces are precisely machined to ensure motion repeatability and minimal flow resistance in the idle position.

The operating principle of the deflector is such that, under normal operating conditions, the deflector blade remains in the retracted position, not interfering with the water jet directed from the nozzle to the Pelton runner buckets. When a control signal is received from the electrohydraulic regulation system (e.g., during a sudden generator load rejection or emergency start-up), the hydraulic actuator rotates the blade into the working position, where it deflects the water jet away from the runner. Consequently, the jet's kinetic energy is not transferred to the Pelton buckets, resulting in an immediate reduction of torque and mechanical power on the shaft. Unlike full nozzle closure—which requires time for the needle to move and involves sudden pressure variations in the conduit—the deflector's action is almost instantaneous and does not generate additional hydraulic loads. Once the transient condition subsides, the deflector returns to its idle position, and flow control is once again taken over by the nozzle needle.

The application of deflectors in a Pelton turbine significantly improves the dynamic response of the power regulation system and the overall safety of the power plant. Proper coordination of deflector operation with needle movement reduces speed fluctuations, minimizes stresses in the penstock, and decreases impact forces acting on the turbine's mechanical components. As a result, the durability of the turbine unit increases, and the stability of power generation under varying load conditions is enhanced. Although auxiliary in nature, deflectors are therefore an indispensable part of modern Pelton turbine systems, ensuring safe and flexible operation under high-head conditions, such as those in the designed installation.

3.6 Spherical shut-off valve

The spherical shut-off valve (fig. 13) is an integral part of the Pelton turbine's hydraulic system, serving as the main safety and hydraulic isolation valve between the penstock and the turbine. In the designed pumped-storage power plant, a spherical valve with a nominal diameter of DN400 and a design pressure of PN100 (10 MPa) has been implemented, equipped with a hydraulic actuator and a mechanical counterweight for closing. Its purpose is to provide rapid and reliable water shut-off to the turbine in emergency situations or during maintenance operations, as well as gradual system opening during start-up, ensuring full control of the flow and minimizing the risk of water hammer.

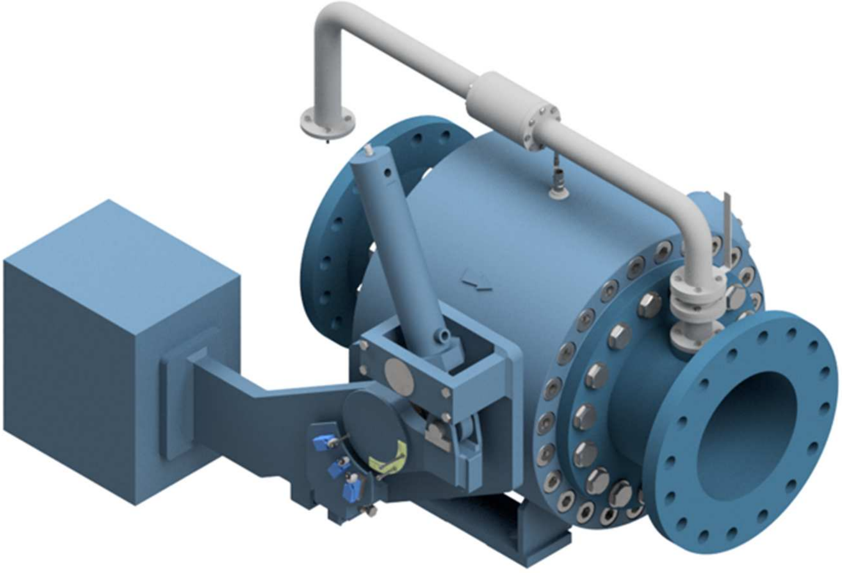

Fig. 13. Spherical shut-off valve [1].

The purpose of using a spherical valve in a Pelton turbine system is to ensure safe isolation of the turbine from the penstock, where a very high static pressure corresponding to a 900 m water column is present. The valve must therefore withstand significant mechanical and hydraulic loads, and its actuator must enable rapid closure in the event of a generator failure or loss of control power. At the same time, the valve serves as a component for controlled start-up—

together with the needle nozzles, it allows gradual filling of the turbine chamber and prevents sudden pressure surges.

The valve structure consists of several main assemblies: a pressure body, a shut-off ball with sealing seats, a rotating shaft with a lever arm, and a hydraulic drive unit. The valve body is made of high-strength carbon steel with connection flanges compliant with PN-EN 1092-1, designed for integration with the DN400 PN100 pipeline. Inside the body, the ball—the shut-off element with an axial through-bore—rotates 90°, enabling full opening or closing of the flow. The ball is made of a material resistant to corrosion and cavitation, and its sealing surface is precisely machined and chrome-plated, ensuring tightness even under very high pressure. The sealing seats are mounted in guide sleeves that compensate for pressure and temperature differences.

The valve is operated by a single-acting hydraulic actuator controlled by pressure from the hydraulic power unit. The actuator is connected to the valve ball shaft via a lever and rotary arm. Under normal operating conditions, the valve is opened hydraulically—the actuator rod rotates the ball into the flow position. In the event of a loss of working pressure, control system failure, or power outage, the valve is automatically closed by the action of a mechanical counterweight, which rotates the ball into the closed position. This design provides a fail-safe mechanism, ensuring the safety of the entire hydraulic installation.

The valve body is also equipped with a DN40 bypass line fitted with its own shut-off valve. The bypass enables slow pressure equalization on both sides of the ball during start-up or after prolonged downtime, preventing sudden water hammer when the main valve is opened. The upper part of the assembly includes venting and filling ports, as well as control valves and connections for pressure testing.

The operating principle of the spherical valve is based on the rotation of a ball with a through-hole inside the body around a transverse axis. To open the valve, the hydraulic actuator, through the lever mechanism, rotates the ball until the through-hole aligns with the axis of the pipeline, allowing free water flow to the turbine. The opening process is carried out in a controlled manner using pressure from the hydraulic power unit. The inclusion of a bypass, which allows gradual filling of the downstream pipeline section and pressure equalization on both sides of the valve, further facilitates opening, as the actuator must only overcome the weight of the counterbalance and the mechanical resistance of the valve. This also helps to avoid sudden pressure changes and dynamic stresses in the pipelines.

During closing, the counterweight acts through the lever to rotate the ball by 90°, positioning its solid surface across the flow and completely cutting off the water supply. The use of a hydraulically operated spherical valve in the Pelton turbine system ensures reliable shut-off of the working medium under all operating conditions. The design is characterized by structural simplicity, high mechanical strength, and safe operation under very high pressures. Combined with the hydraulic power unit and bypass system, the valve ensures smooth flow regulation during turbine start-up and shutdown and serves as protection against uncontrolled water inflow in the event of a control system failure. Consequently, it is one of the key components ensuring the operational safety and hydraulic integrity of the designed power plant.

3.7 Hydraulic power unit

The hydraulic power unit, an integral component of the Pelton turbine control system, plays a key role in regulating water flow by controlling the positions of the needle nozzles, jet deflectors, and the turbine inlet valve. In the designed pumped-storage power plant, an electrohydraulic control system with an independent hydraulic power unit is provided. This unit ensures the required pressure, flow rate, and cleanliness of the hydraulic oil necessary for the reliable operation of all turbine actuators.

The purpose of the hydraulic power unit is to generate and maintain a constant operating pressure within the control system for the turbine's moving components. The system enables precise and dynamic control of the regulating nozzle needles throughout their working stroke, adjustment of deflector positions, and the opening and closing of the DN400 spherical inlet valve. The use of a hydraulic medium with high energy density allows for very fast response times while maintaining high positioning accuracy of the actuating components, which is particularly important in a pumped-storage power plant where frequent and rapid power changes occur.

Fig. 14. Hydraulic power unit [1]

The construction of the hydraulic power unit consists of several main assemblies. The primary component is the oil tank, which serves as a reservoir and provides air and contaminant separation. Connected to the tank are two electrically driven pump units that maintain the operating pressure up to 12 MPa. Each pump is equipped with check valves, safety valves, and oil filtration systems in both suction and pressure circuits. To ensure backup operation during start-up or emergency maintenance procedures, a manual hydraulic pump is also provided.

The pressure circuit of the hydraulic power unit includes a bladder-type hydraulic accumulator filled with nitrogen, which compensates for short-term pressure fluctuations, provides an immediate supply of the working medium in case of sudden demand, and acts as a buffer during pump power interruptions. The unit also includes fast-acting solenoid valves used to shut off or switch circuits depending on the operating mode, as well as proportional valves that control the speed and direction of actuator rod movement. Pressure and return lines are equipped with inline filters, and the entire circuit is complemented by pressure sensors, analogue signal transducers, low-pressure switches, and monitoring manometers. The system also includes an oil cooler, as well as air and water separation systems. The complete assembly is mounted on a rigid steel frame with a drip tray and can be fitted with a soundproof enclosure.

The operating principle of the hydraulic power unit is to maintain a constant oil pressure in the supply lines of the turbine actuator system. When the unit is started, the pumps draw oil from the reservoir and deliver it to the main supply manifold, and then to the actuators of the nozzles, deflectors, and inlet valve. The pressure regulator controls the pumps to maintain the desired working pressure, typically around 12 MPa. Under normal operating conditions, pressure is maintained automatically, while the hydraulic accumulator compensates for short-term drops and stabilizes the overall system operation. When the position of a nozzle needle or deflector changes, the electrohydraulic control system sends a signal to the appropriate proportional valve, directing oil to the relevant actuator chamber. The movement of the actuator rod directly translates into changes in the water flow geometry through the nozzle or the deflection of the jet.

In the event of an electrical power failure or loss of operating pressure, the system is equipped with safety mechanisms—return springs in the actuators automatically move the needles toward the closed position, while the hydraulic accumulator provides the energy necessary for performing essential emergency movements. This design ensures that the hydraulic system remains functional even during transient conditions, maintaining the safety of the turbine's entire hydraulic circuit.

The hydraulic power unit in the designed power plant has been conceived as an independent, prefabricated assembly, complete with piping, fittings, and measuring equipment. All components are made of high-quality materials, and sealing elements are made of elastomers resistant to mineral oils and temperatures up to 80°C.

The use of this type of hydraulic power unit guarantees high reliability and precision in controlling the Pelton turbine system. Due to its fast response times and accurate actuator positioning, it ensures stable generator performance, minimizes the risk of water hammer, and optimizes the overall process of hydraulic-to-electric energy conversion. Combined with a modern electrohydraulic control system, the unit is an indispensable component of the turbine system in high-head power plants such as the one designed in this project.

3.8 Penstock

The penstock has been designed to operate with the Pelton turbine. Its function is to transport water from the surface reservoir to the turbine located in the mine shaft at a depth of approximately 900 m. Due to its complexity, the construction details are shown in drawing no. W52.010PW01.01 [2], which is part of the preliminary design. The penstock is a steel structure of segmental design, facilitating the transport of its components into the shaft. The segments are connected using threaded coupling elements.

Fig. 6 shows a section of the penstock with a description of its main components. The primary load-bearing elements are the support pipes (no. 6), mounted on supporting structures (no. 7) fixed to the shaft walls. These components are evenly distributed along the entire length of the penstock, on average every 80 m. Between the support pipes, the penstock route is formed by pipe segments (no. 8) approximately 6 m in length. The stability of the penstock in the vertical position is ensured by guiding supports (no. 10), which also allow vertical movement of the pipeline. On each section of the penstock between the support pipes, an axial expansion compensator (no. 9) is installed. These elements are essential because, under operating conditions, due to temperature variations, ground vibrations, and other dynamic effects during operation, the pipeline expands or contracts.

At the upper section of the penstock, an inlet pipe (no. 5) is installed, the end of which is located a few meters below the ground surface. It serves to connect the penstock with the outlet installation of the surface reservoir. At the bottom, the penstock terminates in the supply manifold (no. 11), which connects to the spherical shut-off valve (no. 12).

The penstock has a nominal diameter of DN400 throughout its entire length and is made of steel pipes with an outer diameter of Ø406.4 mm. Due to the vertical arrangement and considerable length (approximately 900 m), there is a large pressure gradient along its height. For this reason, the penstock has been divided into three zones with different permissible pressure ratings and corresponding wall thicknesses:

- PN 40 bar, from level 0.0 m to –326.5 m, pipe wall thickness 12.5 mm,
- PN 63 bar, from level –326.5 m to –570.0 m, pipe wall thickness 16 mm,
- PN 100 bar, from level –570.0 m to –900.0 m, pipe wall thickness 20 mm.

Individual segments of the penstock are fabricated as welded structures connected by flanges in accordance with the PN-EN 1092-1 standard. Fig. 15 shows one of the penstock support points on the supporting structure, where the joints between individual components are visible.

Fig. 15. Method of mounting the penstock on one of the supporting structures [2].

3.8.1 Specifications of the penstock components

The inlet pipe (fig. 16) serves to connect the turbine supply penstock with the water discharge system of the upper (surface) reservoir. The pipe has a nominal diameter of DN400. Since it is located at the point in the system where the water pressure is the lowest, the wall thickness is 12.5 mm.

Fig. 16. Inlet pipe [2].

The support pipes (fig. 17) are the components on which nearly the entire penstock structure rests, except for the lowest section, which is supported on the concrete deck of the turbine assembly. Each support pipe is equipped with mounting brackets made of steel plates welded to the pipe surface, allowing it to be seated on the supporting structure. The penstock includes five types of support pipes that differ in their connecting flanges. Three of them correspond to the nominal pressure zones - PN40, PN63, and PN100 - while two are transitional types: PN63/40 and PN100/63.

Fig. 17. Support pipe [2].

The support structure shown in fig. 18 is designed to be anchored in the wall of the mine shaft. It is made of steel profiles as a welded and bolted structure and serves as a support for the support pipe. The structure is available in two types: one with lower and one with higher strength, resulting from the pipeline design divided into three zones with different wall thicknesses and flange sizes. Lighter support structures are used in the upper part of the shaft, while heavier and more durable ones are applied in the lower part.

Fig. 18. Support structure [2].

The main part of the pipeline is built of **pipe segments** (fig. 19). Each segment consists of a steel pipe with an outer diameter of Ø406.4 mm and flanges welded to both ends. Three types of pipe segments are used in the pipeline, corresponding to three pressure zones – PN40, PN63, and PN100. These segments differ in pipe wall thicknesses of 12.5 mm, 16 mm, and 20 mm, respectively, as well as in flange sizes.

Fig. 19. Pipe segment [2].

In addition to the support structures that carry the entire load of the pipeline's weight, the design also includes **guide supports** (fig. 20). Their purpose is to stabilize the pipeline in a vertical position. These supports are arranged along the pipeline route at intervals of approximately 15 m. The structure of the guide supports transfers only radial loads resulting from the pipeline's buckling forces, while allowing free axial movement. Each guide support consists of a bracket anchored in the shaft wall and a clamp securing the pipeline. The bracket is equipped with a guide in which the clamp slider operates. This design allows the clamp to move freely along the pipeline axis while restricting transverse displacements.

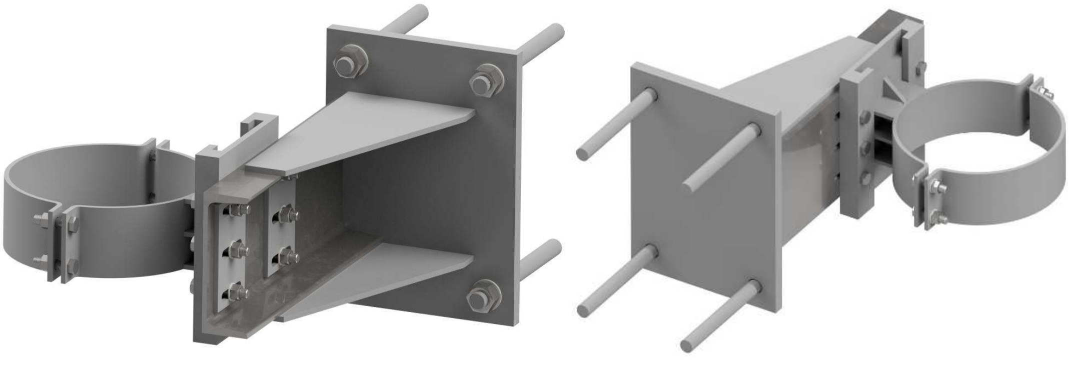

Fig. 20. Guide support [2].

The **gland-type expansion joint** (fig. 21) is used to compensate for axial deformations resulting from pipeline operation and to eliminate the stresses caused by them. The expansion joints are installed in each section between two support pipes, separating adjacent pipeline support points. The length of each pipeline section subject to deformation compensation is approximately 80 m. One connector of the expansion joint is attached to the lower flange of the support pipe (fixed anchor point), while the other is connected to the flange of the pipe segment (movable anchor point). The applied gland-type expansion joint allows axial displacement of the movable connection within a range of ±150 mm. The design includes three versions of expansion joints corresponding to three pressure zones – PN40, PN63, and PN100.

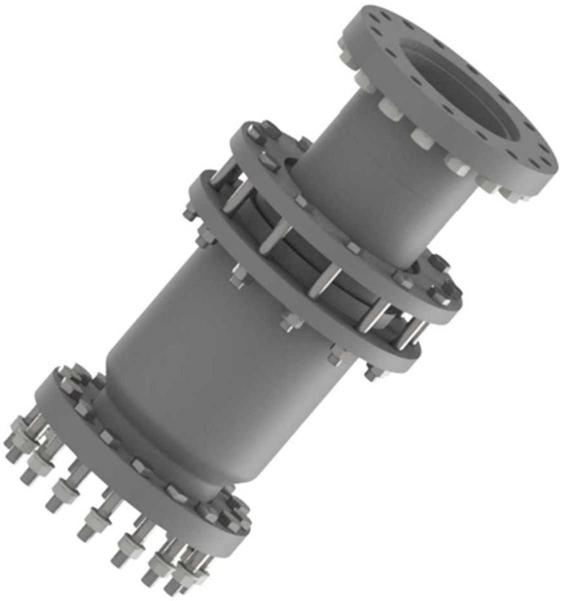

Fig. 21. Gland-type expansion joint [2].

The **supply manifold** (fig. 22) is the lowest section of the pipeline. It connects the vertical part of the pipeline with the horizontally positioned ball valve. As the lowest point of the installation, it belongs to the PN100 pressure zone and is made of a steel pipe with an outer diameter of Ø406.4 mm and a wall thickness of 20 mm. In the manifold section, the water flow direction changes abruptly, which, combined with high flow rate and pressure, imposes significant loads on this component and may cause unfavourable phenomena such as vibrations. The manifold is mounted by embedding it in the concrete load-bearing structure of the turbine assembly.

Fig. 22. Supply manifold [2].

3.8.2 Penstock strength calculation

The downflow pipeline (penstock) supplying the turbine assembly was designed as an installation intended to operate in a mine shaft under mining conditions. All calculations and selection of individual components were carried out in accordance with the mining standard PN-G-05011. Due to the considerable depth reached by the pipeline—and the resulting large pressure difference along its entire route—it was divided into three zones with different allowable pressure values. The following zones were defined:

- PN 40 bar, level 0.0 m to –326.5 m,
- PN 63 bar, level –326.5 m to –570.0 m,
- PN 100 bar, level –570.0 m to –900.0 m.

For each of these zones, strength calculations of the pipe sections were performed separately.

Input data

- DN – nominal diameter of the pipeline; DN = 400 (adopted by the turbine assembly designer),
- D_z – outer diameter of the pipe; for DN = 400, pipes with an outer diameter of D_z = 0.4064 m are used,
- h – height difference between the analysed pipeline section and its highest point; three different height values h were adopted for each pressure zone:
  - h_{40} = 326.5 m,
  - h_{63} = 570 m,
  - h_{100} = 900 m

Calculations

Design pressure p_o:

$$p_o = p_s + p_{ud}$$

Where:
p_s – static pressure;

$$p_s = h \cdot \rho \cdot g \cdot 10^{-6}$$

Where:
ρ – water density; ρ = 1000 kg/m^3,
g – gravitational acceleration; $g = 9.81\ m/s^2$,

$$p_{s40} = h_{40} \cdot \rho \cdot g \cdot 10^{-6} = 326.5 \cdot 1000 \cdot 9.81 \cdot 10^{-6} = 3.203\ MPa$$
$$p_{s63} = h_{63} \cdot \rho \cdot g \cdot 10^{-6} = 570 \cdot 1000 \cdot 9.81 \cdot 10^{-6} = 5.592\ MPa$$
$$p_{s100} = h_{100} \cdot \rho \cdot g \cdot 10^{-6} = 900 \cdot 1000 \cdot 9.81 \cdot 10^{-6} = 8.829\ MPa$$

p_{ud} – surge (water hammer) pressure;

$$p_{ud} = 0.35 \cdot p_s$$
$$p_{ud40} = 0.35 \cdot p_{s40} = 0.35 \cdot 3.203 = 1.121\ MPa$$
$$p_{ud63} = 0.35 \cdot p_{s63} = 0.35 \cdot 5.592 = 1.957\ MPa$$
$$p_{ud100} = 0.35 \cdot p_{s100} = 0.35 \cdot 8.829 = 3.09\ MPa$$

Thus:

$$p_{o40} = p_{s40} + p_{ud40} = 3.203 + 1.121 = 4.324\ MPa$$
$$p_{o63} = p_{s63} + p_{ud63} = 5.592 + 1.957 = 7.549\ MPa$$

$$p_{o100} = p_{s100} + p_{ud100} = 8.829 + 3.09 = 11.919\ MPa$$

k_1 – allowable stress for pipes;

$$k_1 = \frac{R_e}{x_1}$$

Where:
R_e – yield strength; pipe material S355J2, with $R_e = 355\ MPa$
x_1 – safety factor;

- for pipes with a quality certificate $x_1 = 1.8$,
- for pipes without a quality certificate $x_1 = 2$.

The least favourable condition was adopted – $x_1 = 2$.
Thus:

$$k_1 = \frac{355}{2} = 177.5\ MPa$$

z – design strength factor of pipes;

- for seamless hot-rolled steel pipes and helically welded electric-resistance pipes $z = 1$,
- for longitudinally welded electric-resistance steel pipes $z = 0.8$.

Adopted $z = 1$

c_1 – allowance for compensating the negative wall-thickness tolerance; adopted $c_1 = 12.5\%$.

c_2 – allowance for corrosion and erosion by the flowing medium; for an aggressive medium adopted $c_2 = 3\ mm$.

g_o – calculated pipe wall thickness;

$$g_o = \frac{p_o \cdot D_z \cdot 10^3}{2 \cdot k_1 \cdot z + p_o}$$

$$g_{o40} = \frac{p_{o40} \cdot D_z \cdot 10^3}{2 \cdot k_1 \cdot z + p_{o40}} = \frac{4.324 \cdot 0.4064 \cdot 10^3}{2 \cdot 177.5 \cdot 1 + 4.324} = 4.9\ mm$$

$$g_{o63} = \frac{p_{o63} \cdot D_z \cdot 10^3}{2 \cdot k_1 \cdot z + p_{o63}} = \frac{7.549 \cdot 0.4064 \cdot 10^3}{2 \cdot 177.5 \cdot 1 + 7.549} = 8.5\ mm$$

$$g_{o100} = \frac{p_{o100} \cdot D_z \cdot 10^3}{2 \cdot k_1 \cdot z + p_{o100}} = \frac{11.919 \cdot 0.4064 \cdot 10^3}{2 \cdot 177.5 \cdot 1 + 11.919} = 13.2\ mm$$

g_w – required pipe wall thickness;

$$g_w = (g_o + c_2) \cdot \frac{100}{100 - c_1}$$

$$g_{w40} = (g_{o40} + c_2) \cdot \frac{100}{100 - c_1} = (4.9 + 3) \cdot \frac{100}{100 - 12.5} = 9\ mm$$

$$g_{w63} = (g_{o63} + c_2) \cdot \frac{100}{100 - c_1} = (8.5 + 3) \cdot \frac{100}{100 - 12.5} = 13.1\ mm$$

$$g_{w100} = (g_{o100} + c_2) \cdot \frac{100}{100 - c_1} = (13.2 + 3) \cdot \frac{100}{100 - 12.5} = 18.5\ mm$$

The following actual pipe wall thicknesses were selected:

- $g_{40} = 12.5\,mm$,
- $g_{63} = 16\,mm$,
- $g_{100} = 20\,mm$.

3.0 Pressure loss calculation

Definitions and assumptions:

Pressure losses – occur due to wall friction; no local losses
L – Pipeline length = 900 [m]
ρ – Water density at 293 K = 998 [kg/m^3]
$\dot{V}$ – Volumetric flow from documentation = 0.71 [m^3/s]
g – Gravitational acceleration = 9.81 [m/s^2]
D – Internal diameter of the pipeline = 0.368 [m] *
μ – Dynamic viscosity of water at 293 K = 1.005 x 10-3 [Pa·s]
*– External diameter = 0.4 m, wall thickness varies with depth from 12.5 mm to 20 mm. An average value was adopted.

Additional notation:
ε – Absolute roughness [m]
f – Friction factor [–]
PL – Pressure losses [Pa]
Pg – Pressure due to difference in water levels [Pa]
Re – Reynolds number [–]
A – Flow cross-sectional area [m2]
V – Flow velocity [ms-1]
h – Water level [m]

First, the flow regime is defined:

$$Re = \frac{\rho V D}{\mu} = \frac{998 * 6.672 * 0.368}{0.001005} = 2.4 * 10^6$$

Where V is calculated as follows:

$$V = \frac{\dot{V}}{A} = \frac{\dot{V}}{\frac{\pi D^2}{4}} = \frac{0.71}{0.1064} = 6.672\,m/s$$

The flow is turbulent.

The relative roughness was calculated:

The value ε was taken from standard PN-76/M-34034: water pipelines in service; the lower value was selected because the operated pipelines will be new:

$$\frac{\varepsilon}{D} = \frac{0.0012}{0.368} = 0.003261$$

Since the flow is turbulent, the friction factor must be determined from the Colebrook equation:

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\frac{\varepsilon}{D}}{3.7} + \frac{2.51}{Re\sqrt{f}}\right) => f = 0.02684$$

Next, the pressure losses were calculated:

$$\Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = \ 0.02684\frac{900}{0.368}\frac{998 * 6.672^2}{2} = 1.458\ MPa$$

Then, due to the vertical orientation of the pipeline, the pressure change under gravity was calculated from Bernoulli's equation:

$$P_1 + \frac{\rho V_1^2}{2} + \rho g h_1 = P_2 + \frac{\rho V_2^2}{2} + \rho g h_2$$

Since water is incompressible, $V_1 = V_2$

$$P_1 - P_2 = \rho g(h_1 - h_2) => \ \Delta P_g = \rho g L = 8.811\ MPa$$

Therefore, the pressure at the turbine inlet under these assumptions is:

$$P = \ \Delta P_g - \Delta P_L = 7.353\ MPa$$

Temperature sensitivity analysis

Calculations will be carried out for the first case, for the minimum and maximum expected temperatures in the mine shaft. The lowest temperature was estimated at 5°C. Such a value may occur at the bottom of the upper reservoir in winter, when the water surface freezes. The higher temperature, 40°C, was adopted based on the temperature prevailing at a depth of 1000 m in the mine shaft.

Temperature 278 K (5°C)

First, the flow regime is defined:

$$Re = \frac{\rho V D}{\mu} = \frac{1000 * 6.672 * 0.368}{0.001518} = 1.6 * 10^6$$

Where V is calculated as follows:

$$V = \frac{\dot{V}}{A} = \frac{\dot{V}}{\frac{\pi D^2}{4}} = \frac{0.71}{0.1064} = 6.672\ m/s$$

The flow is turbulent.

The relative roughness was calculated:

The value ε was taken from standard PN-76/M-34034: water pipelines in service; the lower value was selected because the operated pipelines will be new:

$$\frac{\varepsilon}{D} = \frac{0.0012}{0.368} = 0.003261$$

Since the flow is turbulent, the friction factor must be determined from the Colebrook equation:

$$\frac{1}{\sqrt{f}} = \ -2\log\left(\frac{\frac{\varepsilon}{D}}{3.7} + \frac{2.51}{Re\sqrt{f}}\right) => f = 0.02686$$

Next, the pressure losses were calculated:

$$\Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = \ 0.02686\frac{900}{0.368}\frac{1000 * 6.672^2}{2} = 1.462\ MPa$$

Then, due to the vertical orientation of the pipeline, the pressure change under gravity was calculated from Bernoulli's equation:

$$P_1 + \frac{\rho V_1^2}{2} + \rho g h_1 = P_2 + \frac{\rho V_2^2}{2} + \rho g h_2$$

Since water is incompressible, $V_1 = V_2$

$$P_1 - P_2 = \rho g(h_1 - h_2) => \Delta P_g = \rho g L = 8.829\, MPa$$

Therefore, the pressure at the turbine inlet under these assumptions is:

$$P = \Delta P_g - \Delta P_L = 7.367\, MPa$$

Temperature 313K (40°C)

First, the flow regime is defined:

$$Re = \frac{\rho V D}{\mu} = \frac{992 * 6.672 * 0.368}{0.000655} = 3.7 * 10^6$$

Where V is calculated as follows:

$$V = \frac{\dot{V}}{A} = \frac{\dot{V}}{\frac{\pi D^2}{4}} = \frac{0,71}{0.1064} = 6.672\, m/s$$

The flow is turbulent.

The relative roughness was calculated:

The value of ε was adopted based on standard PN-76/M-34034: water pipelines in service; the lower value was selected because the pipelines will be new:

$$\frac{\varepsilon}{D} = \frac{0.0012}{0.368} = 0.003261$$

Since the flow is turbulent, the friction factor must be determined from the Colebrook equation:

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\frac{\varepsilon}{D}}{3.7} + \frac{2.51}{Re\sqrt{f}}\right) => f = 0.02682$$

Next, the pressure losses were calculated:

$$\Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.02682\frac{900}{0.368}\frac{992 * 6.672^2}{2} = 1.445\, MPa$$

Then, due to the vertical orientation of the pipeline, the pressure change under gravity was calculated from Bernoulli's equation:

$$P_1 + \frac{\rho V_1^2}{2} + \rho g h_1 = P_2 + \frac{\rho V_2^2}{2} + \rho g h_2$$

Since water is incompressible, $V_1 = V_2$

$$P_1 - P_2 = \rho g(h_1 - h_2) => \Delta P_g = \rho g L = 8.758\, MPa$$

Therefore, the pressure at the turbine inlet under these assumptions is:

$$P = \Delta P_g - \Delta P_L = 7.313\, MPa$$

The change in* water *temperature does not significantly affect the pressure.

A study of the influence of absolute roughness on pressure was also carried out. Its values were selected in accordance with the applicable standard PN-76/M-34034. Values corresponding to the roughness of pipelines in very good condition and with a poor surface with unevenly aligned joints were adopted:

For pipes in very good condition:

First, the flow regime is defined:

$$Re = \frac{\rho V D}{\mu} = \frac{998 * 6.672 * 0.368}{0.001005} = 2.4 * 10^6$$

Where V is calculated as follows:

$$V = \frac{\dot{V}}{A} = \frac{\dot{V}}{\frac{\pi D^2}{4}} = \frac{0.71}{0.1064} = 6.672\, m/s$$

The flow is turbulent.

The relative roughness was calculated:

The value ε was adopted based on standard PN-76/M-34034: **hot-rolled steel pipes, new, unused,** average value:

$$\frac{\varepsilon}{D} = \frac{0.00005}{0.368} = 0.0001359$$

Since the flow is turbulent, the friction factor must be determined from the Colebrook equation:

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\frac{\varepsilon}{D}}{3.7} + \frac{2.51}{Re\sqrt{f}}\right) => f = 0.01328$$

Next, the pressure losses were calculated:

$$\Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = \; 0.01328\frac{900}{0.368}\frac{998 * 6.672^2}{2} = 0.721\, MPa$$

Then, due to the vertical orientation of the pipeline, the pressure change under gravity was calculated from Bernoulli's equation:

$$P_1 + \frac{\rho V_1^2}{2} + \rho g h_1 = P_2 + \frac{\rho V_2^2}{2} + \rho g h_2$$

Since water is incompressible, $V_1 = V_2$

$$P_1 - P_2 = \rho g(h_1 - h_2) => \Delta P_g = \rho g L = 8.811\, MPa$$

Therefore, the pressure at the turbine inlet under these assumptions is:

$$P = \; \Delta P_g - \Delta P_L = 8.090\, MPa$$

For ***pipes in poor condition/with unevenly aligned joints***

First, the flow regime is defined:

$$Re = \frac{\rho V D}{\mu} = \frac{998 * 6.672 * 0.368}{0.001005} = 2.4 * 10^6$$

Where V is calculated as follows:

$$V = \frac{\dot{V}}{A} = \frac{\dot{V}}{\frac{\pi D^2}{4}} = \frac{0.71}{0.1064} = 6.672\, m/s$$

The flow is turbulent.

The relative roughness was calculated:

The value ε was adopted based on standard PN-76/M-34034: **water pipelines with a poor surface and unevenly aligned joints:**

$$\frac{\varepsilon}{D} = \frac{0.005}{0.368} = 0.001359$$

Since the flow is turbulent, the friction factor must be determined from the Colebrook equation:

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\frac{\varepsilon}{D}}{3.7} + \frac{2.51}{Re\sqrt{f}}\right) => f = 0.04219$$

Next, the pressure losses were calculated:

$$\Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = \ 0.04219\frac{900}{0.368}\frac{998 * 6.672^2}{2} = 2.292\, MPa$$

Then, due to the vertical orientation of the pipeline, the pressure change under gravity was calculated from Bernoulli's equation:

$$P_1 + \frac{\rho V_1^2}{2} + \rho g h_1 = P_2 + \frac{\rho V_2^2}{2} + \rho g h_2$$

Since water is incompressible, $V_1 = V_2$

$$P_1 - P_2 = \rho g(h_1 - h_2) => \ \Delta P_g = \rho g L = 8.811\, MPa$$

Therefore, the pressure at the turbine inlet under these assumptions is:

$$P = \ \Delta P_g - \Delta P_L = 6.519\, MPa$$

Changes in pipeline roughness have a very large impact on the final pressure value.

Summary

The performed calculations examined the influence of various parameters on the water pressure at the turbine inlet. The results are presented in the Table 2.

Table 2. The influence of parameters on pressure.

Condition	Pressure change [MPa]
Influence of temperature: 278K -> 313K	0.054
Influence of roughness: 0.05 mm -> 5 mm	1.57

The most significant effect on pressure was caused by the internal surface roughness of the pipeline, resulting in a difference of over 1.5 MPa.

The influence of temperature changes was relatively small compared to other variables.

The fig. 23 shows the relationship between pressure losses and absolute roughness in the range of 0.05–5 mm.

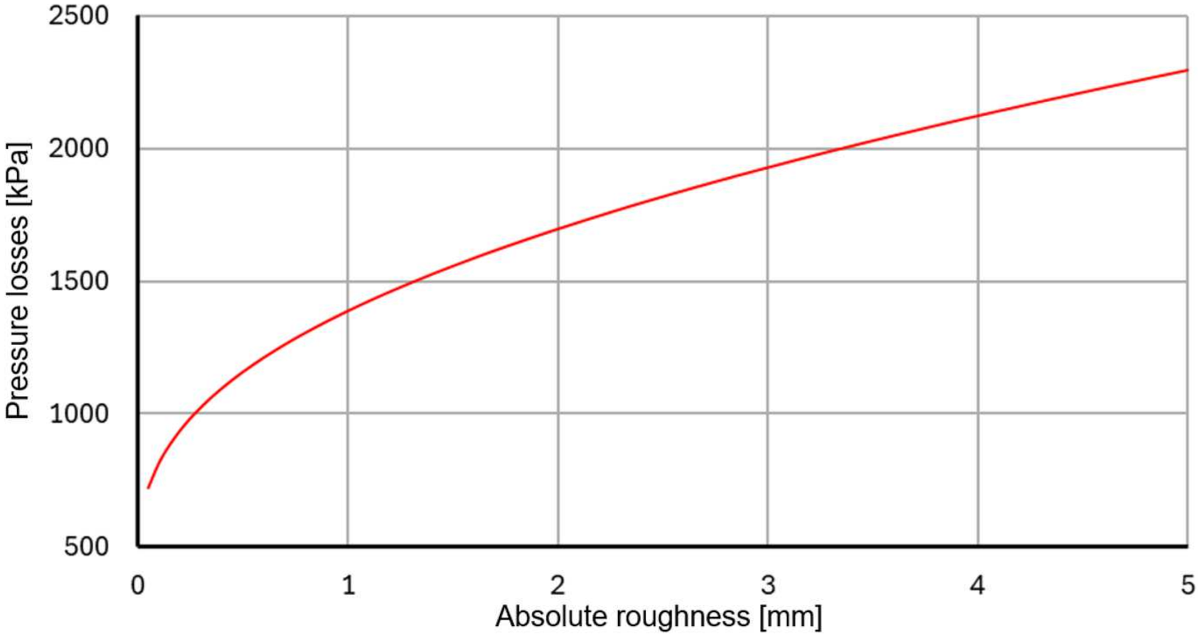

Fig. 23. Relationship between pressure losses and absolute roughness

Pressure losses below 1 MPa were maintained for roughness values below 0.25 mm. Above 3 mm, δ²y/δx² begins to stabilize. At this point, the relationship starts to resemble a linear function, where the rate of pressure drop change is approximately 1 kPa/mm. Reducing pressure losses associated with surface roughness should be considered a priority task.

It should also be emphasized that pressure losses increase proportionally with pipeline length. This phenomenon has a significant impact on system performance, especially at higher values of internal surface roughness. This provides an additional argument for using pipelines with the lowest possible roughness.

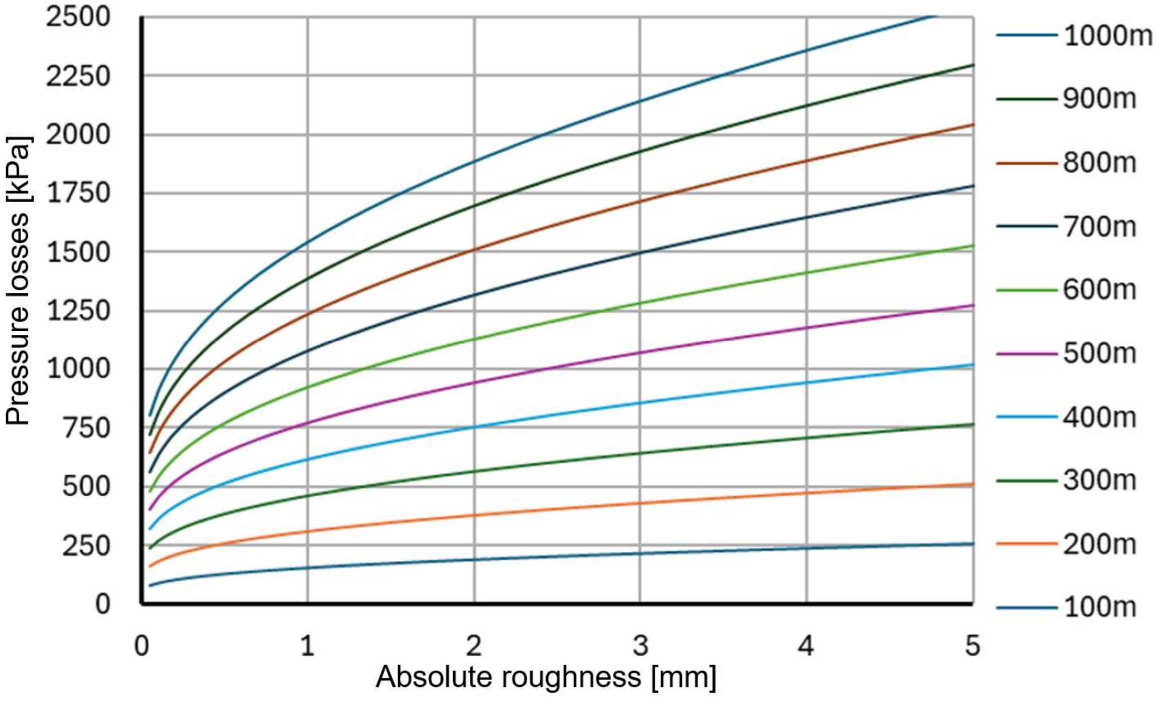

Fig. 24. The pressure losses as a function of roughness values

The fig. 25 shows the pressure losses as a function of temperature for a constant absolute roughness of 1.2 mm. As seen in the graph, the pressure differences in this case are negligible.

The hydrostatic pressure also changes with temperature due to variations in water density. For this reason, the pressure at the turbine inlet ultimately changes more as a result of these density differences than from hydraulic losses. Nevertheless, these variations are small enough to be considered normal fluctuations inherent to turbine operation. Actions aimed at minimizing these losses should therefore be regarded as a low-priority task.

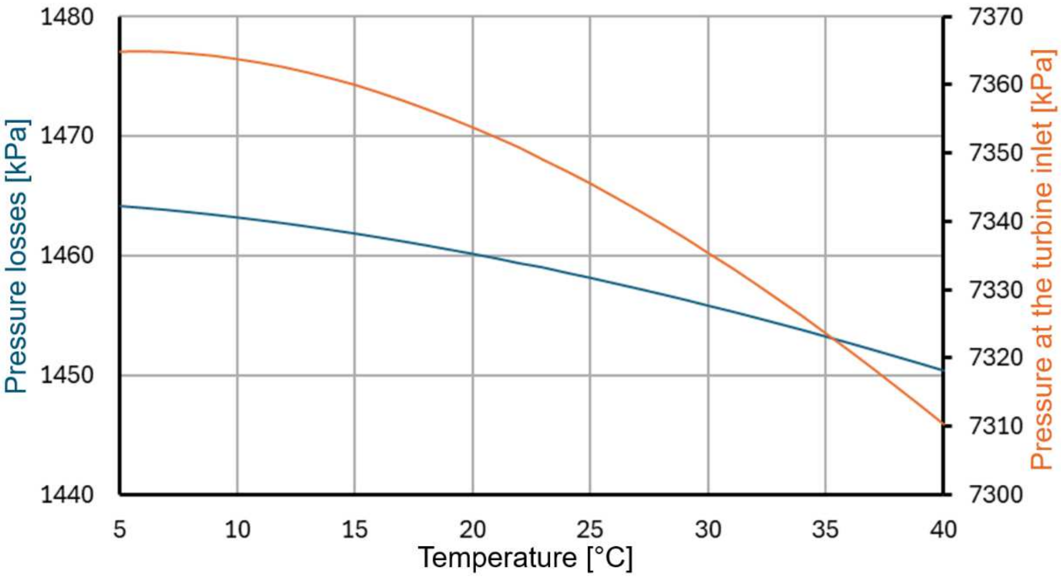

Fig. 25. The pressure losses as a function of temperature

The calculations presented in this study are of an illustrative nature and were performed under the assumption of ideal flow. They do not account for local losses resulting from the presence of valves, bends, welds, or contractions.

In the verification calculations, which analysed pressure losses in the penstock depending on the internal surface roughness, significant differences were observed in pressure drop values for various technical conditions of the pipeline. For low roughness values—corresponding to new, smooth surfaces without signs of corrosion or erosion—the calculated pressure drop remains very close to the value assumed by the turbine manufacturer Voith Hydro (H_net = 837.81 m for H_gross = 900 m). This indicates that the manufacturer performed its calculations assuming a new pipeline with minimal flow resistance, corresponding to a hydraulic roughness in the range of 0.03–0.05 mm (typical for newly rolled or enamelled steel pipes).

Over the course of pipeline operation, however, a gradual increase in internal surface roughness can be expected as a result of natural material aging, hydraulic erosion, electrochemical corrosion, and mineral deposition from flowing water. Even a minor increase in roughness—by a few hundredths of a millimetre—leads to higher friction factors and, consequently, a significant rise in pressure losses along the entire pipeline length. The analyses demonstrated that for higher roughness values, corresponding to operational conditions after an extended service period, the pressure drop in the pipeline becomes noticeably greater than that calculated for the new condition. This means that, under real operating conditions, the pressure at the turbine inlet may be lower than the nominal value, directly affecting the effective net head (H_net) and consequently the turbine's output power.

From a practical standpoint, this finding is highly relevant both for the design phase and for the subsequent operation of the power plant. When selecting and verifying the parameters of the turbine and high-pressure fittings, it is important to consider that actual flow conditions in the pipeline will gradually deviate from the laboratory conditions assumed by the manufacturer. This is particularly critical when the pipeline is made of carbon steel without an internal protective coating-under such circumstances, corrosion and surface microdamage may progress faster, increasing flow resistance.

Therefore, it is recommended that control calculations and hydraulic simulations take into account a range of roughness values representing not only the initial state but also the expected condition after several or even a dozen years of operation. Such an approach enables a more realistic estimation of pressure losses and net head, and allows for appropriate planning of safety margins in the turbine and valve system parameters.

In the case of the designed pumped-storage power plant, where the hydraulic head exceeds 800 m, even a few percent change in friction losses can correspond to tens of meters of water column, translating into power losses of several tens of kilowatts.

The analysis also confirms that the turbine manufacturer's calculations are fully reliable for a new installation, while the additional calculations performed for higher roughness values provide valuable insight into the expected behaviour of the system after long-term operation. These differences do not undermine the design's validity but highlight the necessity of considering aging effects in long-term analyses. The calculations provide important information for future operation and maintenance planning of the pipeline system. Accounting for the aging of hydraulic components and the potential increase in flow resistance is a crucial aspect of ensuring the long-term reliability and stability of the entire pumped-storage power plant.

3.9 Technical service platforms

Fig. 26 shows two levels of technical service platforms installed in the shaft at the –900 m level.

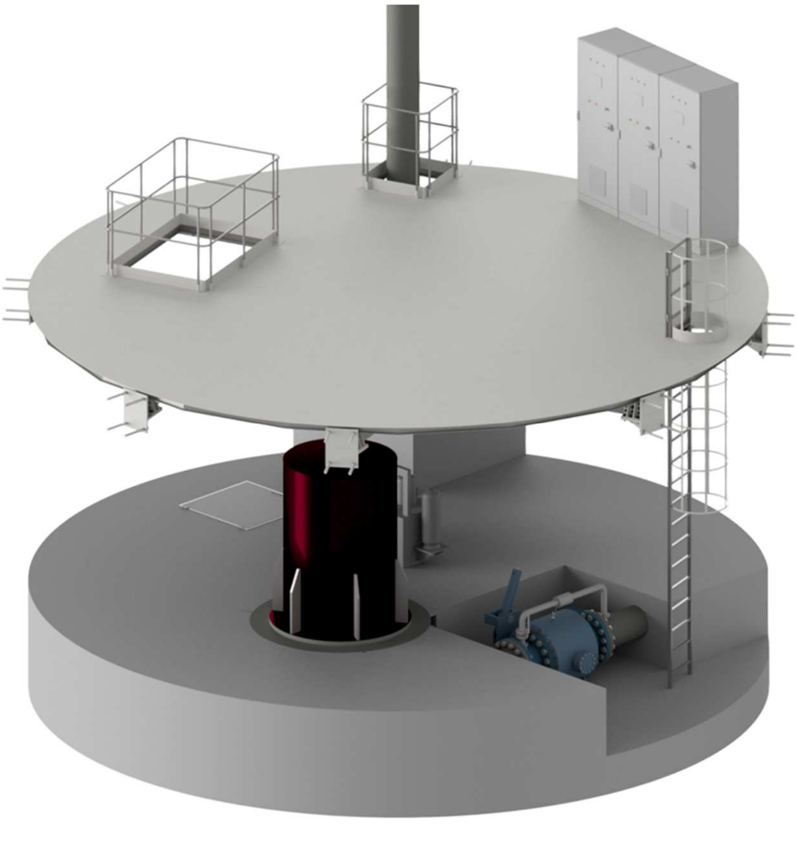

Fig. 26. Lower and upper levels of service platforms installed in the shaft [2].

The purpose of the upper service platform is to allow operating and maintenance personnel to descend to the turbogenerator set and the control equipment of the pumped-storage plant. The platform is a bolted steel construction. Located on the platform are the control cabinets, a service/handling hatch for transferring components to the lower level, and a fixed access ladder leading to the lower service platform. The main load-bearing system consists of girders (fig. 27).

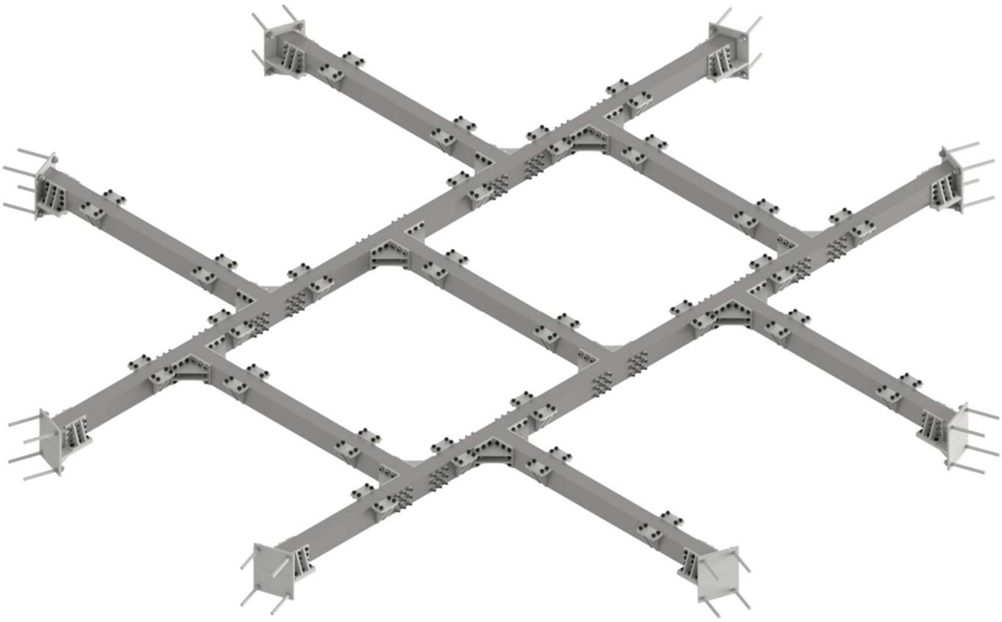

Fig. 27. Primary load-bearing structure of the upper service platform [2].

Each girder is fabricated from two welded C220×80 channels forming a closed box section. Their ends are seated on brackets anchored to the shaft wall with anchor bolts. The girders are bolted together using angle connectors. Additionally, along their length, brackets are provided for fastening the upper platform frame to the primary load-bearing structure (fig. 28).

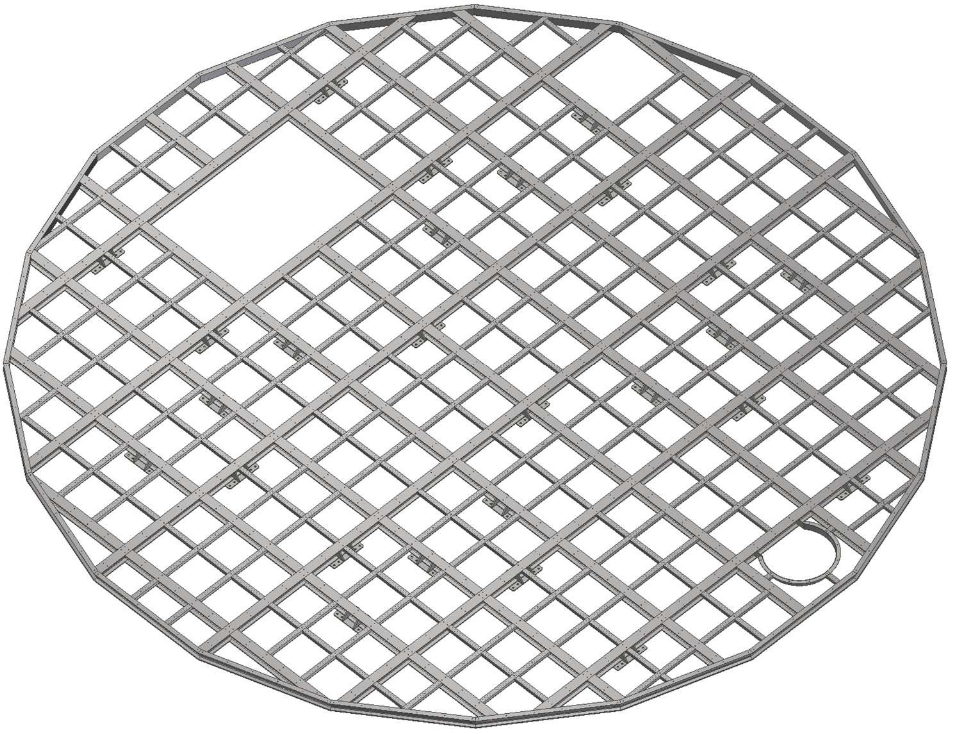

Fig. 28. Upper platform frame [2].

The upper platform frame is executed as a welded structure. The outer ring of the frame is made of U100 channels, while the steel grillage consists of HE100B beams. Additionally, to stiffen the structure, cross-members made of U100 channels are welded between the grillage beams. Checker plates (fig. 29) are bolted to the frame structure to form the platform floor surface. The plates are installed using bolted connections fastened to the steel grillage and the cross-members.

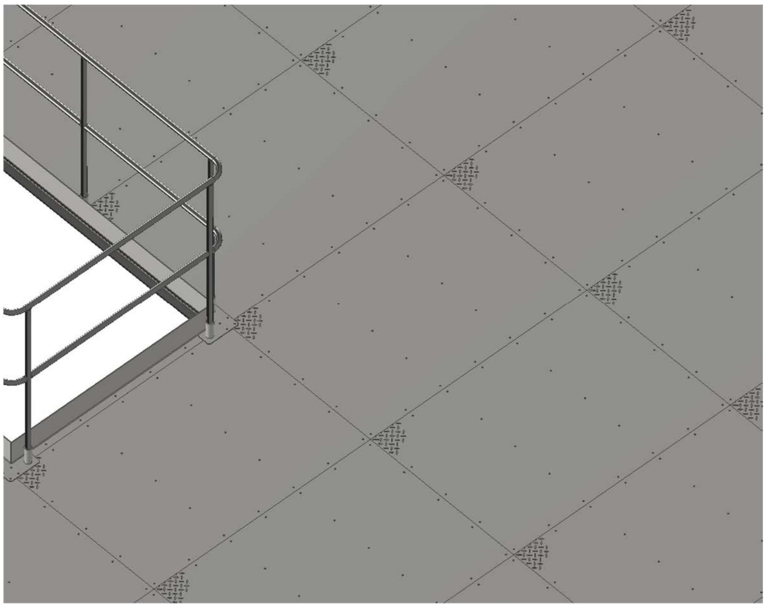

Fig. 29. Platform floor made of bolted tread (checker) plates [2].

Fig. 30 shows the platform in a bottom view following the bolted assembly of the frame with the girders.

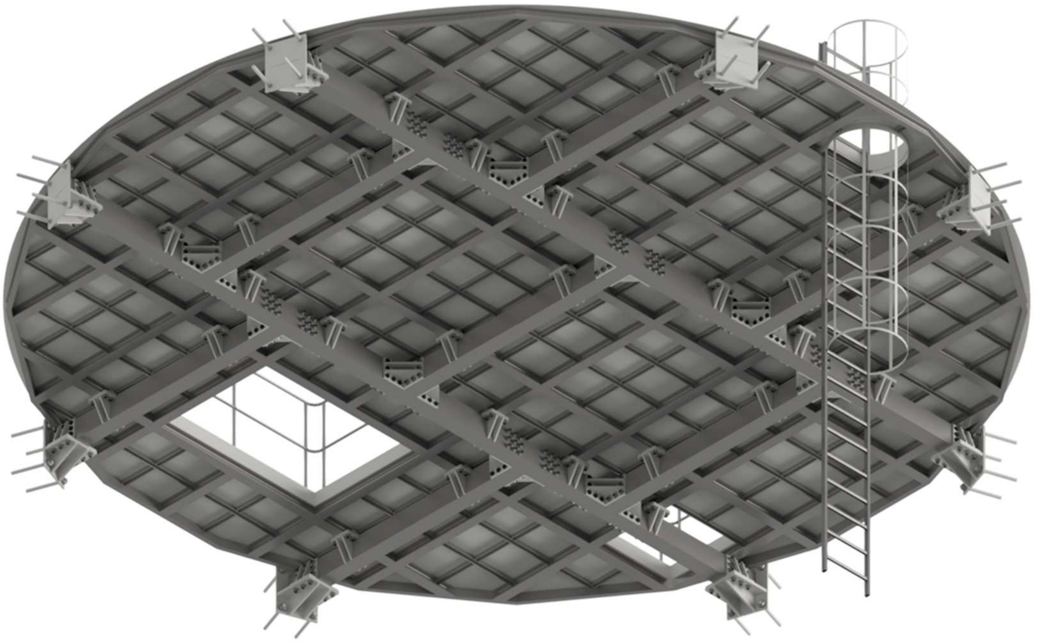

Fig. 30. Bottom view of the platform with the bolted frame and girders [2].

The service/handling opening and the pipe opening on the upper platform are secured with fixed guardrails designed and manufactured in accordance with PN-EN ISO 14122-3 (fig. 31). These safeguards are intended to ensure safe operation of the platform and to eliminate the risk of falls from height during personnel work.

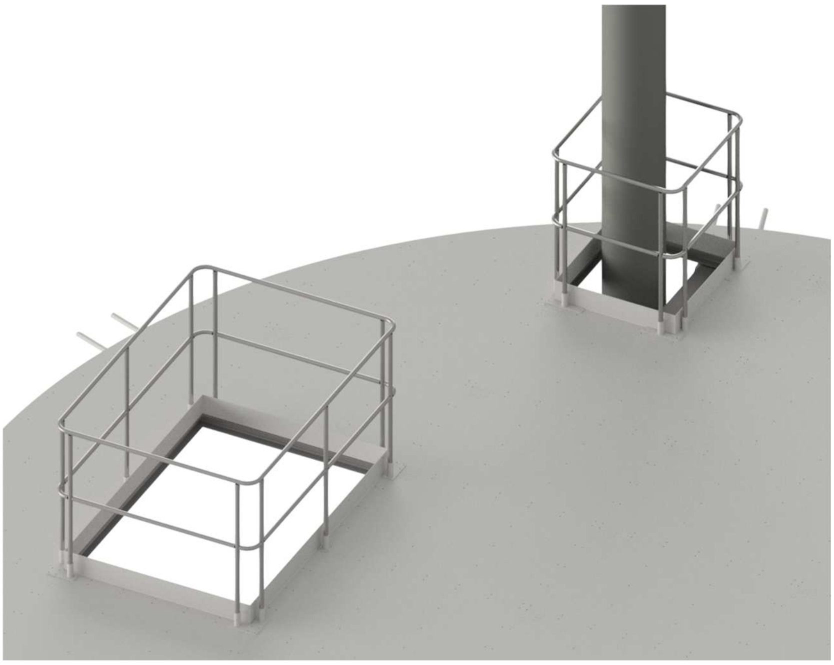

Fig. 31. Technical opening and pipe-penetration opening protected by guardrails [2].

A fixed vertical ladder is installed on the upper platform to provide vertical communication between the upper and lower levels of the technical platforms (fig. 32). The ladder has been designed and manufactured in accordance with PN-EN ISO 14122-4, which specifies requirements for the construction, sizing, and protective features of industrial ladders. The rungs are made of sections with an anti-slip surface, ensuring safe access even under increased humidity.

Fig. 32. Fixed ladder for descent to the lower platform level [2].

The ladder access point is protected by a hinged steel safety gate that prevents accidental entry into the ladder opening (fig. 33). The gate is mounted on hinges, allowing one-way opening and automatic closing under its own weight or by a self-closing spring, in compliance with PN-EN ISO 14122-3.

Fig. 33. Safety gate securing the ladder descent [2].

In the original concept, the lower platform level housing the turbogenerator set was designed as a steel structure based on a system of load-bearing beams and a platform frame (fig. 34).

This solution was intended to provide relative ease of installation as well as the possibility of prefabrication and rapid assembly in the shaft. During subsequent technological consultations and analysis of the turbogenerator's impact on the surrounding structures, it was determined that the material and nature of the structure had to be changed. This is because the turbogenerator generates significant vibrations and dynamic loads during operation. The steel structure does not provide sufficient damping of vibrations, which could lead to elevated vibration levels, noise, and reduced durability. Consequently, the steel solution was replaced with a massive concrete structure (machine slab/foundation), which—thanks to its greater mass and inherent material damping—limits the transmission of vibrations to the shaft lining and installation components (including the DN400 penstock and associated valves). The increased stiffness and the possibility of tuning the natural frequencies minimize resonance risk while improving the geometric stability of the machine seating (maintaining shaft alignment and turbine/generator working clearances). The concrete platform is also less susceptible to corrosion in a humid environment, which lowers maintenance requirements and increases system durability and reliability. The solution facilitates integration of auxiliaries (anchor inserts under foundation feet, cable trays, sleeve penetrations, recesses for the HPU and the DN400 spherical/ball valve) and the use of appropriate grout systems and vibration isolators beneath the machine foundation. In addition, the concrete structure distributes loads more effectively to the shaft lining, allows for robust edges and inspection openings, and is compatible with the required sealing and drainage solutions, collectively resulting in higher safety, better operating culture, and improved service life of the entire turbogenerator node.

Fig. 34. Original design with a steel lower-platform structure [2].

3.10 Shaft reservoir

The discharge reservoir of the planned pumped-storage plant will be formed within an inactive mining shaft and will constitute the primary water reservoir of the pumped-storage system operating cyclically in generation and pumping modes. Functionally, the reservoir serves as a stable, high-pressure "store" of the working fluid (water), enabling safe and repeatable daily cycles as well as rapid power interventions. The reservoir is integrated within the same shaft with the Pelton turbine–synchronous generator set (≈5.3 MW) and with the DN400 penstock divided into PN40/PN63/PN100 pressure zones, which together determine both the hydraulic and structural requirements for the entire system.

The use of an existing deep shaft minimizes mining and civil works while providing a large natural hydrostatic head resulting from a substantial elevation difference. The adopted concept is based on adapting a shaft of approximately 9 m diameter and 1158 m depth (with reinforced-concrete lining) to serve as the reservoir, while maintaining its load-bearing capacity, tightness and stability under variable internal-pressure loads arising from plant cycling. This implies appropriate shaft preparation, including: inspection and possible reinforcement of the lining; selection and application of internal protective/sealing coatings (epoxy or membrane-type, with verified tightness and crack-bridging capability); installation of necessary hydraulic connections (manifolds, valves, fittings); construction of water dams isolating unused headings and workings; and deployment of structural and hydraulic monitoring.

The shaft reservoir will be directly connected to the Pelton turbine hydraulic circuit and to the zoned-pressure DN400 penstock, which imposes requirements for stiffness and tightness at the "shaft–penstock–valving" interfaces and for control of transient phenomena (water hammer) during start-ups and shutdowns. Given the operating character (frequent cycles, large pressure fluctuations), the project provides for lining-deformation monitoring and periodic assessment of coating condition, as well as verification of water parameters (corrosivity, scaling tendency) in the context of material durability.

Below is a concise characterization of the shaft as the lower reservoir, a description of geological–structural conditions, required modifications and preparations for pumped-storage operation, and the interfaces between the reservoir, the Pelton–generator set, and the DN400 penstock. The current design assumptions and material/coating assessment results are also outlined.

3.10.1 Functional scope of the shaft reservoir

The lower shaft reservoir is the terminal hydraulic element in generation mode and the initial (buffer) element in pumping mode. During generation, water descends from the upper reservoir through the DN400 penstock to the Pelton turbine and, after energy release, is received by the shaft reservoir; it must safely accommodate the entire working discharge ($Q \approx 0.71$ m³/s), maintain a stable water level in the inlet/outlet zone, and provide conditions that limit air entrainment and cavitation in the short turbine tailrace. In pumping mode, the same reservoir acts as the water source: the water level and the geometry of the suction intake must guarantee uninterrupted supply to the pumps without the risk of vortices or air draw-in. The reservoir should have a volume margin that allows short-term level fluctuations and balances flow mismatches during start-ups/shut-downs, while damping local unsteady phenomena (hydraulic transients at the turbine outlet and during rapid nozzle/deflector adjustments). Operational requirements include: maintaining the water level within the design envelope for both directions of operation; minimum submergence at inlet/outlet nozzles; limiting velocities in the turbine discharge zone; and controlling fluid quality (suspended solids, aeration, temperature) with respect to erosion, corrosion, and scaling. Because the reservoir is realized within an existing mining shaft, the reinforced-concrete lining and/or protective coatings must be suited to cyclic internal-pressure loading and thermal conditions characteristic of great depth (on the order of several MPa and ~30-40 °C), ensuring tightness (reduced gas/capillary permeability), lining load-bearing capacity, and acceptable displacements. Integration with the turbogenerator assembly requires geometric compatibility of nozzles, isolation valves, and safety systems, as well as provision of inspection access and service drainage. The monitoring system should include continuous level and pressure measurement, temperature, vibration/noise in the discharge area, and leakage/seepage control with potential sediment build-up tracking. In maintenance mode, periodic desilting, coating inspections, and verification of anti-cavitation and anti-vortex devices are envisaged. Overall, the reservoir must guarantee stable hydraulics for the bidirectional operating cycle of the pumped-storage plant-reliable water reception in generation and dependable pump supply in pumping-while meeting durability, tightness, and operational safety requirements in shaft conditions.

3.10.2 Shaft construction and geometry

The considered shaft has a diameter on the order of 9–9.5 m (after reinforced-concrete lining of approx. 0.45 m thickness) and a total depth of ~1158 m. In its deep section, at around –900 m, the turbine-generator set together with the necessary installation and service infrastructure is envisaged. The shaft space below the machine foundation slab will serve as the lower reservoir of the pumped-storage plant.

The foundation slab of the turbine-generator set will be seated on a suitably prepared roof of the inlet machine chamber, bearing on the shaft lining and/or on designated ring beams/stiffening ribs. The top surface of the slab will constitute the technological level for installation and operation of the unit; below the slab, an active space of the lower reservoir will be maintained, with provision for inspection access: manholes in the lining, service walkways and/or ladders, possible short vertical descents, and short connectors to circumferential drifts (where present and integrated into the working volume). At installation penetrations (pipes, sensors, vents), sleeves and through-penetration seals will be designed, with local reinforcement of the rebar and collar rings to preserve lining integrity.

The volumetric parameters of the lower shaft reservoir are preliminarily estimated based on the geometry of the shaft and adjacent workings. Analyses to date indicate an available workings volume of approx. 16 411 m³. Final balances will be adjusted after updating the geometry model (scan survey, descent inventory) and accounting for wall convergence/relaxation and the effect of cyclic water-level changes. The adopted reinforced-concrete lining ($t \approx 0.45$ m) has been analysed under cyclic loading; the numerical models to date showed no cracking of the lining/rock mass within the assumed cycling ranges, maximum displacements on the order of ~10 mm, and circumferential tensile stresses of ~60–65 MPa—forming a starting point for further strength calculations and selection of sealing coatings. Given the integration of the reservoir with the shaft hydrotechnical installation, the design will reserve corridors for the DN400 penstock (vertical routing with guiding supports at ~15 m spacing and support pipes at ~80 m), the venting/desilting system, and measurement points (level, pressure, vibration, seepage).

All geometric and structural solutions will ultimately be aligned with the updated geotechnical/engineering model (in-situ stress state, thermal parameters, influence of formation waters) and with the technological requirements of cyclic operation. In particular:

- the operating water level and active height of the reservoir will be optimised for the required operational capacity and minimum reserves,

- the inspection-access arrangement will ensure safe maintenance and monitoring of the lining,
- space allowances for valves, penetrations and protective devices will be reserved at the detailed-design stage, including the selection of coatings/insulations with low gas permeability and adequate crack-bridging capability on the lining, in line with the previously indicated material findings.

3.10.3 Geological conditions and the in-situ stress state of the rock mass

The rock mass surrounding the shaft is an integral part of the lower reservoir structure and governs both its stiffness and tightness. Based on the analyses to date and project data, the surrounding medium is assumed to be a sedimentary rock mass dominated by clay-sand lithologies (assemblages of sandstones, siltstones and mudstones) with local lenses exhibiting variable fracturing. The mechanical parameters (deformation modulus E, Poisson’s ratio ν, cohesion c and internal friction angle φ) should ultimately be adopted from laboratory testing of specimens and in-situ calibration; at the preliminary stage it is permissible to use interval values and calibrate them in UDEC models based on the deformation response of the lining. A representative in-situ primary stress state was adopted for modelling: σ_z ≈ 25 MPa, σ_x ≈ 25 MPa, σ_y ≈ 12.5 MPa, which reflects the dominance of the vertical and one horizontal component over the other. In such a stress field, with cyclic internal loading of the shaft lining by pressure in the range of 2-8 MPa, numerical calculations showed no initiation of cracking in the reinforced-concrete lining or in the rock mass, and maximum radial displacements on the order of ≈10.1 mm [7]. The recorded range of circumferential (hoop) stresses in the lining was ≈60-65 MPa, providing a safety margin relative to the adopted lining material classes and operational assumptions. In interpreting the results, effective stresses (including the Biot coefficient α) should be considered due to the possibility that pore pressure in conductive zones partially unloads the rock skeleton.

The temperature conditions in the reservoir zone fall within 30-40°C, with a geothermal gradient of ≈0.03°C/m. Temperature variability together with cyclic pressure loading generates a thermal-stress component in the lining; the adopted material characteristics correspond to a unit stress increment of ≈0.375 MPa/°C. This means that for typical ΔT fluctuations (a few kelvins over daily/weekly scales) the additional thermal stresses remain significantly lower than those from internal pressure and do not govern the limit state, although they should be included in combination with hydro-mechanical cycles [7]. Over the long term, the influence of temperature on the rheology of the lining and rock mass (creep, relaxation) should be monitored, and the computational model should be extended with a coupled thermo-mechanical component, with parameter updates based on in-situ measurements (strain and extensometer points in the lining, piezometers in the rock mass). All analyses are conducted for the assumed shaft geometry (Ø ≈ 9-9.5 m) with ~0.45 m reinforced-concrete lining, accounting for excavation convergence and possible reduction of the original working volume. Conclusions from the simulations to date indicate that within the adopted operating pressure ranges there is no risk of lining cracking or opening of fractures in the rock mass, whereas an increased risk of tension may occur at shallow depths (up to ~80 m) at high internal pressures and low horizontal stress components-this requires local verification and, if necessary, strengthening of the shallow zone (monitoring and, where applicable, barrier coatings). Consequently, from the standpoint of the lower-reservoir function, the rock mass and shaft lining-given the adopted parameters-provide the required stability and stiffness and a repeatable cyclic response, provided structural monitoring is implemented and operating parameters remain within the design envelope [7].

3.10.4 Characteristics of concrete coating/barrier systems

In the adapted shaft, it is essential to limit the transport of gases and water through the concrete lining. Sound concrete, even without visible cracking, has a connected pore network capable of flow under pressure, which can generate medium losses during extended standstill periods. For this reason, gas and liquid permeability tests were undertaken for concrete protected with protective coatings.

D.4.3. report developed by SUT [8] focuses on the selection and laboratory verification of surface protection systems ("concrete + coating") with respect to gas permeability at pressure ranges relevant for storage operation (helium measurements, steady-state flow method with reduction to intrinsic permeability using the Klinkenberg correction). Tests were carried out on cylindrical specimens Ø25×30 mm with precise pressure/temperature control and a circumferential sealing sleeve rated up to ~90 bar. To verify procedural repeatability, measurements were performed for two pressure gradients: 60 bar to atm (atmospheric) and 60 bar to 50 bar. The test rigs and procedures provided detectability of very low flow rates and stable conditions for determining intrinsic permeability k in the Darcy regime.

Concrete classes C20/25 and C35/45 were used as substrates, and the reference coatings were: a thin-film epoxy system Sikagard® WallCoat T (DFT ~0.25 mm) and a flexible Xolutec® membrane - Sikagard® M 790 (typical DFT 0.7-0.8 mm, up to ~1.1 mm), both classified under EN 1504-2 as concrete surface protection systems (functions: protection against ingress, moisture control, increase of surface resistivity). The M 790 membrane provides crack-bridging capability and permits application to damp substrates (no condensation), whereas WallCoat T exhibits very low capillary absorption and a high CO_2 barrier (sD); as a thin layer it requires high-quality substrate preparation.

Gas-permeability results confirmed the effectiveness of the barriers: for uncoated C20/25 concrete, mean values of ~4.64×10^{-4} mD (60 bar to atm) and ~4.30×10^{-4} mD (60 to 50 bar) were obtained; for C35/45 the mean was ~1.37×10^{-4} mD (≈×3 reduction vs. C20/25) [8]. Application of Sikagard® M 790 reduced permeability to the 10^{-7} mD range (mean ~5.54×10^{-7} mD for 60 bar to atm and ~4.95×10^{-7} mD for 60 to 50 bar). For Sikagard® WallCoat T, the 10^{-8} mD range was achieved (mean ~3.67×10^{-8} mD). Although scatter is evident for coatings, all values remain 3–4 orders of magnitude lower than uncoated concrete, which unambiguously confirms the sealing effect.

Translation to object scale was illustrated by a "large cylinder" scenario (Ø 8 m, H = 1000 m, He 60 bar, 12 h, isothermal): for C20/25 without coating, the computed pressure drop was ~0.086 bar, whereas for the coated variant (Sikagard® M 790) it was ~0.00015 bar — a ~×600 reduction in losses [8]. This demonstrates that 3–4 orders of magnitude differences at material scale translate into dramatically lower medium losses and longer hold-up times at storage scale.

Water and brine permeability tests of concrete classes C20/25 and C35/45 - both uncoated and coated with Sikagard® WallCoat T and Sikagard® M 790 at a pressure of 6 bar - were carried out in accordance with the PN-EN 12390-8 standard by ITPE [9]. Tests results showed clear differences between the concrete classes: C35/45 exhibited stable and low penetration depths for both media, confirming its dense and homogeneous microstructure, while C20/25 showed higher variability and greater sensitivity to the type of liquid. Concrete samples protected with Sikagard® coatings demonstrated complete resistance to liquid penetration, both after 72 hours (according to PN-EN 12390-8 standard) and after few months of testing. Additionally, the Sikagard® coatings improved mechanical performance by increasing fracture resistance. Both systems showed similarly high barrier effectiveness within the test time frame.

The structural material of the PSH tank is exposed to prolonged contact with water/brine. Therefore, long-term corrosion resistance tests were also conducted on C20/25 and C35/45 concrete, both uncoated and coated with selected Sikagard® systems [9]. Corrosion testing was carried out using 5% chloride, magnesium, sulphate, hydrochloric acid solutions as well as brine solution prepared on the basis of literature data simulating the composition of groundwater over exposure periods of 28, 90, and 180 days. Samples were evaluated for mass loss, dimensional changes, surface condition, and compressive strength.

Across all exposure durations, no mass loss was detected for coated concretes, and their compressive strength remained within the expected range for each concrete class. In most cases, the substrate concrete remained intact, with a smooth, uniform surface. Only Sikagard® WallCoat T showed visible blistering when exposed to 5% HCl, indicating reduced resistance in strongly acidic conditions. Sikagard® M 790 showed no structural degradation, apart from a minor colour change.

Additional corrosion resistance tests under 6-bar CO_2 pressure, simulating CGES conditions, confirmed the high stability of both coatings. Minor surface changes observed for WallCoat T did not affect mechanical strength. Overall, both Sikagard systems provided effective and durable protection in chloride-, sulphate- and magnesium-rich environments, high-salinity mine waters, acidic media, and CO_2-saturated conditions [9]. Sikagard® M 790 demonstrated the highest chemical resistance, while both coatings significantly outperformed uncoated concrete.

The combined results of gas-permeability, water-permeability, and corrosion-resistance testing clearly demonstrate that protective coating systems are essential for ensuring the tightness and long-term durability of concrete structures used in PSH and CGES installations. Both Sikagard® WallCoat T and Sikagard® M 790 meet the performance requirements of EN 1504-2, significantly reducing gas and liquid transport and maintaining structural integrity under chemically aggressive and cyclic operating conditions.

Sikagard® WallCoat T, a thin epoxy coating with very low permeability (10^{-8} mD range), is best suited for stable concrete substrates with minimal deformation. Sikagard® M 790, a flexible membrane capable of bridging cracks and tolerating moisture, offers superior performance in deeper shaft sections subjected to structural movement, dynamic loading, and elevated moisture levels. Together, the coatings form a complementary protection strategy for adapting mine-shaft infrastructure to HESS.

In both cases, EN 1504-2 requirements are met, and the choice should account for the concrete class, structural state of the lining, and cycle conditions. Normative and QA/QC requirements for application include, among others, compliance with EN 206 and EN 12390-2 for specimen/substrate preparation and curing, measurement and interpretation criteria (SSF methodology, Klinkenberg correction), as well as performance requirements for surface protection systems per EN 1504-2. Evaluation of coating performance in the documentation includes, inter alia, EN 1062-3 (capillary absorption), EN 1062-6 (CO_2 barrier, sD), EN ISO 7783 (water-vapour transmission), and EN 1542 (pull-off adhesion).

The operational context from D.3.2. report (cyclic operation 2-8 MPa, medium temperature ~30-40°C, geothermal gradient 0.03°C/m) confirms that coating selection and application quality must be considered together with the mechanical-thermal response of the lining - but as regards gas tightness alone, the D.4.3 results provide directly validated permeability values for designing acceptable standby losses and selecting coating technologies.

3.10.5 Water dams in the context of underground Hydro Pumped Storage systems and the protection of shaft stations

In underground Hydro Pumped Storage (UPSH) installations, where the shaft functions as a vertical water conduit or reservoir, a critical safety requirement is the complete separation of the shaft station and horizontal workings from the water-filled working chamber. For this purpose, water dams are massive concrete structures built across the cross-section of the excavation at the entrances to shaft stations where are used. These dams take the hydrostatic pressure of the water column inside the shaft and provide the level of tightness necessary for the safe operation of the hydraulic system. Their task is not to carry the load of the shaft as a whole but to provide local sealing and isolation of the connection between the shaft and the horizontal excavations.

The dams used in such installations must meet requirements identical to those of classical water-protection dams in mining. They are located in sections of the excavation where the rock mass is as uniform and unfractured as possible, with a sufficiently thick layer of impermeable strata ensuring structural stability. Depending on geological conditions, cylindrical, conical, or multi-stage dams are used, allowing the hydrostatic pressure to be transferred evenly to the sidewalls and roof of the excavation. In rocks with high porosity or low compressive strength, additional grouting and rock-mass reinforcement are applied.

In the UPSH context, the primary role of the dam is to create a permanent and impermeable barrier separating the hydraulic working zone from the access and service areas of the shaft station. In the event of leakage, water could rapidly flood horizontal workings, creating hazards for infrastructure and operational safety. Dams also prevent backflow, reduce hydraulic losses, and stabilize the hydraulic conditions required for the proper operation of pumps and turbines, which rely on constant and predictable water pressures.

The construction of water dams at shaft-station inlets involves careful preparation of the rock surfaces, their cleaning, and, when required, strengthening by grouting. The dam body itself is built from concrete of an appropriate strength class, according to standards for structures in water-hazard conditions. In rocks with low strength parameters, multi-stage dams are used; their increased length reduces the hydraulic gradient and limits filtration. Modern solutions may include injection piles and reinforced anchors to improve the stability of the rock mass surrounding the dam.

From the perspective of UPSH system performance, water dams are a critical component ensuring hydraulic separation. They prevent water under high pressure from migrating into areas not designed to withstand it, stabilize the pressure inside the working chamber, and protect shaft stations from the consequences of hydraulic failure. Their construction, materials, and dimensions must be selected considering the maximum expected water pressure, the strength of the surrounding rock, and the operating parameters of pumps and turbines.

The detailed sizing of the dam is including its minimum thickness, total length, angles of bearing planes, and the relationships between internal forces and rock-mass properties what should be provided in the calculation section that follows.

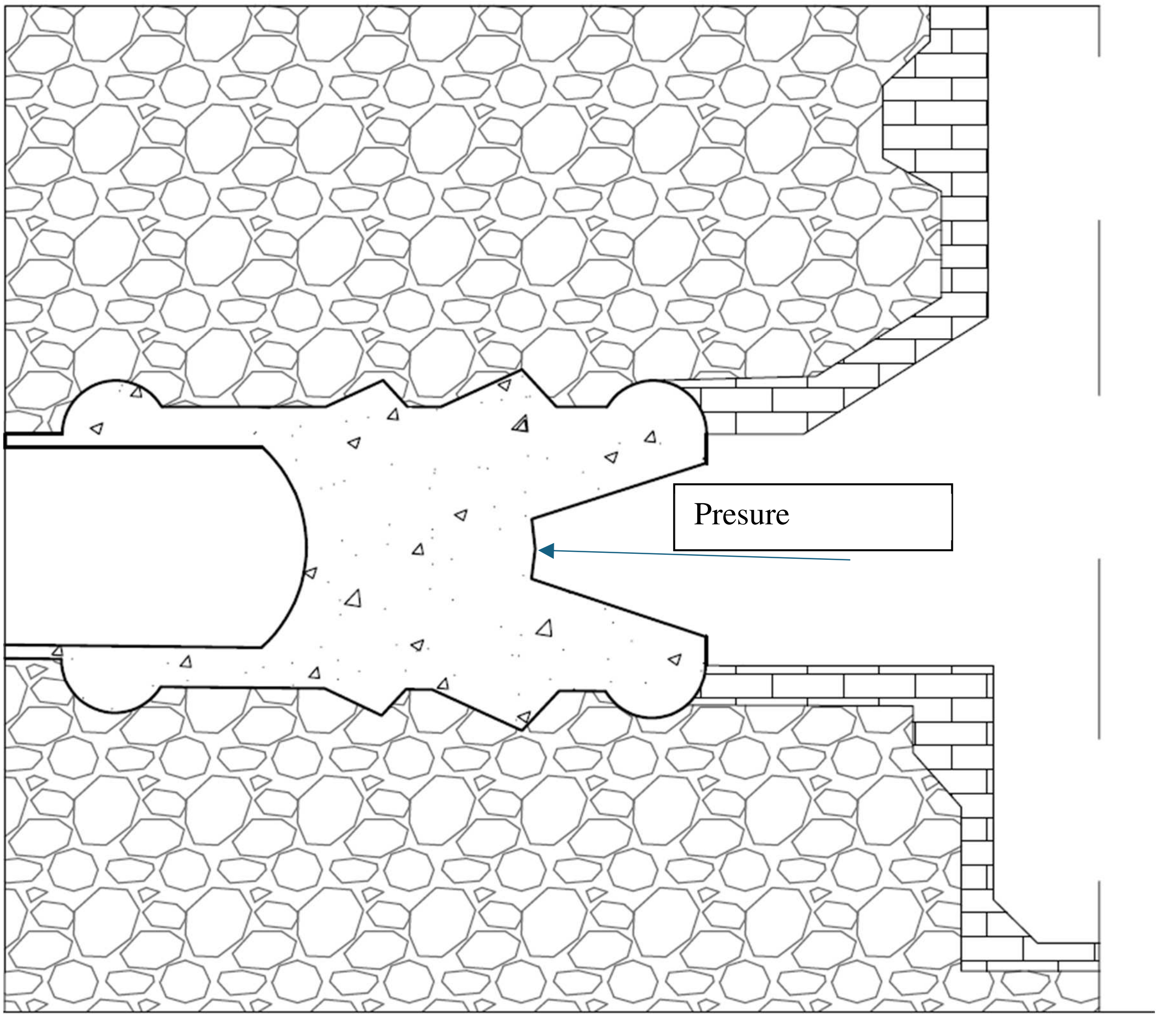

Fig. 35. Water dam

Calculation of the size of a water dam such as in fig. 35 on the example of the shaft discussed in the geometric parameter should be considered for the working pressure of the dam above 0.5 MPa, therefore all calculations of the parameters of this dam should be assumed as for concrete pyramid dams. The calculations are carried out as for two cylinders inscribed in the dam at right angles [10].

The thickness of a concrete pyramid dam should be calculated in centimeters according to the formula: d_t

$$d_t \geq \frac{p_x * r}{2k_c - p_x} + \frac{a^2}{8r}$$

The inner radius r (in the horizontal cross-section) of the dam should be taken for rocks with a strength of:

- to 30 MPa r = 2,0 a,
- above 30 MPa r = 1,5 a.

The inner radius r (in the vertical cross-section) should be taken for rocks with strength:

- to 30 MPa r = 2,0 b,
- above 30 MPa r = 1,5 b.

The angle of inclination of the dam retaining planes to the roadway axis should be assumed for rocks with compressive strength: $\alpha/2$

- to 30 MPa $\alpha/2$ = 15°,
- above 30 MPa $\alpha/2$= 20°.

The thickness of the pyramid dam should be checked at: d_t

- shearing according to the formula

$$d_t \geq \frac{p_x\, ab}{2l\tau}$$

- pressure on the rock according to the pattern

$$d_t \geq \frac{p_x ab}{k_s l\, sin\frac{\alpha}{2}}$$

where:

b – the height of the excavation in the breach, cm;
τ – design Value on wall of dam material by PN-84/B-03264
k_s – allowable compressive stress for rocks, MPa;
l = 2(a+b) – dam circuit, cm.

Depending on the size of the dammed excavation and the exact working pressure of the storage, it is possible to achieve a dam thickness of over 250 cm for one segment. In this case, a multi-resisting dam should be used. Its thickness should be calculated using the formula:

$$d_t \geq \frac{p_x r}{2Nk_s - p_x} + \frac{a^2}{8r}$$

where:

N- the number of dam steps (i.e. the number of retaining planes, or the number of notches around the circumference).

The thickness of a multi-stage pyramid dam should be checked due to:d_t

- shearing of the dam and rock mass structure material according to the formula:

$$d_t \geq \frac{p_x ab}{2Nl\tau}$$

- pressure on the rock according to the formula:

$$d_t \geq \frac{p_x ab}{2Nk_s l - \ sin\frac{\alpha}{2}}$$

The total thickness of a multi-stage retaining dam should be calculated using the formula D_t

$$D_t = \ Nd_t + c(N-1)$$

where:

c - The distance between retaining notches (single dams placed one behind the other) depends on the type of rocks and is:

- 80 cm for rocks with a strength of up to 20 MPa,
- 50 cm for rocks with a strength above 20 MPa.

The above calculation sequence allows us to calculate the number of running meters of concrete needed to stop the shaft bottom. In this part of the shaft analysis, we are able to use basic and publicly available data such as:

- type of housing,
- height of the housing,
- width of the enclosure,
- type of rocks and their parameter,R_c
- pre-approved type of concrete for the construction of the dam,

calculate the volume of the dam, thus estimating the material consumption. Thanks to this knowledge, we can correlate the cost of repairing the shaft casing with the cost of damming the shaft. This procedure will allow you to properly select scales for individual structural aspects.

3.10.6 Preparation and adaptation of the shaft to serve as the reservoir

The primary objective of the preparatory works is to bring the shaft lining and construction interfaces to a condition that ensures long-term tightness and stability under cyclic internal pressure within the operating range adopted in D.3.2 report, while limiting standby losses to levels consistent with the barrier characteristics presented in D.4.3 report. The first stage is a comprehensive condition assessment of the lining: visual inspection and mapping of defects, cracks and leakage points, geometry and deformation measurements referenced to the computational model (D.3.2 indicates no crack initiation under the analysed pressure cycles and maximum displacements on the order of ~10 mm – this state is the baseline for field verification). In parallel, adhesion tests for local repairs and trial barrier applications are performed, and all service penetrations and expansion joints that will require system sealing are inventoried.

Local repairs are carried out prior to barrier application in accordance with the practice described in the standards cited in D.4.3 report. They include reprofiling losses with repair mortars of appropriate class, bonding or sealing of cracks using a system compatible with the subsequent coating, and surface preparation (mechanical cleaning, roughness equalization, dust removal). It is crucial to tailor the technology to the substrate moisture conditions: D.4.3 report indicates that thin-film epoxies (e.g., Sikagard® WallCoat T) provide very low gas permeability but exhibit lower crack-bridging capacity and higher sensitivity to moisture, whereas thick-film membranes (e.g., Sikagard® M 790) offer crack-bridging and applicability on substrates with elevated moisture, at slightly higher – yet still strongly reduced – permeability. On this basis, selection of the barrier system should account for zone-dependent conditions: membranes with crack-bridging capability are preferred in areas with potential micro-movements and at service penetrations, while thin epoxy with top-tier barrier performance is acceptable on large, stable wall areas. Regardless of the system, detail design shall include expansion details and sleeves/gaskets for pipe penetrations, chemically compatible with the coating.

Barrier application is subject to a QA/QC regime that D.4.3 relates to the standards suite: EN 1504-2/-10 (surface protection principles and execution control), EN 206 and EN 12390-2 (references for concrete substrate preparation and curing of test specimens), EN 1062 and EN ISO 7783 (coating parameters, including water-vapour permeability), and EN 1542 (pull-off adhesion testing). Control includes dry-film thickness (DFT) measurements on a point grid, adhesion tests, and inspection of layer continuity/porosity.

The D.4.3 test results are decisive here: coatings reduce the gas permeability of concrete by 3–4 orders of magnitude compared to uncoated substrate (for C20/25, mean values on the order of 4.3–4.6×10^{-4} mD; for membrane Sikagard® M 790 on the order of 5.0–5.5×10^{-7} mD; for epoxy membrane Sikagard® WallCoat T ~3.7×10^{-8} mD), which directly translates into a substantial reduction of standby losses and improved long-term tightness of the system.

Acceptance procedures include two-stage tightness and loading tests. First, sectional/partial tests are performed (e.g., technological chambers, segments with critical details) using a test medium and measurement of through-flow under steady conditions, drawing on the steady-state gas flow methodology and Klinkenberg correction employed in D.4.3 laboratory testing. Subsequently, the entire reservoir is filled and pressure-tested in a stepped sequence up to the operating range resulting from D.3.2, with continuous monitoring of pressure decay, any leaks, and the deformation response of the lining (referenced to the predicted displacements and hoop-stress ranges from the model; D.3.2 reported no crack initiation under cycles and hoop stresses around 60–65 MPa). Positive criteria include: coating continuity and integrity (no material defects), adhesion and thickness compliant with the system requirements, no symptoms of crack initiation in the lining during loading, no increasing trend of medium escape, and stabilization of parameters in the pressure-hold test.

Readiness criteria for filling and cyclic operation follow directly from both documents. The facility shall be deemed ready when:

- repairs and substrate preparation have been accepted, and the barrier meets QA/QC requirements under the standards set indicated in D.4.3;
- sectional tests and the full-reservoir test confirm a reduction in permeation to levels consistent with the applied system class (i.e., orders of magnitude lower than uncoated concrete, in line with the mD values reported in D.4.3);
- the structural response in tests aligns with D.3.2 predictions for 2–8 MPa cycles (no undesired damage mechanisms, displacements and strains within model-permissible limits);
- expansion details and service penetrations are tight and compatible with the coating;
- an operational monitoring plan has been implemented covering pressure/temperature recording, tracking of medium escape trends, and periodic inspection of barrier condition.

Meeting these conditions ensures that the shaft—as the lower reservoir of the scheme—is prepared for safe, repeatable operation in generation/pumping cycles with minimal losses and in accordance with the modelling and material assumptions established in D.3.2 and D.4.3.

3.10.7 Monitoring and diagnostics

Monitoring and diagnostics of the shaft reservoir should form an integrated measurement system comprising the sensor layer, data acquisition, state-assessment algorithms, and response procedures. In a shaft with a diameter of ~9 m and an operating depth up to ~1158 m, multi-level measurement points (e.g., every 100–150 m) are recommended on the shaft lining and in critical zones (e.g., the turbogenerator foundation, DN400 penstock penetrations, expansion joints). The sensor layer should include:

- hydrostatic/operating pressure sensors, including high-pressure piezoresistive transmitters with venting and anti-cavitation protection;
- lining displacement measurements—anchor extensometers, rod or fiber-optic (FBG) extensometers, and crack gauges in the vicinity of expansion joints;
- low-frequency vibration monitoring induced by unit operation (triaxial accelerometers in the 0.5–500 Hz band, optionally acoustic-emission sensors for early detection of micro-crack initiation);
- moisture/leak monitoring—observation points with material moisture sensors, capillary detection strips behind the coating in selected locations, and metered control drains with flowmeters for leakage balancing;
- water and lining temperature along the vertical profile (Pt100/thermistors and fiber-optic DTS for temperature profiling), enabling correlation of thermal effects (30–40 °C, daily–weekly fluctuations) with deformations and coating tightness performance;
- geodetic reference points—permanent benchmarks in the shaft and at the surface, correlated with penstock plumb-line inclinometry, for periodic control surveys.

Diagnostics should rely on trend indicators and tiered alarm thresholds: informational thresholds (e.g., daily leakage growth >×2 versus the 30-day median), warning thresholds (e.g., local increase of sub-coating moisture above the reference limit, adhesion dropping below acceptance), and emergency thresholds (e.g., uncontrolled medium loss, step-change lining displacement, or time-increasing pressure–deformation hysteresis). Results should be consolidated in periodic (monthly/quarterly) reports covering: standby-loss balances, histograms of vibration amplitudes in selected bands, temperature profiles and their correlation with displacements, an incident map, and conclusions on the condition of coatings and lining.

Inspection procedures should provide for visual reviews of coatings and concrete as well as control tightness tests of selected sections.

At acceptance and during operation, readiness criteria must be defined, e.g.: stable pressures with no incremental deformation drift after a full cycle; no active leaks above the design limit; compliance with coating DFT/adhesion requirements; correlation of the temperature profile with the model; and absence of above-norm vibrations induced by the turbogenerator set. In case of exceedances, corrective procedures should be implemented. Optionally, the system may be equipped with AI-based modules using machine-learning methods. In such a configuration, the "learning" function can continuously analyze diagnostics (e.g., seasonal leakage patterns, temperature effects), update alarm thresholds and maintenance schedules, and feed computational models that forecast the durability of the lining and coatings across subsequent operating cycles. In the absence of AI modules, analogous updates can be carried out in expert mode based on monitoring data and periodic condition assessments.

4 Water pumping system from the lower reservoir

The pumped-storage power plant system installed in the shaft of the Budryk coal mine utilizes the mine's main drainage system infrastructure to pump water from the lower reservoir to the upper reservoir. The main drainage pumping station (fig. 36), located at the 700-meter level, is equipped with five stations, each operating OWH-200/10 pumps. Alternatively, OWH-200M/10Ex pumps can be used. The nominal capacity of each pump is 315 m³/h. Two water reservoirs are located adjacent to the pumping station. Taking into account water distribution, the working capacity of the waterways is 1,332 m³ for moderately saline water and 1,236 m³ for brine. Water is discharged from the pumping station to the surface through two Ø350 mm discharge pipelines:

The first pipeline runs to Shaft III, then to the surface through Shaft III to the groundwater reservoir. This pipeline channel transports water with moderate salinity.

The second pipeline runs through the pipeline channel to Shaft I, then to the surface through Shaft I to the groundwater reservoirs. This pipeline channel transports water with a higher salinity.

It is also possible to pump from any water reservoir to any of the pipelines mentioned above using any pump. Additionally, pump chambers are located at levels 1050 m and 1158 m, pumping water to reservoirs at the 700 m level. The design of the lower reservoir of the pumped-storage power plant includes isolating the shaft from the corridor workings with water stops at levels 1158 m and 1050 m. Water pumping is carried out in three stages:

- Stage I – pumping water from level 1158 m to level 1050 m,
- Stage II – pumping water from level 1050 m to level 700 m,
- Stage III – pumping water from level 700 m to the surface.

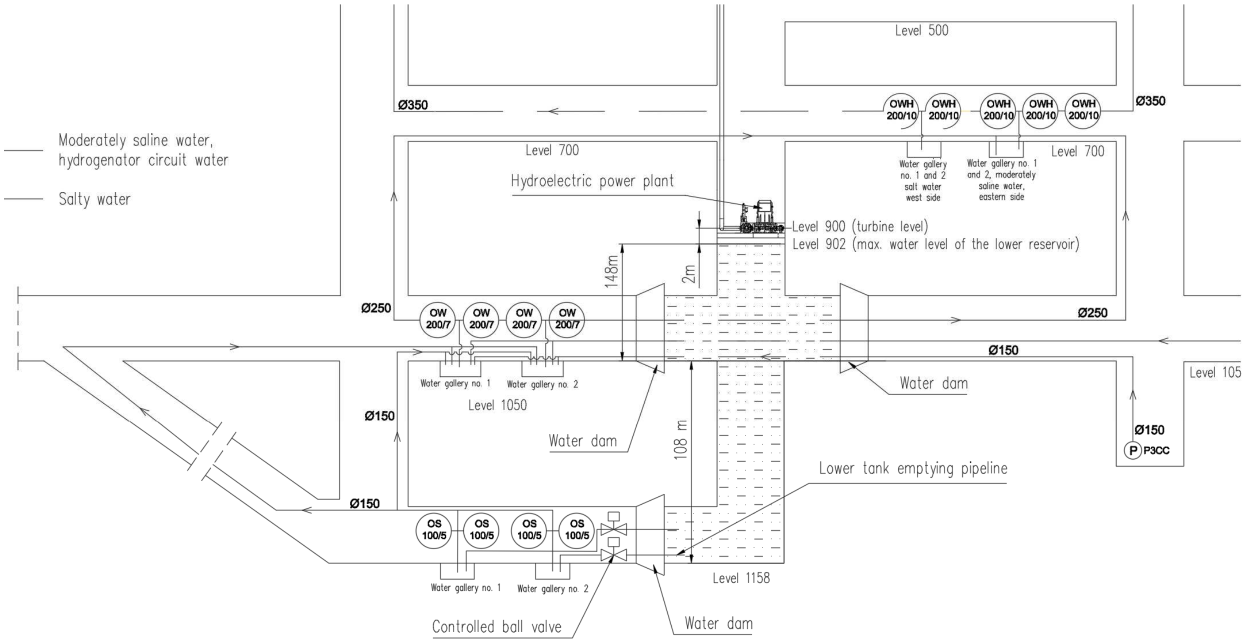

Fig. 36. Schematic diagram of pumping water from the lower reservoir [2]

In stage I, water from the lower reservoir accumulates and flows into two water channels (fig. 37). Water flow is regulated by ball valves mounted in the housings. The pump chamber contains four OS-100/5 pumps. OS pumps are designed for pumping drinking water or slightly contaminated water, equipped with mechanical units with a diameter of up to 2 mm. The pumps consist of five pumping stages, a nominal capacity of 81 m³/h, and a discharge head of 145 m (fig. 38).

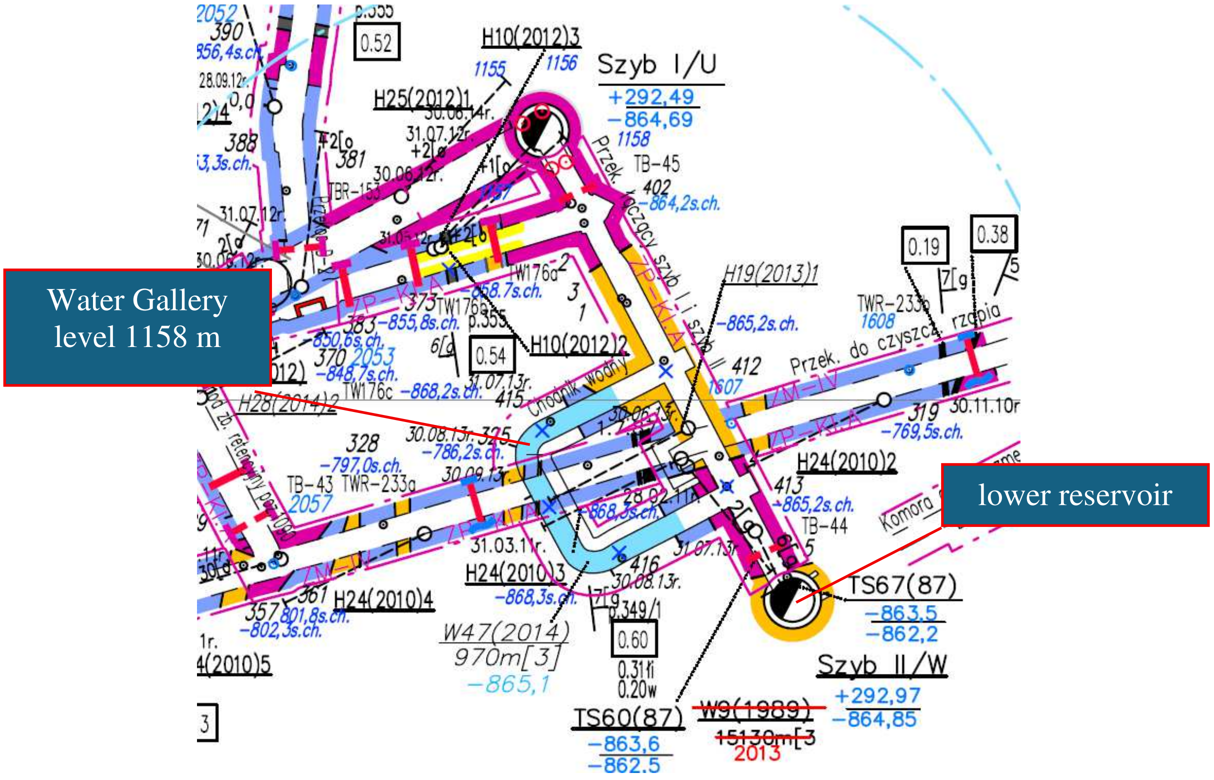

Fig. 37. View of level 1158 m

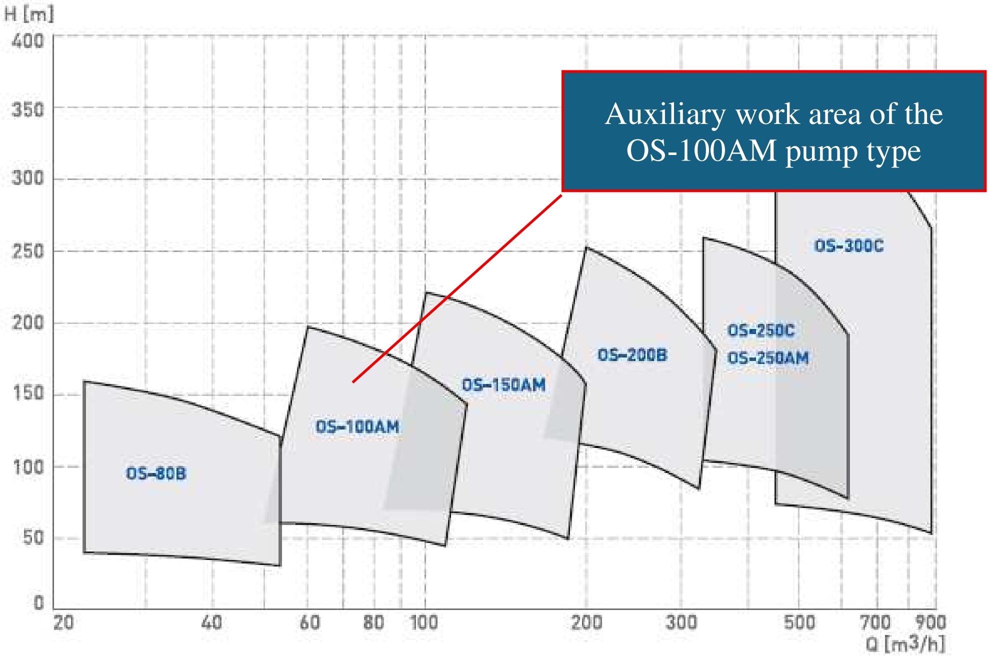

Fig. 38. Operating range of the OS-100AM pump [11]

At the 1,158 m level, the four-pump system has a capacity of 4 × 81 m³/h = 324 m³/h. Assuming pump availability of 50–80%, the effective pumping capacity in Stage 1 is 162–259.2 m³/h. Given the usable water volume of the storage/energy system of 12,000 m³, the resulting pumping time is 72–46.2 hours

In Stage II, water is transported between levels 1050 m and 700 m, i.e., at an altitude of 350 m. Four OW-200/7 pumps are located in the pump chamber at the 1050 m level, pumping water from two reservoirs (water gallery) (fig. 39). OW pumps are rotodynamic, centrifugal, multistage pumps in horizontal configurations, with closed, single-flow impellers and vane guide vanes. The pump impellers and guide vanes are arranged in series within the stage casings. The stage casings, along with the suction and discharge casings, are connected by tie rods. Axial forces are balanced by a disc-type axial balancing system. The pump chamber is used with seven pump stages with a nominal capacity of 300 m³/h and a head of 420 m (fig. 40). The capacity of a four-pump unit at the 1050 m level is 4 x 300 m³/h = 1200 m³/h. This represents a significant surplus of the pump capacity at the 1158 m level. A capacity of 259.2 m³ is assumed for servicing the lower reservoir, which can be achieved using one of the four available pumps. For the operating load of the pumps to the lower reservoir, 259.2 m³/h / 300 m³/h = 86.4%.

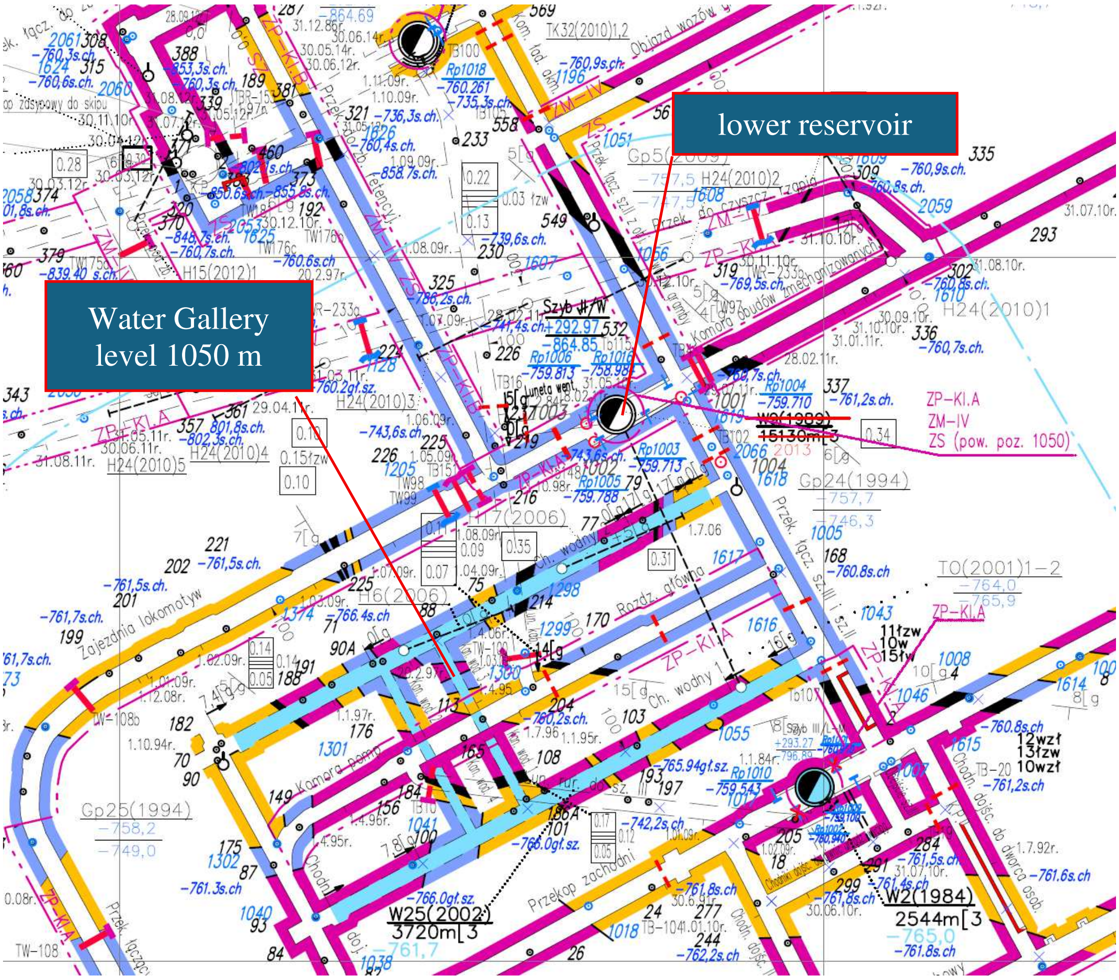

Fig. 39. View of the 1050 m level

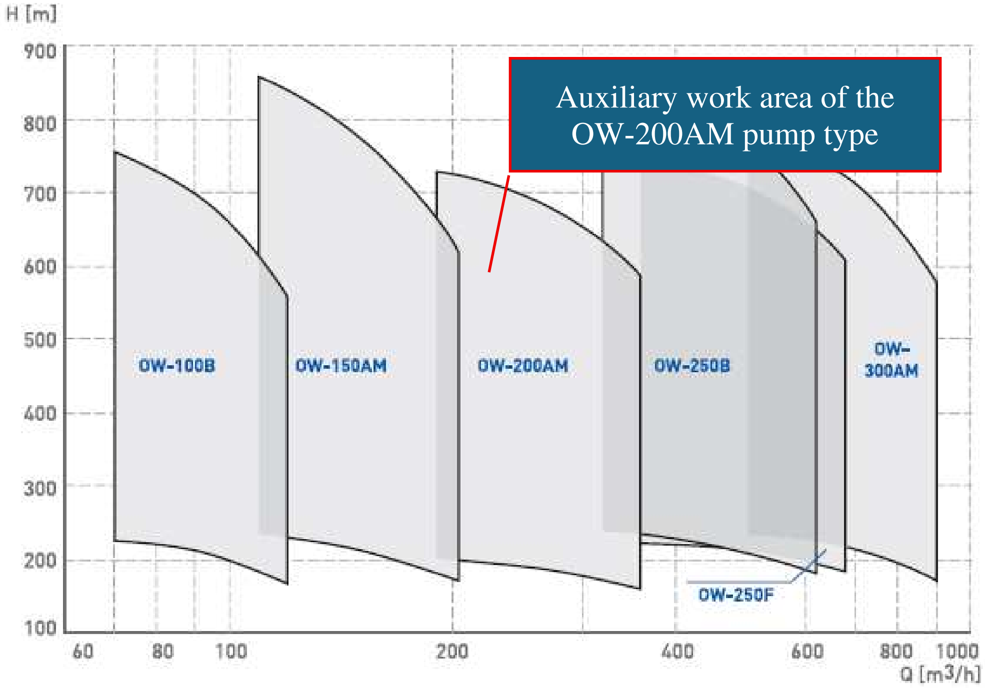

Fig. 40. Operating range of the OW-200AM pump [11]

In Stage III, water is transported from the ground to the surface. In this case, pumps installed in the pump chamber at the 700-meter level will be used. At this level, water flows into two reservoirs, depending on salinity. The water used in the pumped-storage power plant is low-salinity mine water. Low-salinity water is pumped by three OWH-200/10 pumps (fig. 41).
OWH pumps are designed to operate at inflows up to 500 m, allowing them to be connected in series with a backing pump to achieve a higher head. To achieve this, the pump's suction housing and suction port are reinforced. The pumps used in the chamber at 700 m above sea level consist of 10 pump stages, ensuring a high head. The pumps' capacity is 315 m³/h and their head is 800 m (fig. 42).
The capacity of the 4 pumps unit at the 700 m level is 4 x 315 m^3/h = 1260 m^3/h for pumping light saline water. The pump capacity is significantly higher as them at the level 1158 m. As at the 1050 m level, the operation of one pump is sufficient to service the lower reservoir of the pumped-storage power plant.

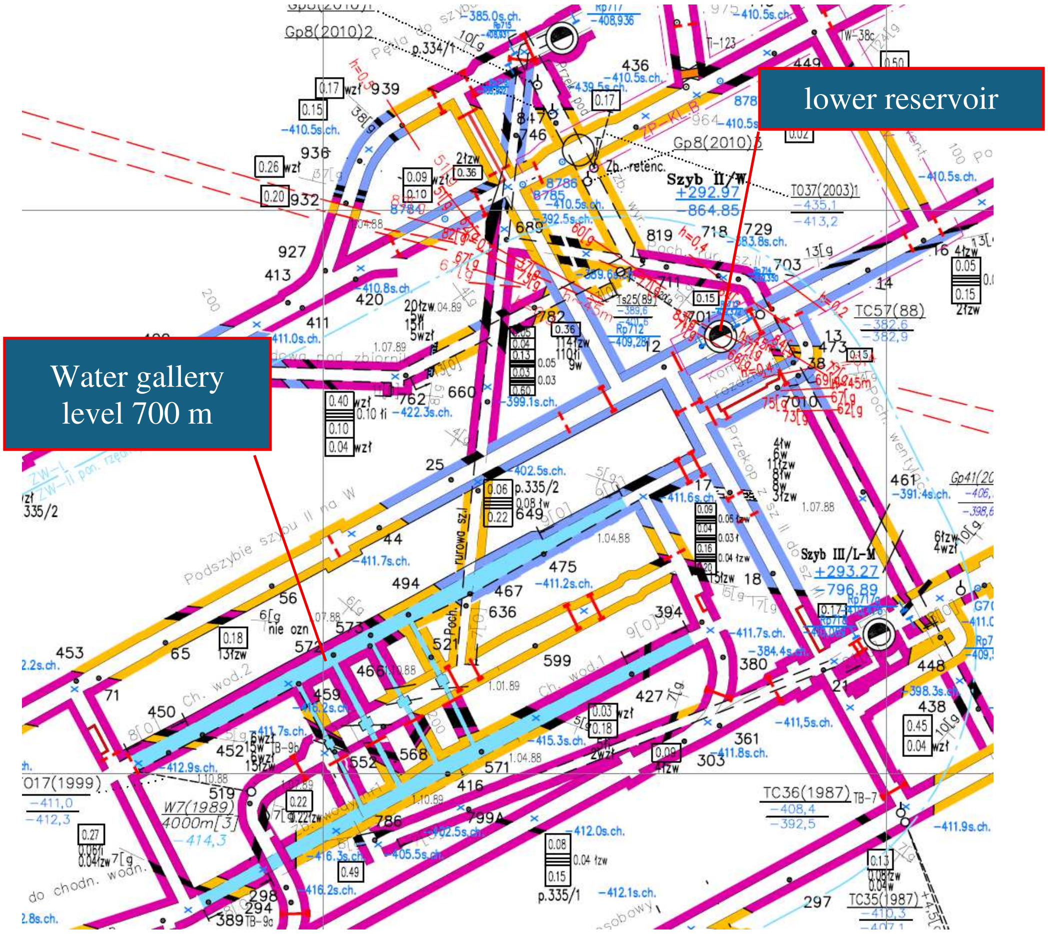

Fig. 41. View of the 700 m level

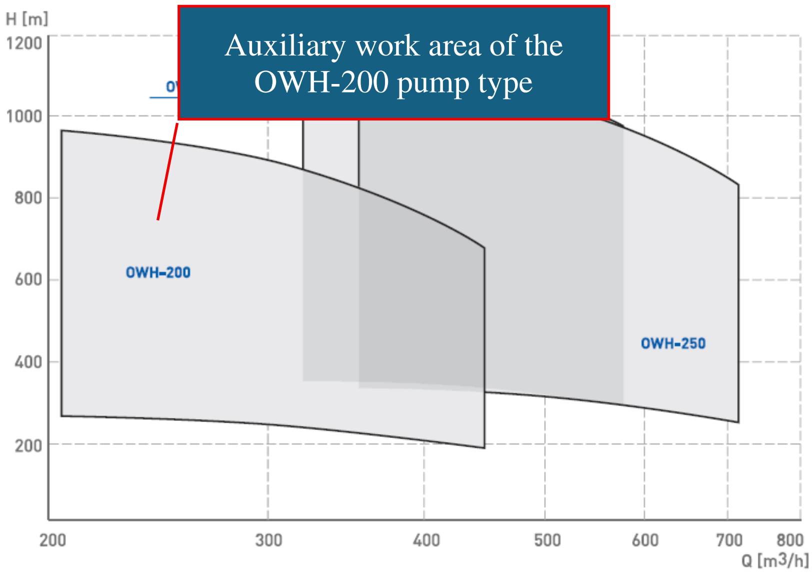

Fig. 42. Operating range of the OWH-200 pump [11]

Table 3 summarizes the results of the pump load analysis and their energy requirements. Calculations indicate that pumping the 12,000 m³ of water stored in the lower reservoir to the surface will consume 64 MWh of energy.

Table 3. Evaluation of the energy efficiency of the lower reservoir pumping process

	Pump type					
	OS-100/5		OW-200/7		OWH-200/10	
Capacity $Q\,[m^3/h]$	81		300		315	
Head $H\,[m]$	145		420		800	
Nominal power $P\,[kW]$	47,5		490		981	
Number of operating pumps N	4		1		1	
$N \cdot P\ [kW]$	190		490		981	
$N \cdot Q\ [m^3/h]$	324		300		315	
Lower tank emptying min/max $[m^3/h]$	Min. 162 Max. 259,2					
Pump load $[\%]$	50%	80%	54,0%	86,4%	51,4%	82,3%
Pumping power $[kW]$	95,0	152,0	264,6	423,4	504,5	807,2
Operating capacity of the lower tank reservoir $[m^3]$	12000,0					
Operating time $[h]$	Min. 74,07 Max. 46,30					
Pumping energy $[kWh]$	7037		19600		37371	
Energy of pumping water from the lower tank $[kWh]$	64008 (64 MWh)					

5 Upper reservoir

The upper reservoir is a key component of a pumped-storage power plant, serving as a storage facility for potential energy. During periods of surplus electrical energy in the system, water is pumped from the lower reservoir into the upper reservoir. During periods of increased power demand, energy is recovered through turbines, into which water flows under gravity.
Main functional features:

- enables short- and medium-term energy storage,
- provides stability to the power system (e.g., frequency regulation, power reserve),
- must be designed for a large number of filling and emptying cycles,
- should minimize water and energy losses (tightness, wind-induced wave reduction, hydraulic efficiency),
- must be resistant to atmospheric conditions and dynamic loads.

The main types of upper reservoirs in pumped-storage power plants include:

- dam reservoirs – created by constructing earth, rockfill, or concrete dams enclosing natural depressions in the terrain,
- artificial reservoirs on flat terrain – built by forming earth embankments around a flat area; offering full flexibility in location,
- concrete or reinforced-concrete reservoirs – used where very high impermeability or limited space is required,
- reservoirs with synthetic membranes – geomembranes used as the primary or additional sealing of the bottom and slopes, reducing infiltration into the ground,
- adapted post-mining pits or quarries – natural basins repurposed as reservoirs.

The upper reservoir of the pumped-storage power plant installed in the mining shaft is located in the western part of the Budryk coal mine area, next to the stockpile area for coal slurry concentrate No. 3 (fig. 43).

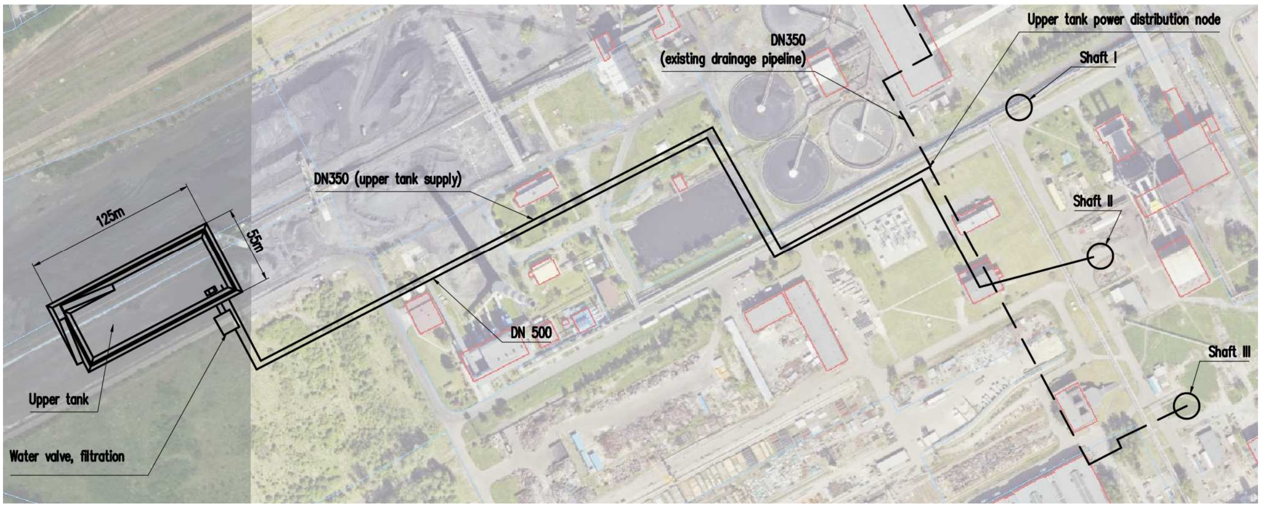

Fig. 43. Location of the upper reservoir [2].

The reservoir has been partially excavated into the ground, to a maximum depth of 1.1 m. The bottom of the reservoir slopes eastwards, towards the water intake. In the sections of the reservoir situated above ground level, the reservoir is surrounded by an embankment, with the crest located at an elevation of 2.9 m. The embankment slopes and the bottom of the reservoir consist of three layers (fig. 44):

1. Geotextile – laid directly on the excavation surface; separates the native soil from the compacted stone layer,

2. Compacted stone material (approximately 30 cm thick),
3. Concreto topping layer (approximately 30 cm thick).

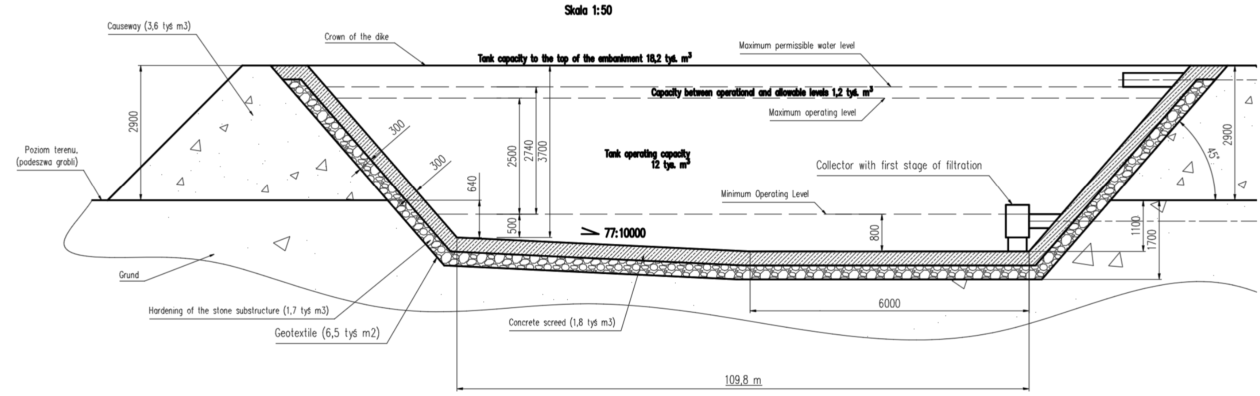

Fig. 44. Cross-section of the upper reservoir [2].

Service ramps are located in the north-western section of the reservoir—both on the outer and inner sides of the embankment—and allow maintenance personnel access to the reservoir basin. The slope of the ramps must not exceed 10°.

The water intake and reservoir supply point are located in the south-eastern part of the reservoir. The intake also serves as a filter for the water supplying the turbine. Its structure is made of a grate and a filtration mesh with an appropriately selected mesh size. This design prevents solid particles from entering the supply pipelines.

Water from the intake is conveyed through two DN350 pipes to the technical building located next to the reservoir. In both pipelines, the optional installation of basket strainers is foreseen as a second filtration stage. Gate valves are installed before and after the filters to enable isolation of the flow and safe maintenance work. Downstream of the filters, a manifold connects the two DN350 pipelines into a DN500 main pipeline, which transports water to Shaft II, where it is reduced to DN400 and directed to the hydrogenerator.

The reservoir is supplied with water from the lower reservoir located in Shaft II, using mine pumps and partly through existing mine drainage pipelines. The water is pumped to the surface through Shaft III, and then, via a distribution node, is directed either to settling tanks or to the upper reservoir of the pumped-storage power plant.

The permissible geometric size of impurities for Pelton turbines, as defined by the turbine supplier (Voith), is calculated as the product of a coefficient of 0.2 and the nozzle outlet diameter. For a nozzle diameter of φ = 61 mm, the maximum allowable mesh opening size must not exceed:

0.2 × 61 = 12 mm.

6 Summary

The project confirms the technical feasibility of using a mining shaft as a lower water reservoir integrated with an electrical power-generation unit. The adopted system architecture turbo-generator supplied by a DN400 downflow (penstock) pipeline, a staged pumping system, and a surface reservoir – forms a coherent, scalable scheme for cyclic operation in generation and pumping modes. Geometric and structural analyses of the shaft, together with the concepts for service platforms, the machine foundation, and the penstock routing, indicate the possibility of safely integrating the equipment while maintaining operational access and separation of functional zones.

In particular, the decision to replace the steel lower platform with a massive concrete structure improves the operating behavior of the machine node: it reduces vibration transmission, stabilizes the foundation, and simplifies maintenance in a humid environment. At the same time, the outlined flow – control solutions (needle nozzles with jet deflectors) and the hydraulic power unit establish a predictable and well-defined dependency chain between hydraulics, automation, and grid interconnection requirements.

From a reliability and safety perspective, proper preparation of the shaft lining for cyclic operation under variable internal pressure is critical. The current results of materials and coating studies suggest that appropriately selected barrier systems on concrete significantly reduce permeability – and thus standby losses and degradation risk – while requiring strict application quality control. In geomechanics, the adopted in-situ stress state and thermal effects do not indicate fundamental obstacles, but they underscore the need for further calibration of numerical models and deformation monitoring of the lining during operation.

Operationally, the project confirms that staged pumping from intermediate levels and the adopted working capacity of the system enable planning of duty cycles consistent with the energy-storage objective. In the case of implementing the solution, it will, however, be necessary to complete a full transient analysis (water hammer), validate the start-up and shutdown sequences to limit dynamic loads in the penstock, and agree on the synchronization criteria with the grid operator. In parallel, occupational safety and access issues (platforms, guardrails, ladders, safety gates, evacuation routes) should be refined, and final maintenance procedures prepared for zones with elevated humidity.

Overall, the picture emerging from the study is positive: if the specified design and execution rigors are observed, the solution can deliver flexible peak power, shorten system response times, and enhance the balancing capability for renewables. The principal added value lies in the reuse of post-mining infrastructure – this reduces both costs and environmental impact while creating a replicable template for similar locations.

7 Literature

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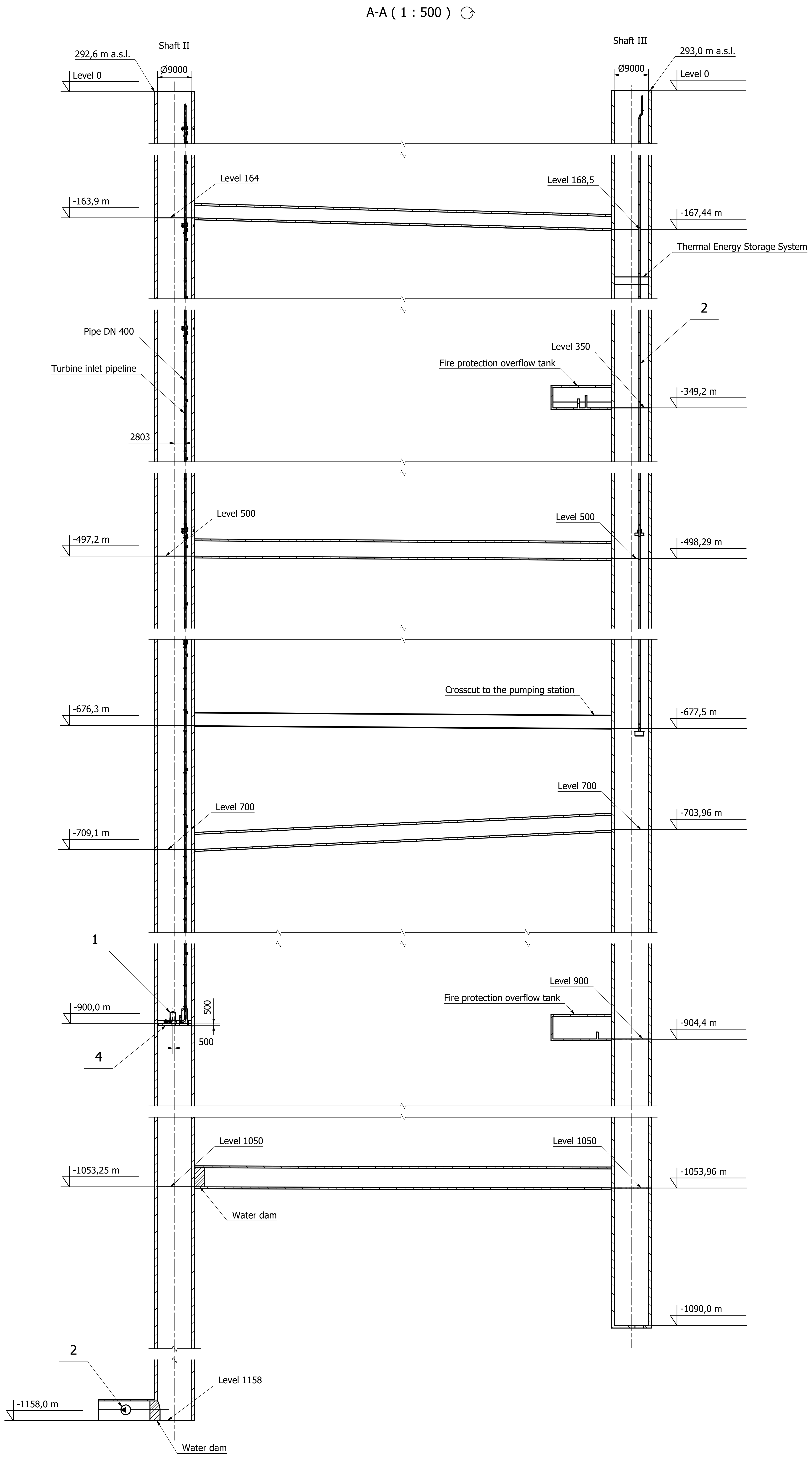
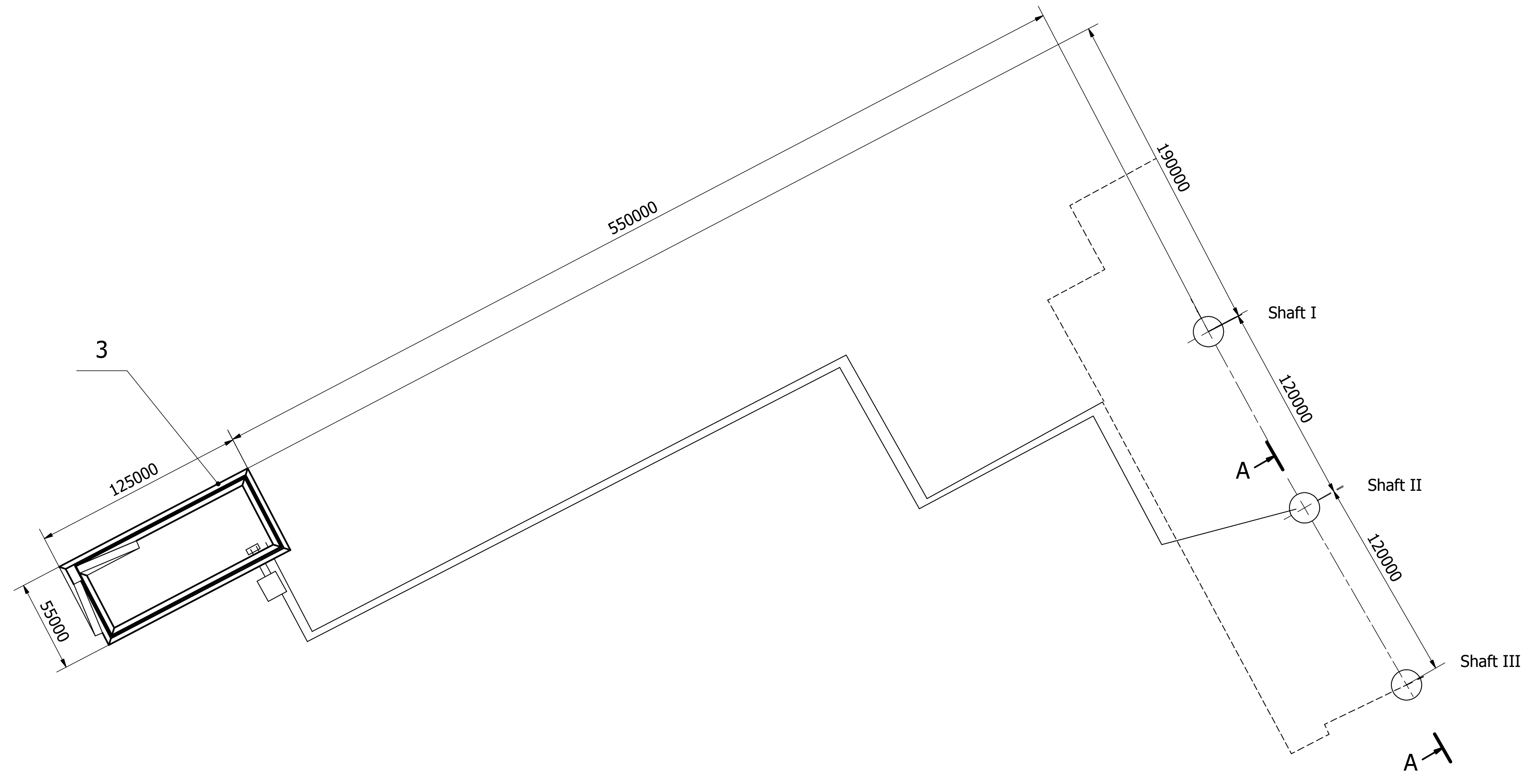
3. Renewables First, https://renewablesfirst.co.uk/
4. VOITH Hydro GmbH & Co KG, https://www.voith.com
5. VOITH Hydro GmbH & Co KG, https://www.voith.com/corp-en/products-services/hydropower-components/turbines.html
6. VOITH- Pelton turbines, https://www.voith.com/corp-en/VH_Pelton-turbines_20_vvk_VH3371_en.pdf
7. Deliverable 3.2. of HESS project, Technical parameters of the example underground water reservoir model, G. Smolnik, K. Kołodziej, A. Niewiadomski, M. Lutyński, 31.12.2024, https://itpe.pl/wp-content/uploads/2025/03/D.3.2_HESS_Technical-parameters-of-the-example-underground-water-reservoir-model.pdf
8. Deliverable 4.3 of HESS project, Lining materials selection and properties, K. Kołodziej, M. Lutyński, 30.09.2025, https://itpe.pl/wp-content/uploads/2025/11/D4.3_HESS_Lining-materials-selection-and-properties.pdf
9. ITPE report on the implementation of the work entitled: Hybrid Energy Storage System using post-mining infrastructure (no 84/2025), 31.12.2025, A. Czardybon, M. Dobras, J. Bigda, K. Kulik, S. Dobras, T. Spietz, K. Supernok, K. Głód, R. Fryza

10. Zasady likwidacji szybów i wyrobisk przyszybowych w kopalniach węgla kamiennego, S. Stałęga, D. Golec, Z. Mrowiec. P. Guzik, Główny Instytut Górnictwa, Katowice 1998.
11. Powen-Wafapomp S.A. Group — Pumps for Mining: Catalogue





INSTITUTE OF MINING TECHNOLOGY
KOMAG

RFCS Project: HESS
No.: 101112380

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	Department	Last name	Signature	Date
Task leader	BG	P. Matusiak		10.2025
Constructor	BG	R. Baron		10.2025
Verifier	BG	K. Kwaśny		10.2025
Approver	BG	P. Matusiak		10.2025

No.: W52.010PW.en

Sheet / number of sheets 1/1

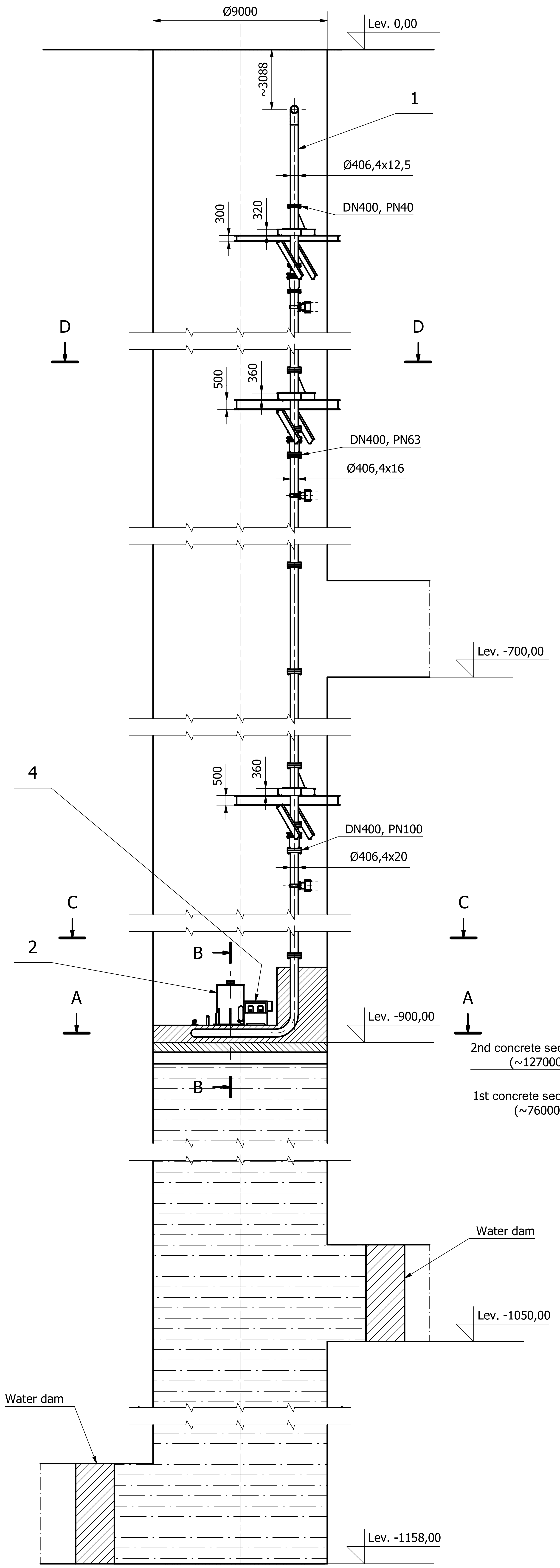
Scale	Mass (kg)	Material
1:1500	-	-

Drawing format A0

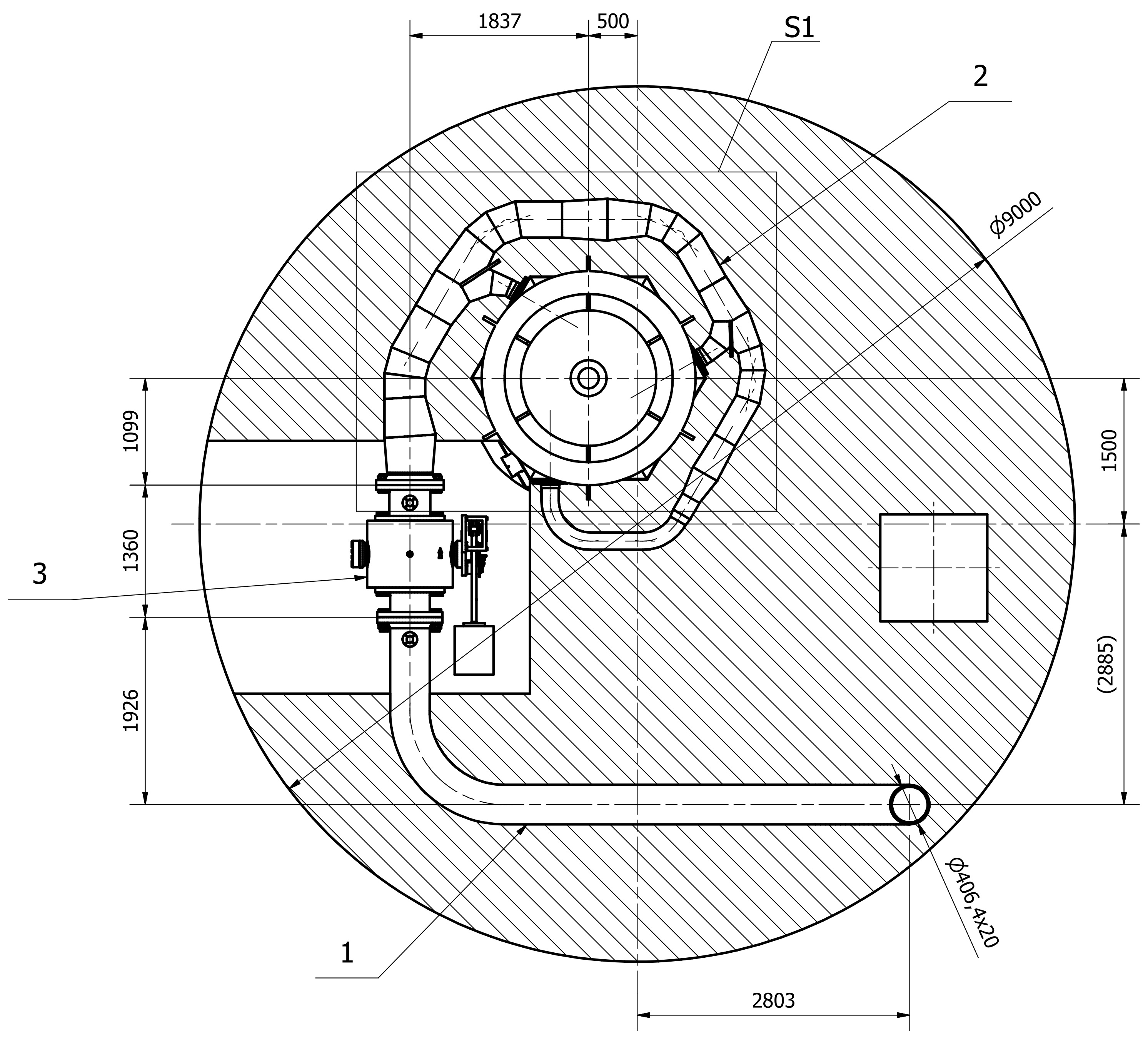
Add No. (or file)

Name of the subject

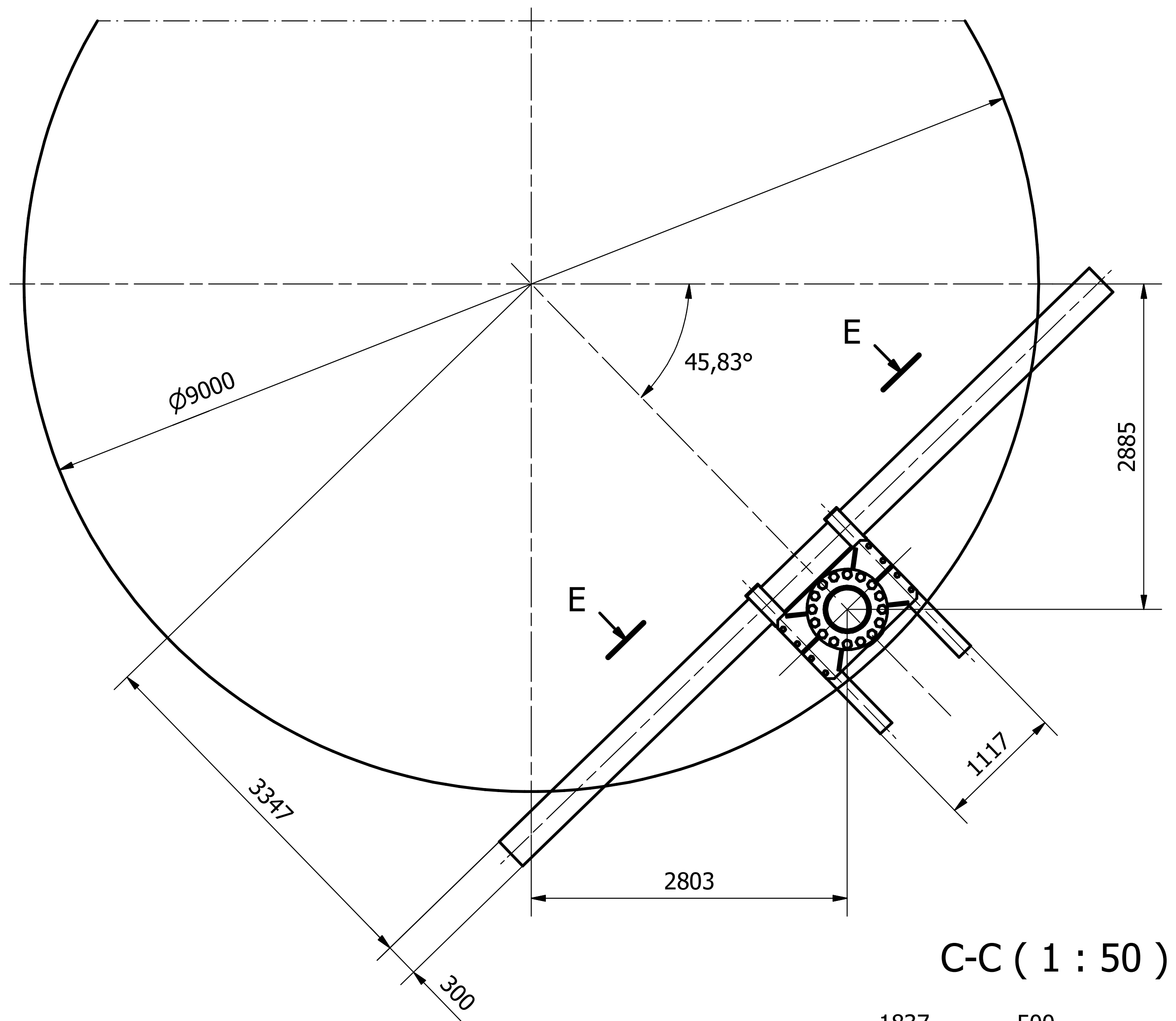
Pumped Storage Hydroelectricity



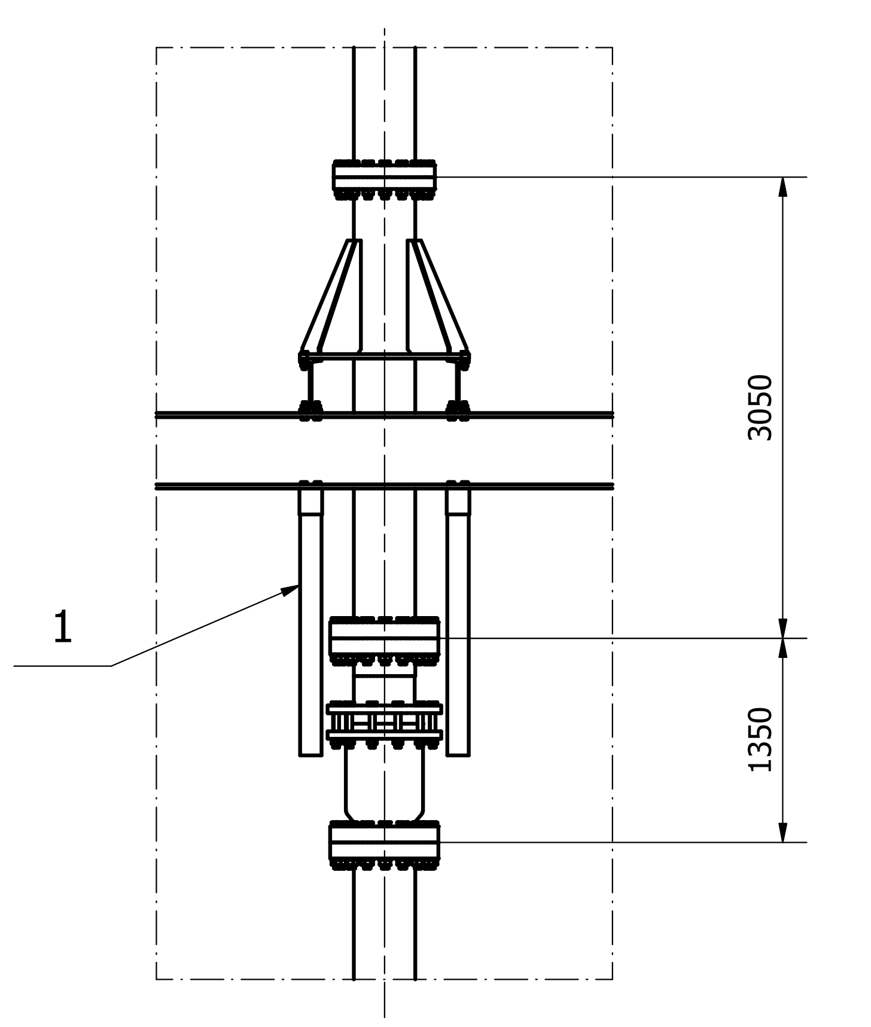
A-A (1 : 50)



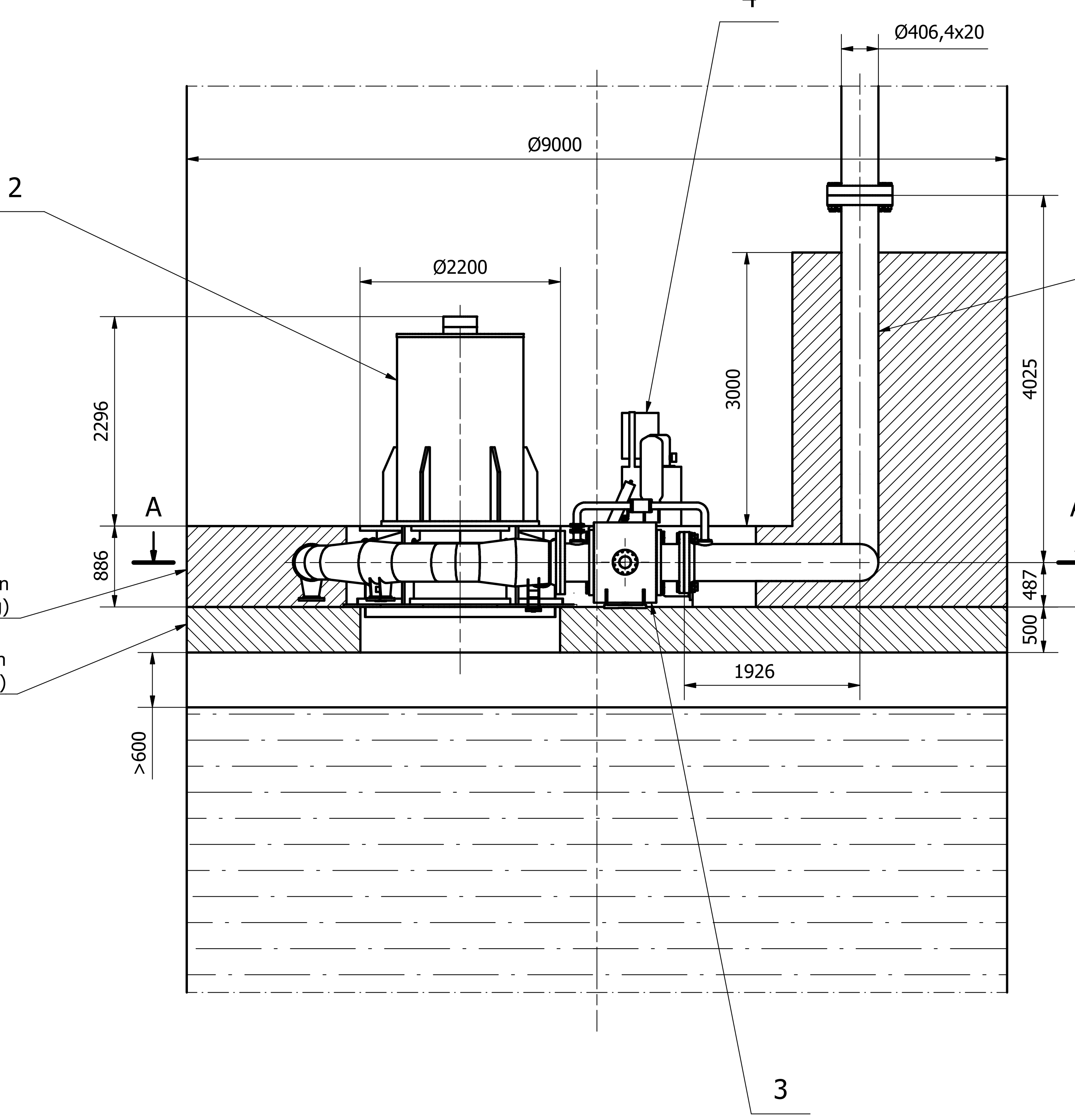
D-D (1 : 50)



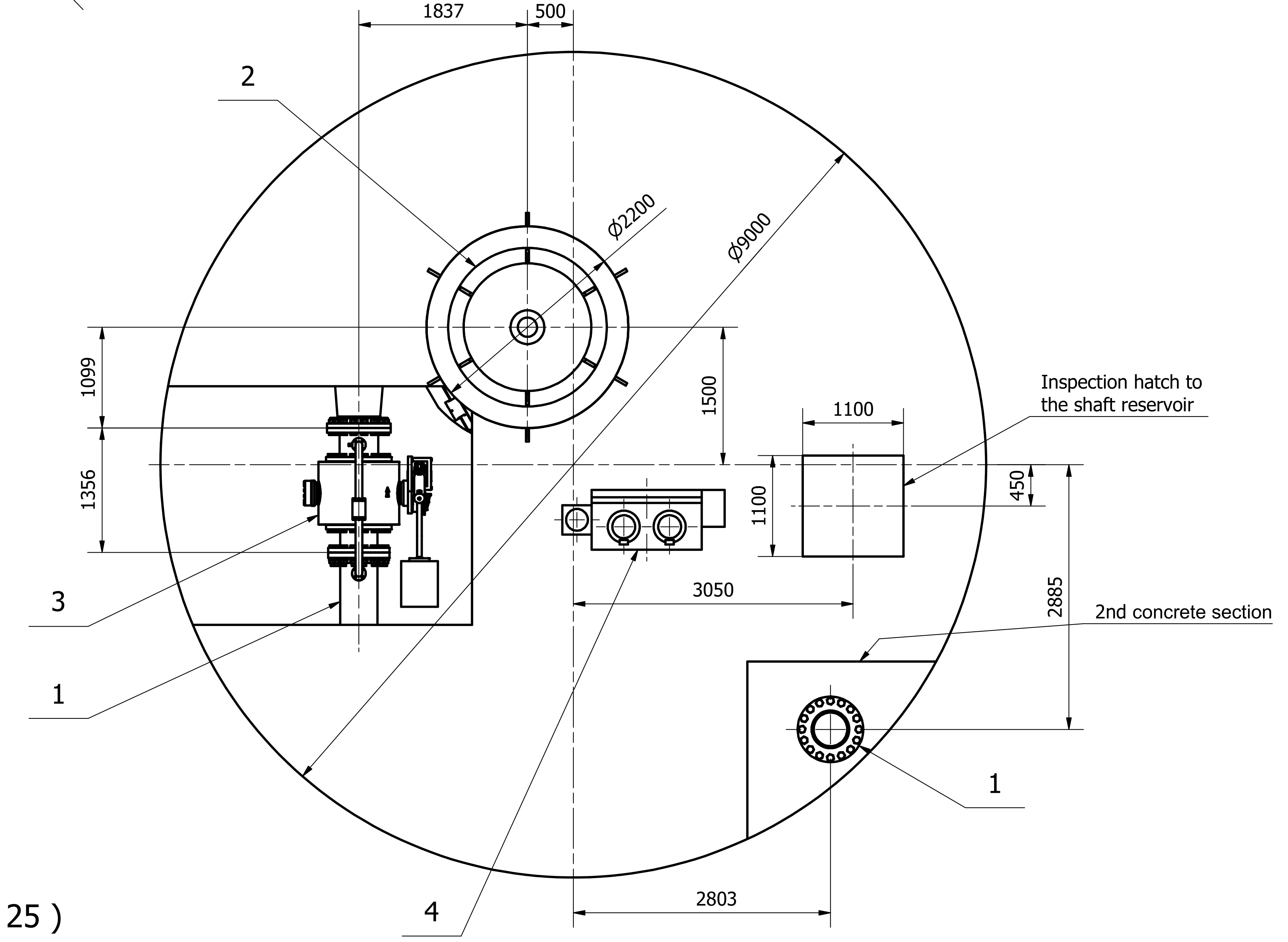
E-E (1 : 50)



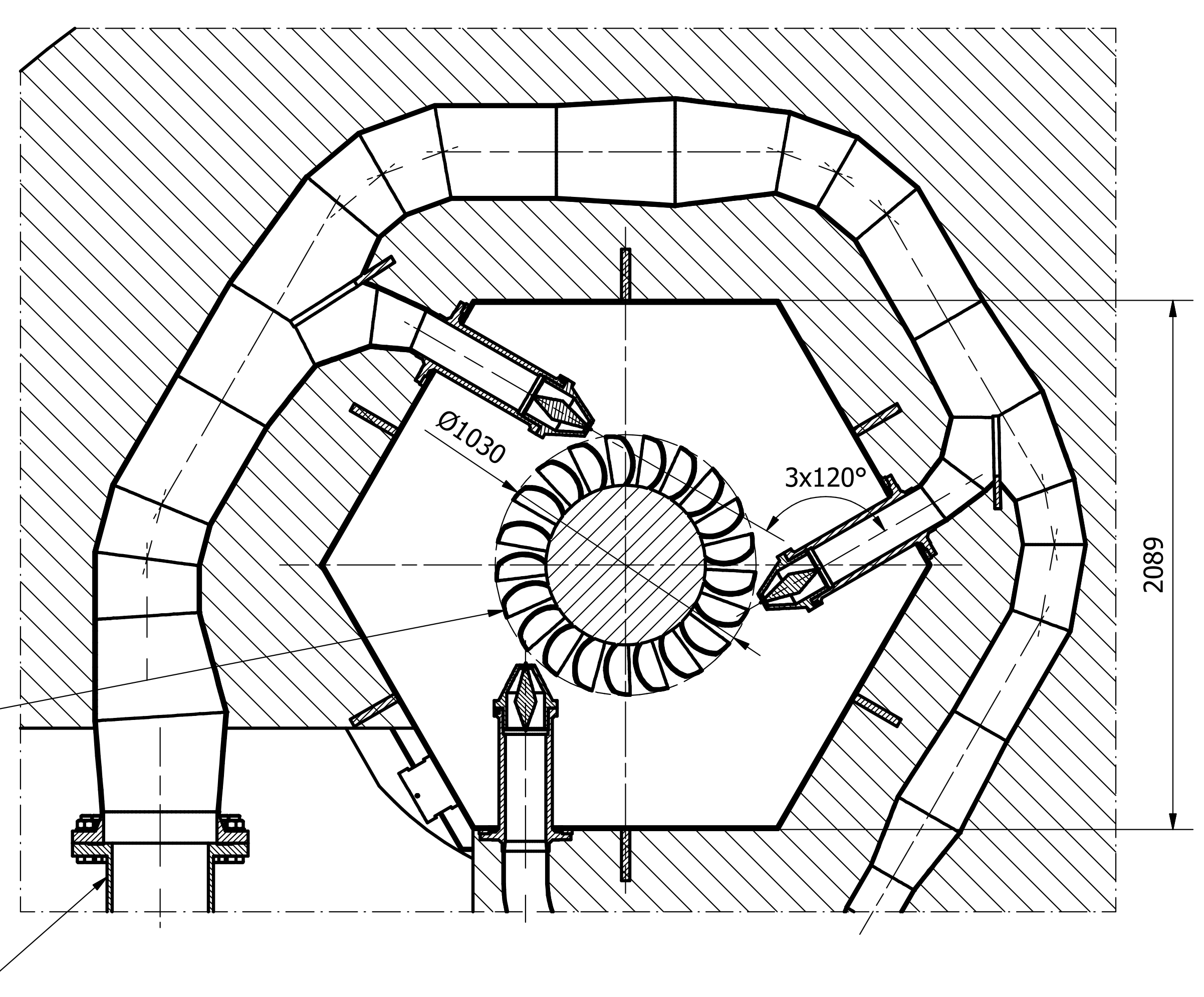
B-B (1 : 50)



C-C (1 : 50)



S1 (1 : 25)



Notes:
1. The detailed description of the turbine feed pipeline (item 1) is shown in drawing No.: W52.010PW01.01.
2. The weight indicated in the drawing title block does not include the weights of the 1st and 2nd concrete sections.

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RFCS Project: HESS
No.: 101112380

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Task supervisor	BG	P. Matusiak		08.2025
Designed by	BG	K. Kwaśny		08.2025
Verified by	BG	M. Bal		08.2025
Approved by	BG	P. Matusiak		08.2025

No.: W52.010PW01.en

Sheet / number of sheets: 1/1

Scale	Mass (kg)	Material
1:150	265285,0	-

Drawing size: A1

Add No. (or file)

Item name:
System of the hydro turbine and generator with the shaft reservoir

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No.: 101112380

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Verified by	BG	M. Bal		08.2025
Approved by	BG	P. Matusiak		08.2025

No.: W52.010PW01.01.en

Sheet / number of sheets
1/2

Scale	Mass (kg)	Material
1:150	245655,0	-

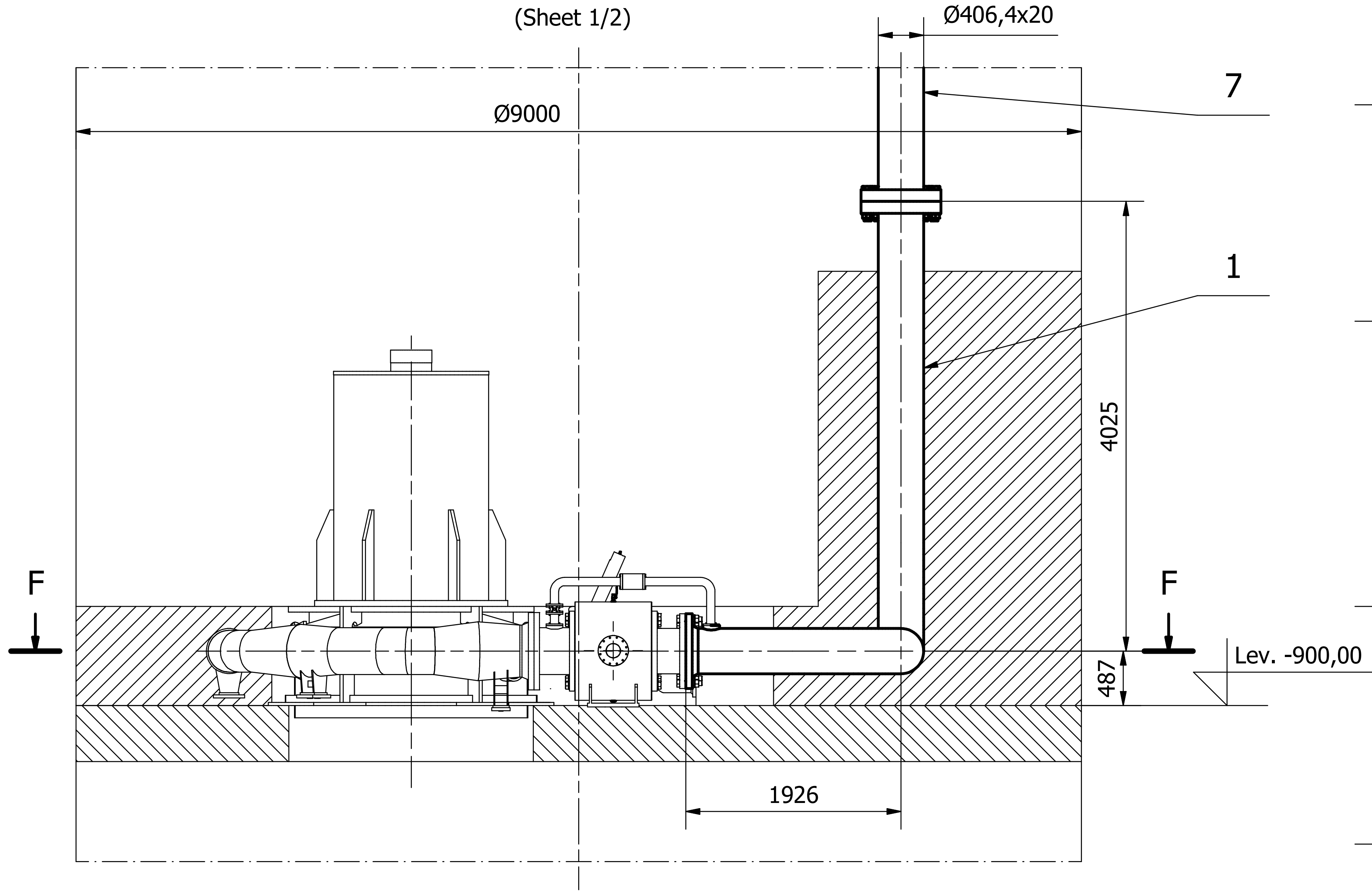
Drawing size
A0

Add No. (or file)

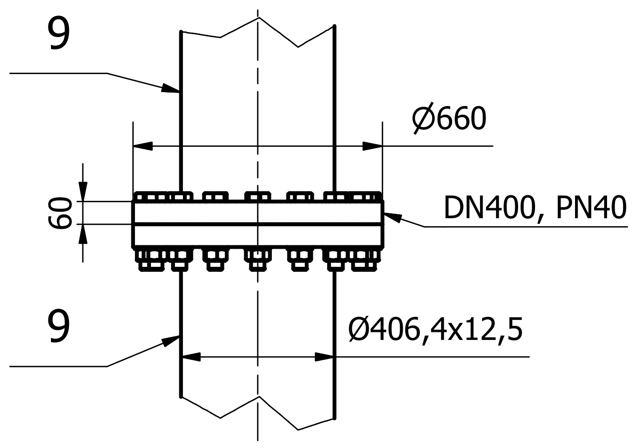
Item name:

Penstock

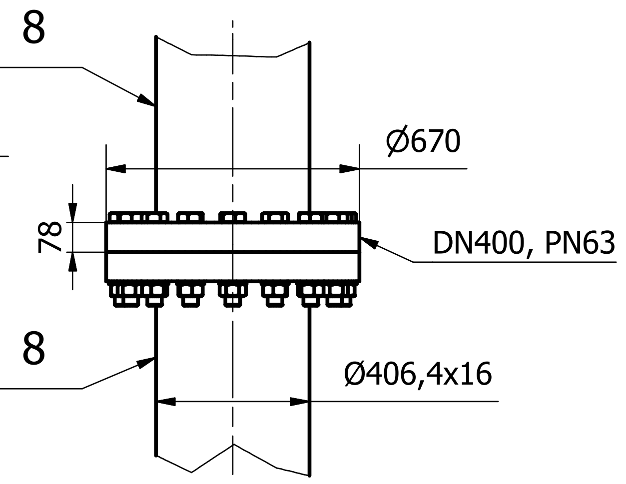
C-C
(Sheet 1/2)



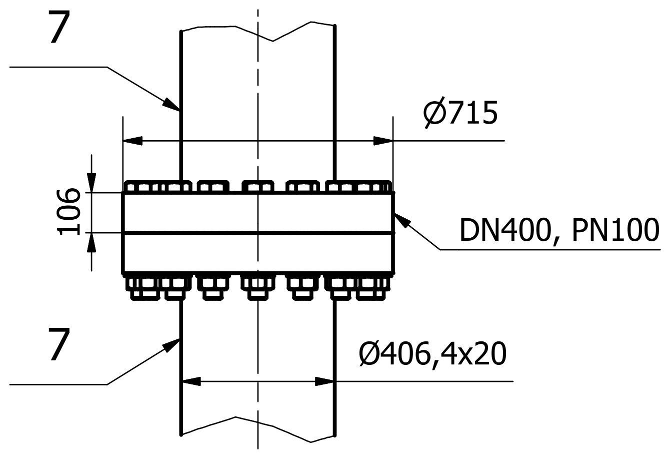
S1 (1 : 20)
(Sheet 1/2)



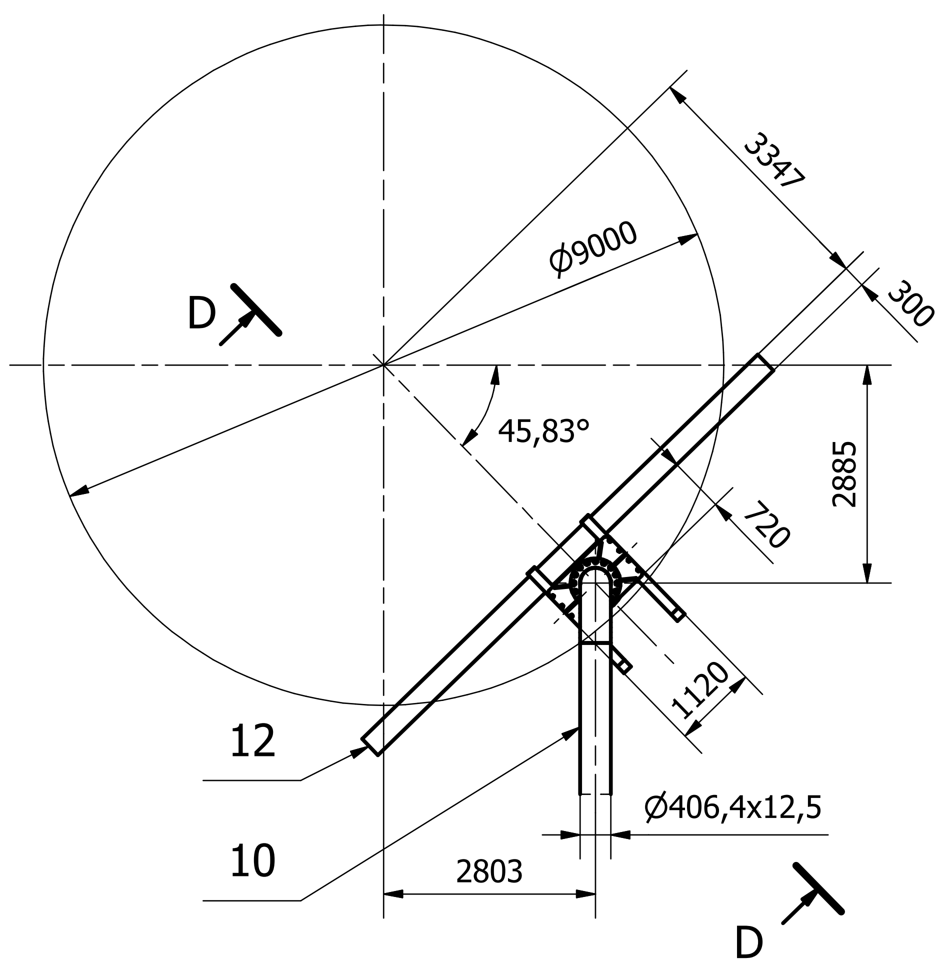
S2 (1 : 20)
(Sheet 1/2)



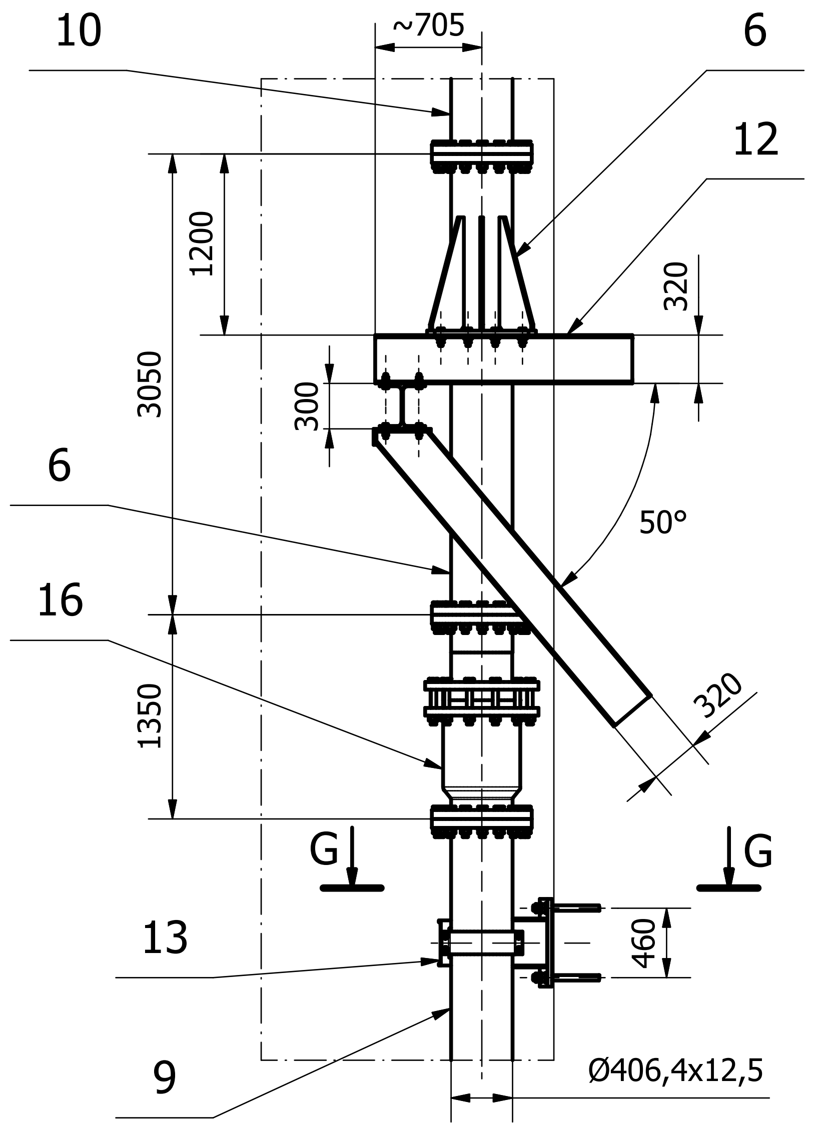
S3 (1 : 20)
(Sheet 1/2)



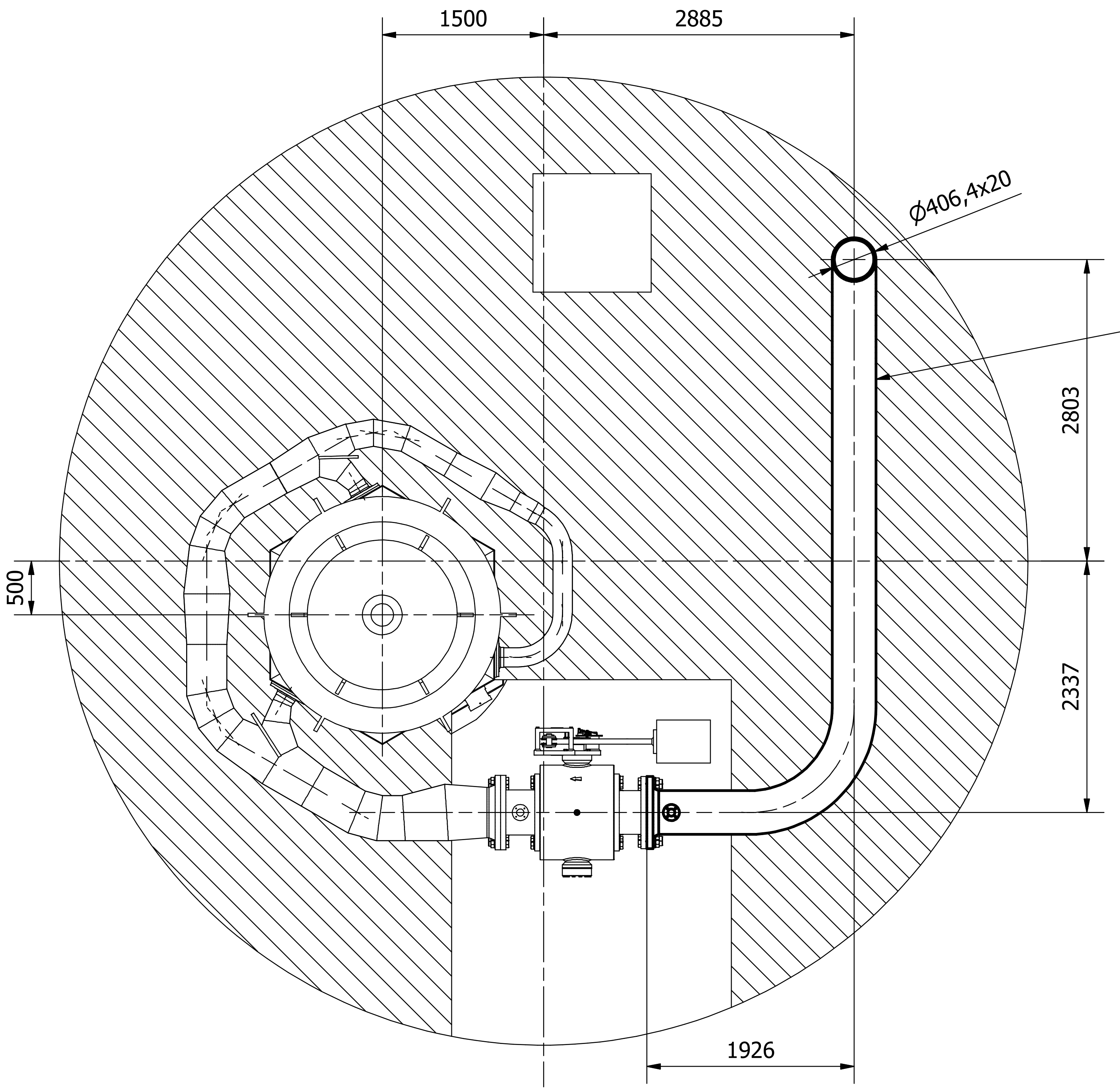
A-A (1 : 100)
(Sheet 1/2)



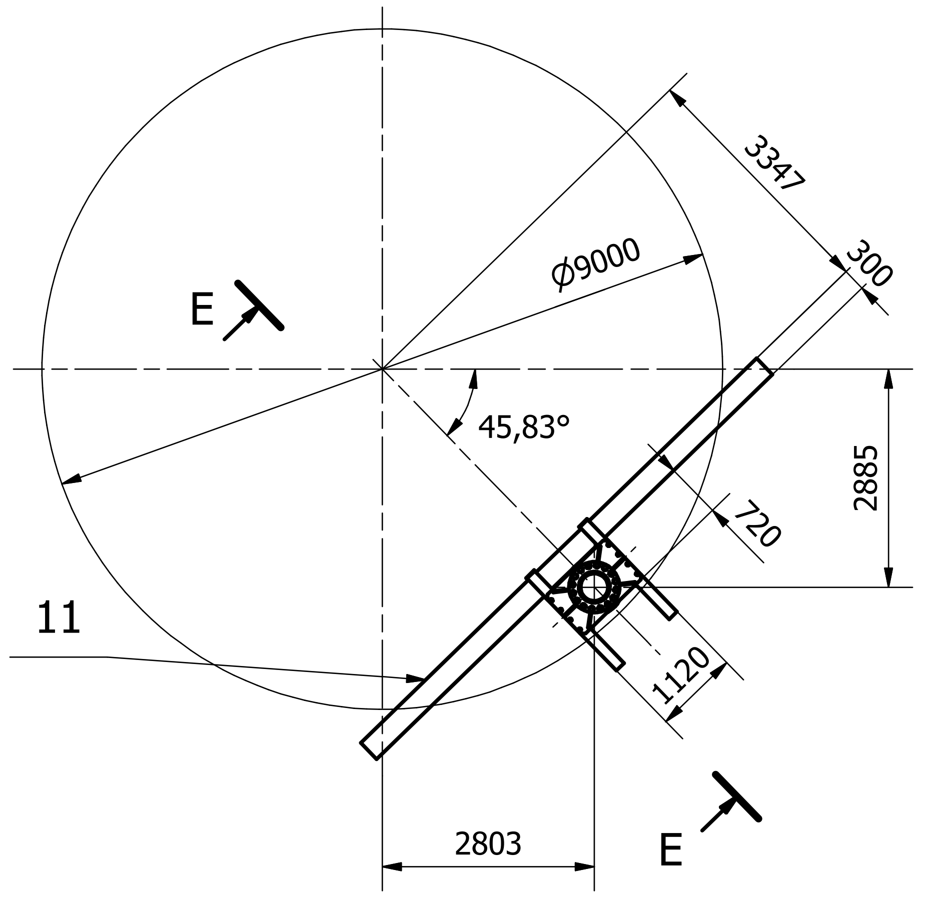
D-D (1 : 50)



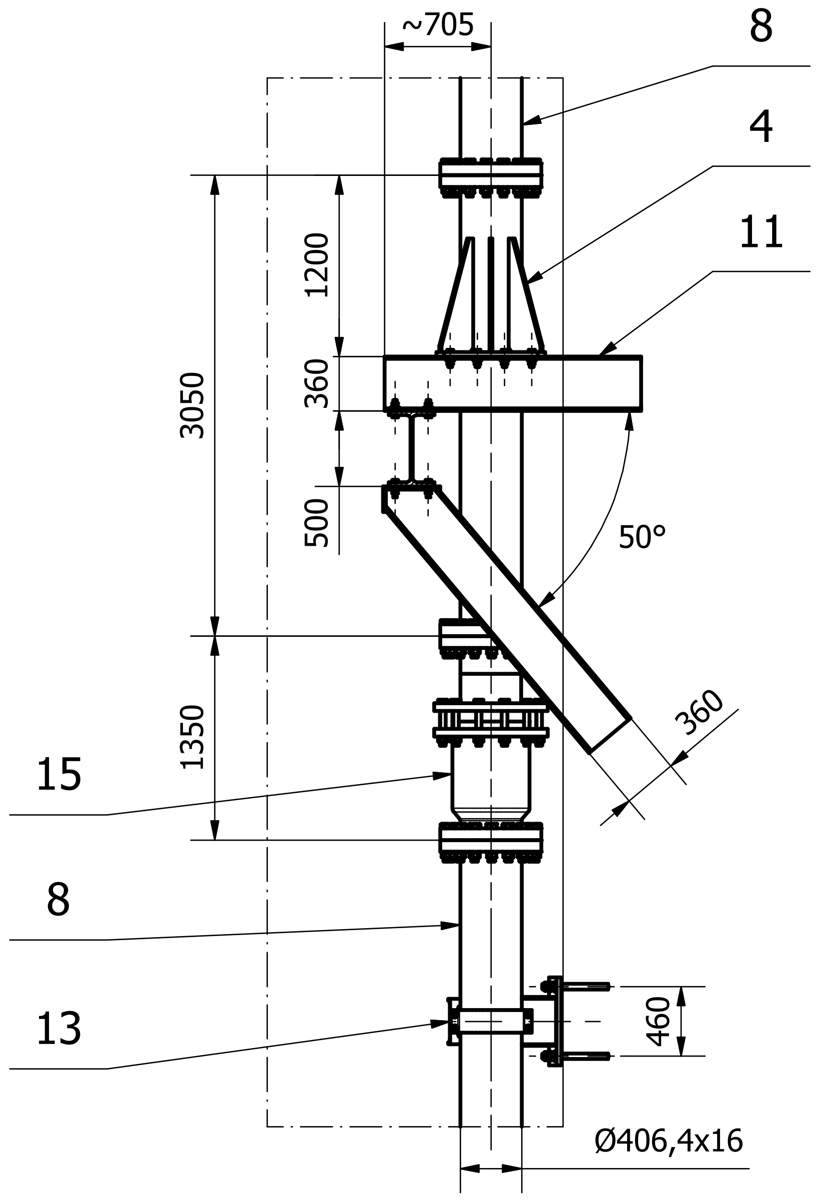
F-F



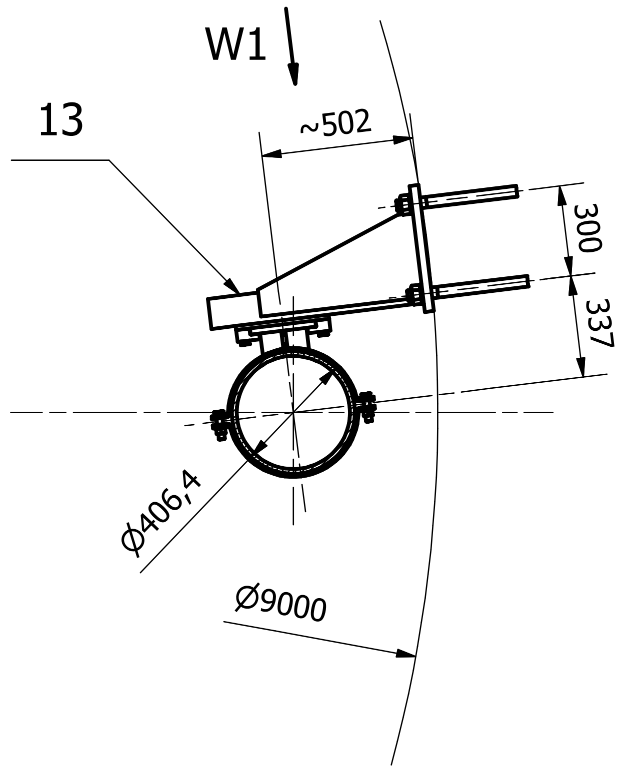
B-B (1 : 100)
(Sheet 1/2)



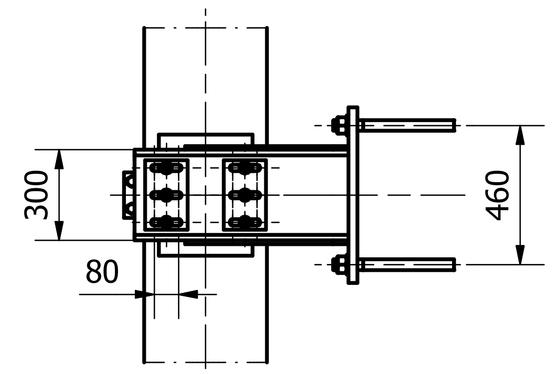
E-E (1 : 50)



G-G (1 : 25)



W1 (1 : 25)



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RFCS Project: HESS
No.: 101112380

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	Department	Surname	Signature	Date
Task supervisor	BG	P. Matusiak		08.2025
Designed by	BG	K. Kwaśny		08.2025
Verified by	BG	M. Bal		08.2025
Approved by	BG	P. Matusiak		08.2025

No.: W52.010PW01.01.en

Sheet / number of sheets 2/2

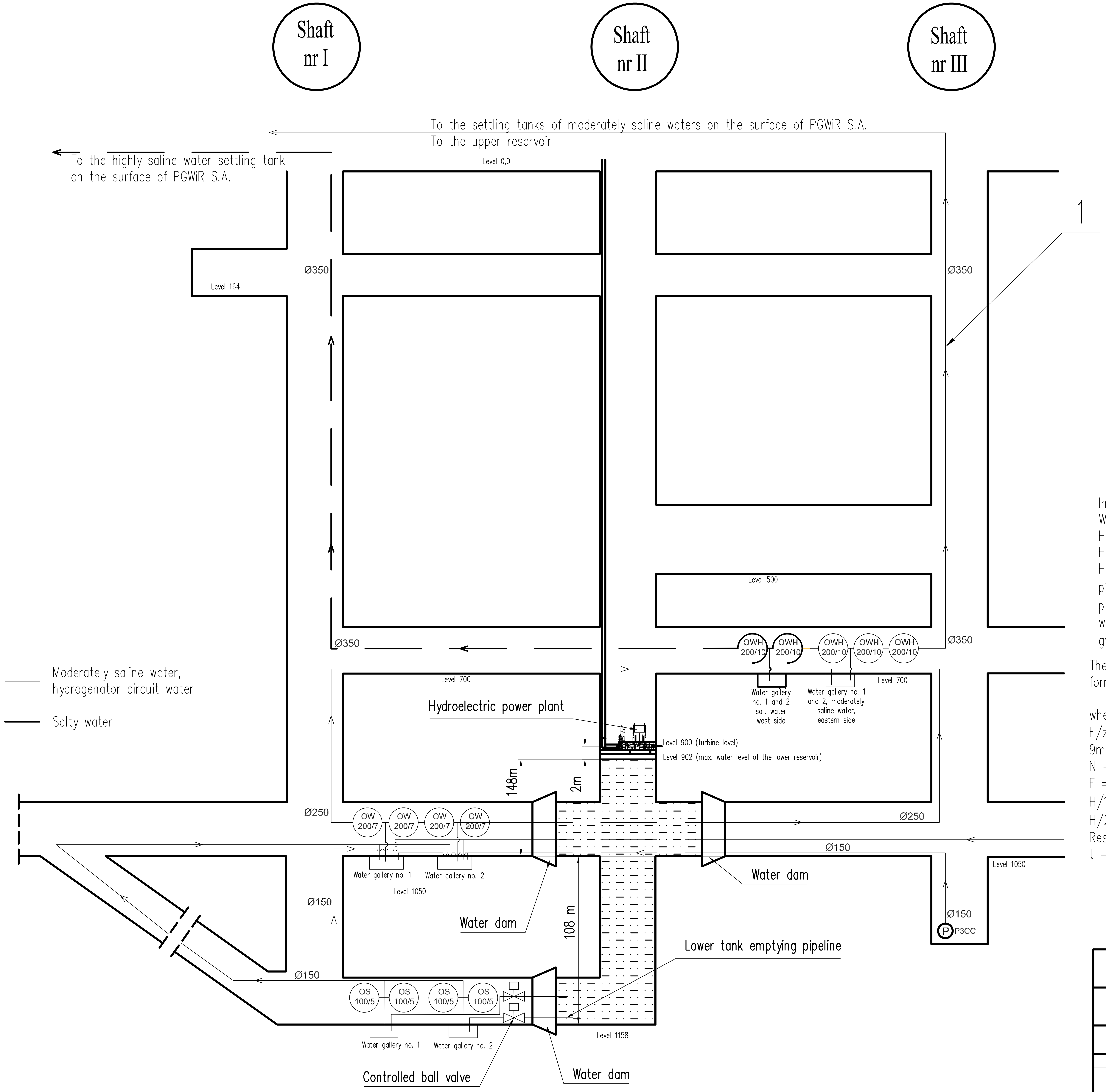
Scale	Mass (kg)	Material
1:50	245655,0	-

Drawing size A2

Add No. (or file)

Item name:

Penstock



Input Parameters for Dam Design

Water Head:

Ht1 – for a dam at level 1050, Ht1 = 1050 – 902 = 148m

Ht2 – for a dam at level 1158, Ht2 = 1158 – 902 = 256m

Hydrostatic Pressure

p1 – for a dam at level 1050, p1 = 148m x 9.81kN/m3 = 1.45MPa

p2 – for a dam at level 1158, p2 = 256m x 9.81kN/m3 = 2.51MPa

where:

gw = 9.81 kN/m3 – specific gravity of water

The emptying time of the lower reservoir is determined using the formula:

$$t = \frac{2F_{zb}}{\mu \cdot F_p \cdot \sqrt{2g}}\left(\sqrt{H_1} - \sqrt{H_2}\right)$$

where:

F/zb = 62m2/ – cross-sectional area of the lower reservoir for D = 9m

N = 0.6 – discharge coefficient

F = 0.035m2/ – discharge cross-sectional area – two DN150 pipes

H/1 = 256m – maximum water level – 256m

H/2 = 1m – minimum water level – 1m

Results

t = 21287s = 5.9 hours

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	Department	Surname	Signature	Date
Task supervisor	BG	P. Matusiak		10.2025
Designed by	BU	K. Kaczmarczyk		10.2025
Verified by	BU	D. Rupik		10.2025
Approved by	BG	P. Matusiak		10.2025

No.: **W52.010PW02.en**

Sheet/number of sheets: 1/1

Scale	Mass (kg)	Material
–	–	–

CAD editing:

Drawing size: A2

Add No. (or file)

Item name: **Water pumping system**

INSTITUTE OF MINING TECHNOLOGY
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RFCS Project: HESS

No.: 101112380

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	Department	Surname	Signature	Date
Task supervisor	BG	P. Matusiak		10.2025
Designed by	BG	R. Baron		10.2025
Verified by	BG	M. Bal		10.2025
Approved by	BG	P. Matusiak		10.2025

No.: W52.010PW02.01.en

Sheet / number of sheets: 1/1

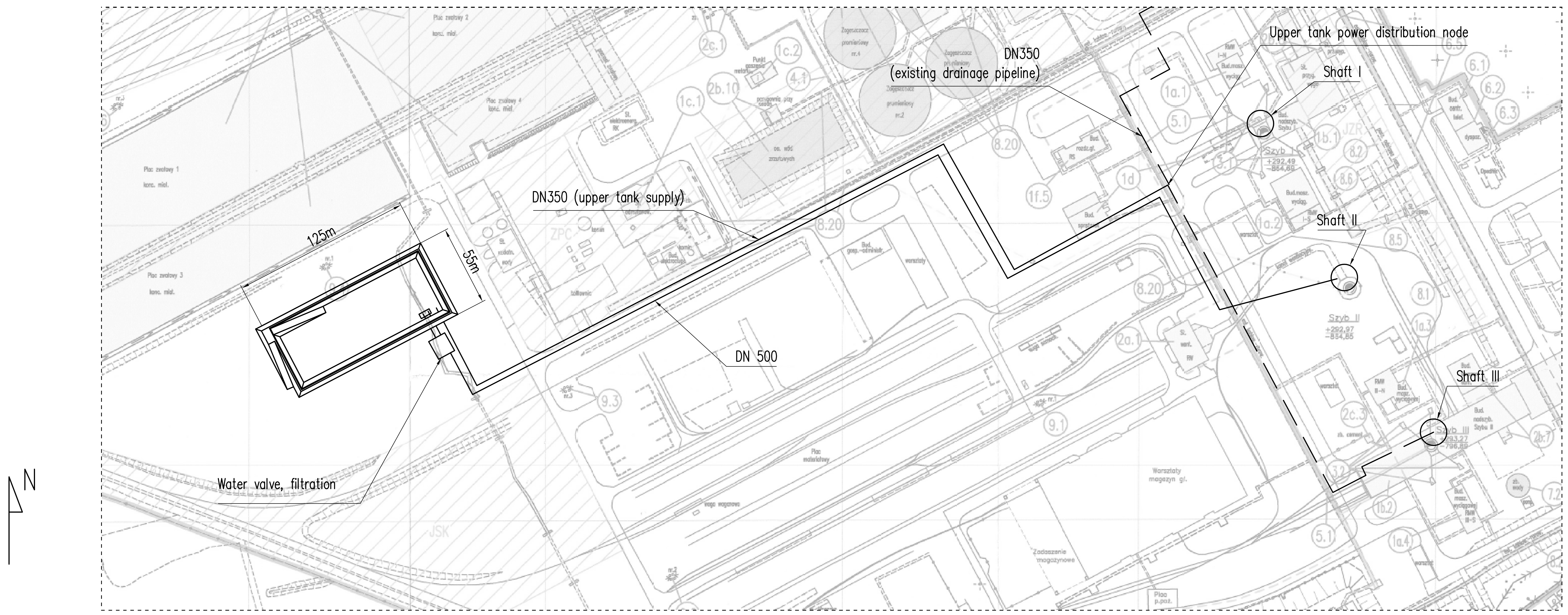
Scale	Mass (kg)	Material
1:150	-	-

Drawing size: A0

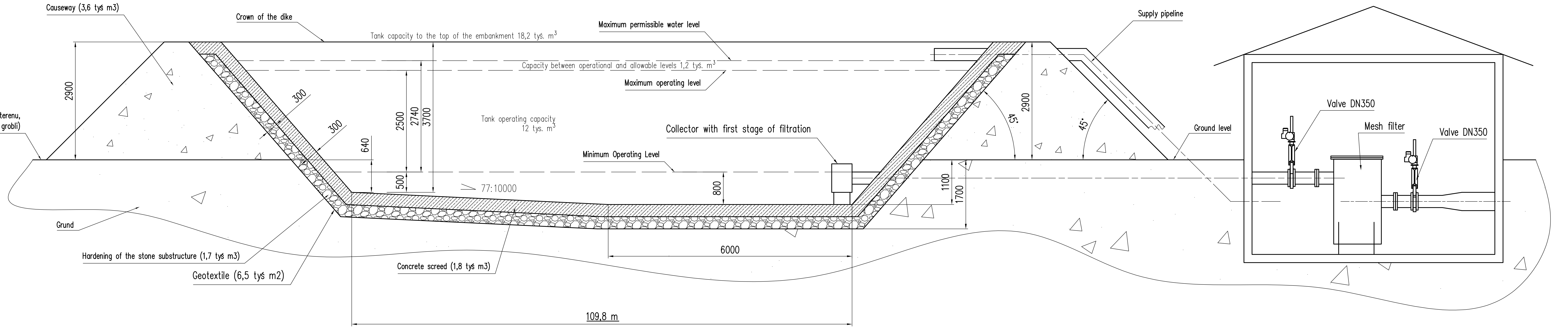
Add No. (or file)

Item name: Existing dewatering pipeline

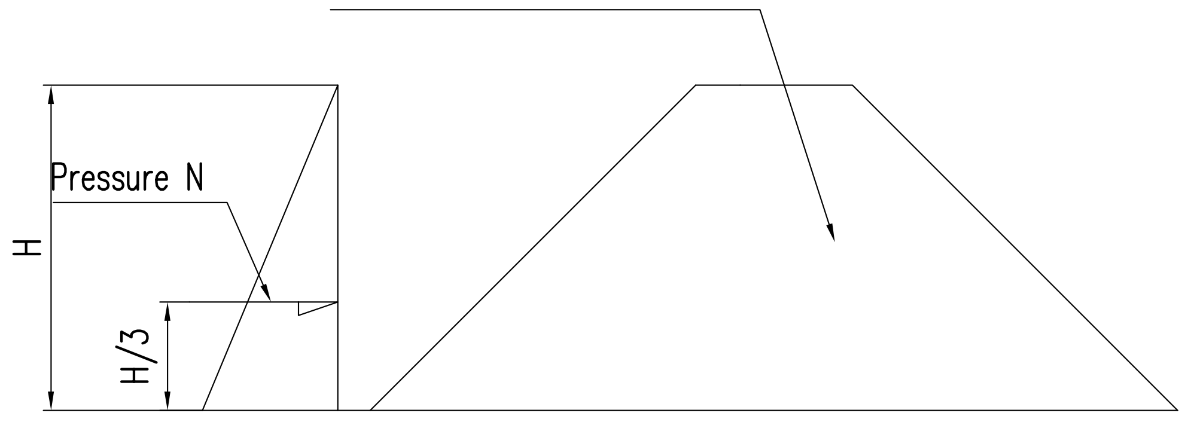
Skala 1:2000



Skala 1:50



Cross-section of the dike A



Data:

- dyke cross-sectional area A = 12.5 m^2
- water column height H = 2.9 m
- specific gravity of water gw = 9.81 kN/m^3
- dyke volumetric density 18 kN/m^3 (homogeneous dyke)
- dyke internal friction angle alpha = 30 degrees
- impermeable dyke

Results:

Unit hydrostatic pressure	N = 1/2 * gw * H = 41.2 kN/m
Unit dyke weight Gg	G = A * gg = 224.5 kN/m
Unit soil friction force T	T = Gg * tan(alpha) = 129.5 kN/m
Safety factor	X = T / N = 3.14

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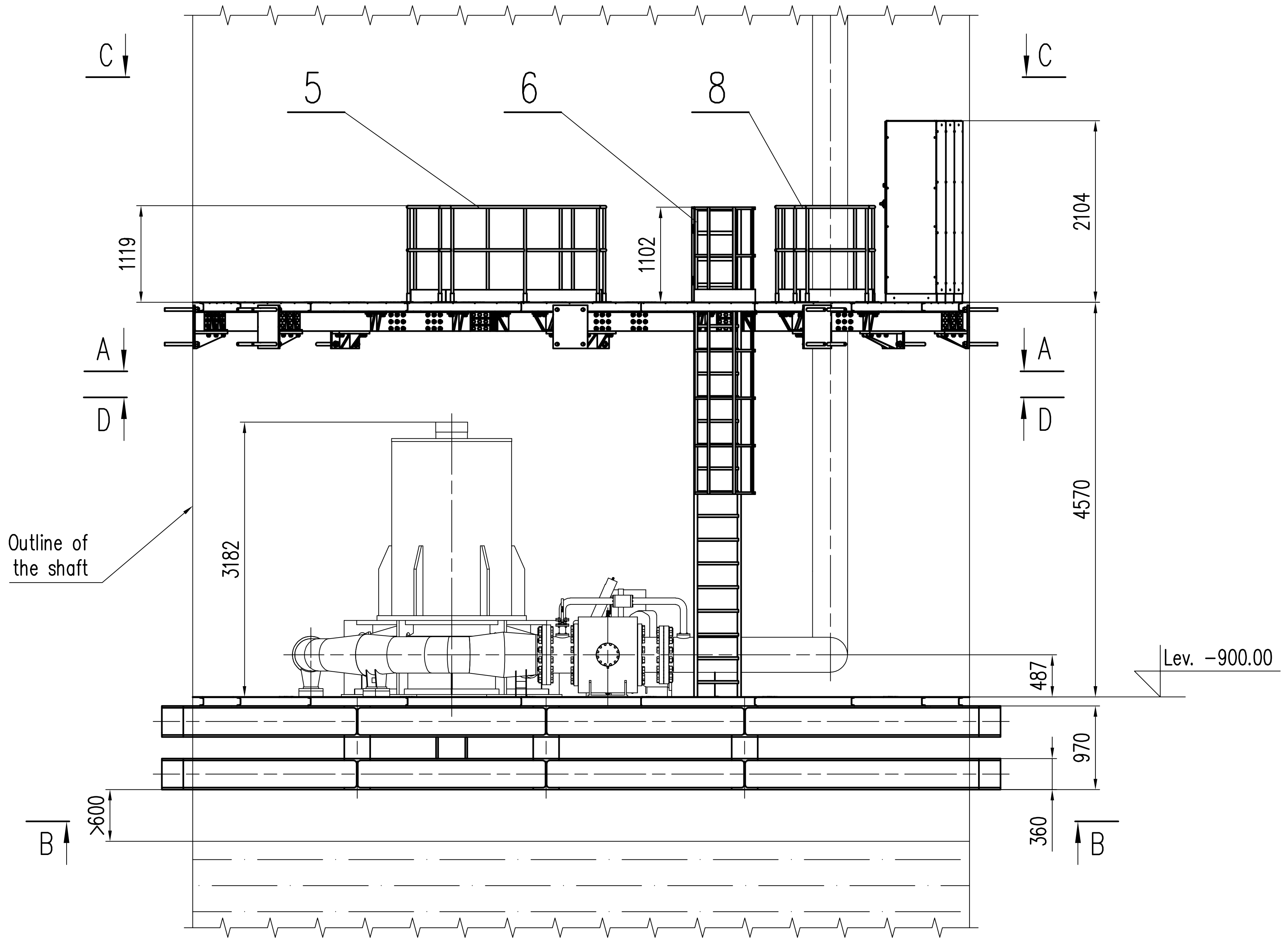
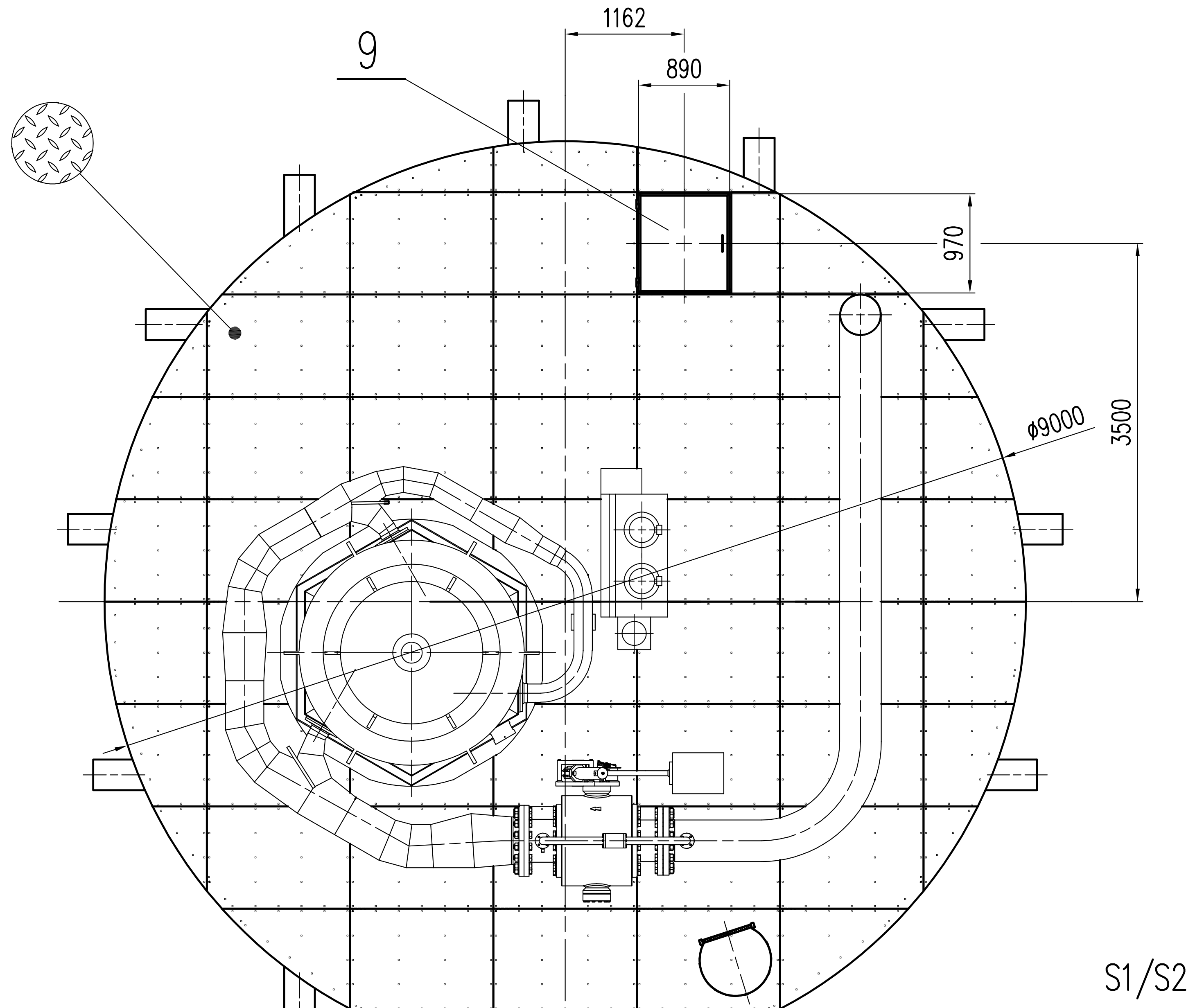
	Designed by	BU	K. Kaczmarczyk		07.2025
RFCS Project: HESS No.: 101112380	Verified by	BU	D. Rupik		07.2025
	Approved by	BG	P. Matusiak		07.2025
Using the technical solutions presented in the drawing requires written permission from KOMAG	No.: W52.010PW03.en				Sheet/number of sheets 1/1
Scale: 1:2000 / Mass (kg): – / Material: –	Item name: Upper reservoir				
CAD editing: Drawing size A1 / Add No. (or file)					

Department / Surname / Signature / Date

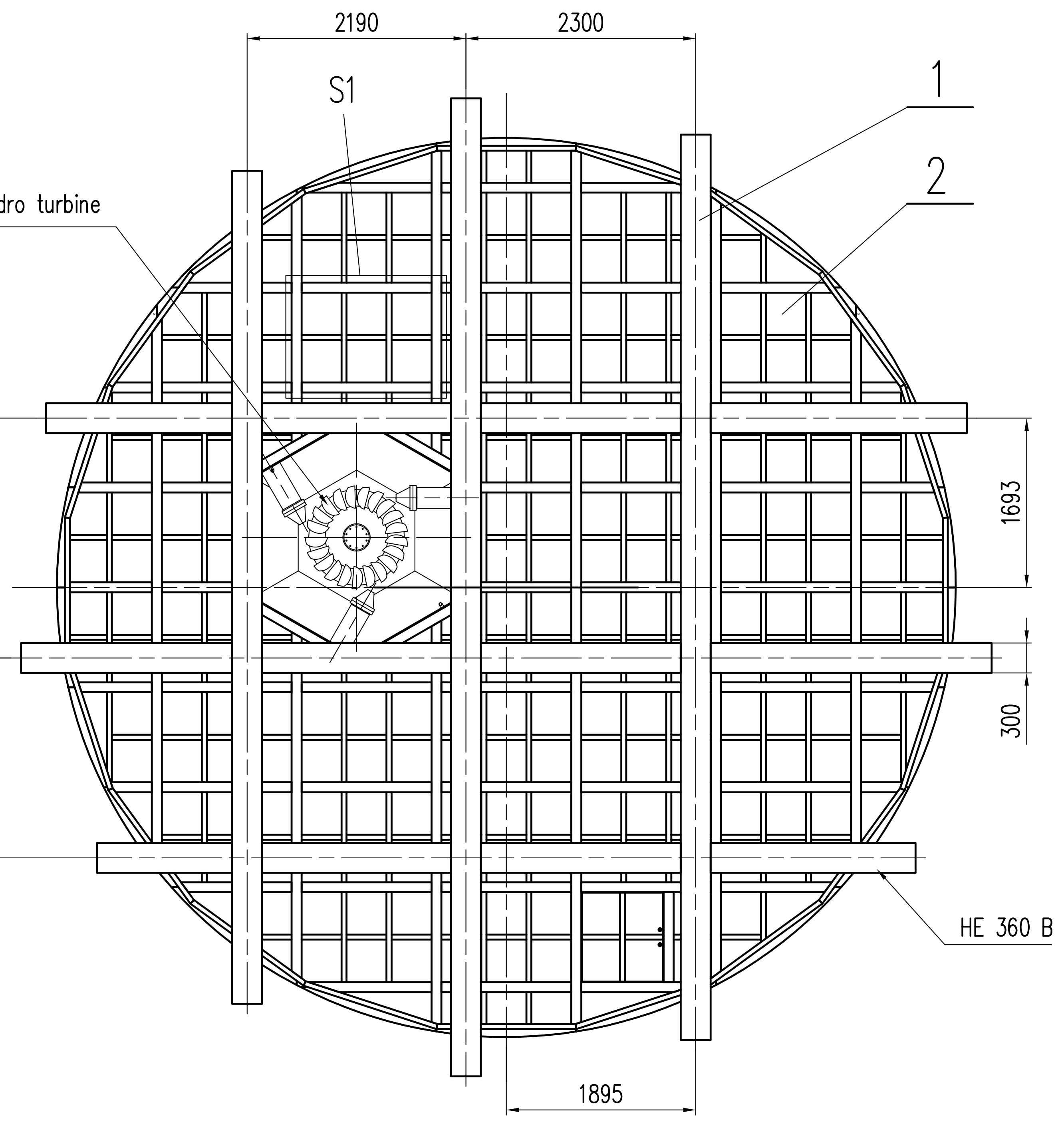
Variant I



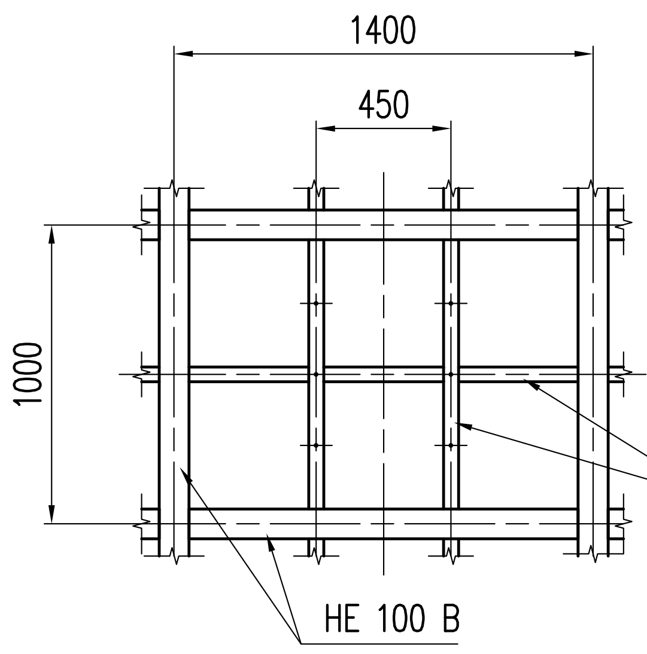
A–A



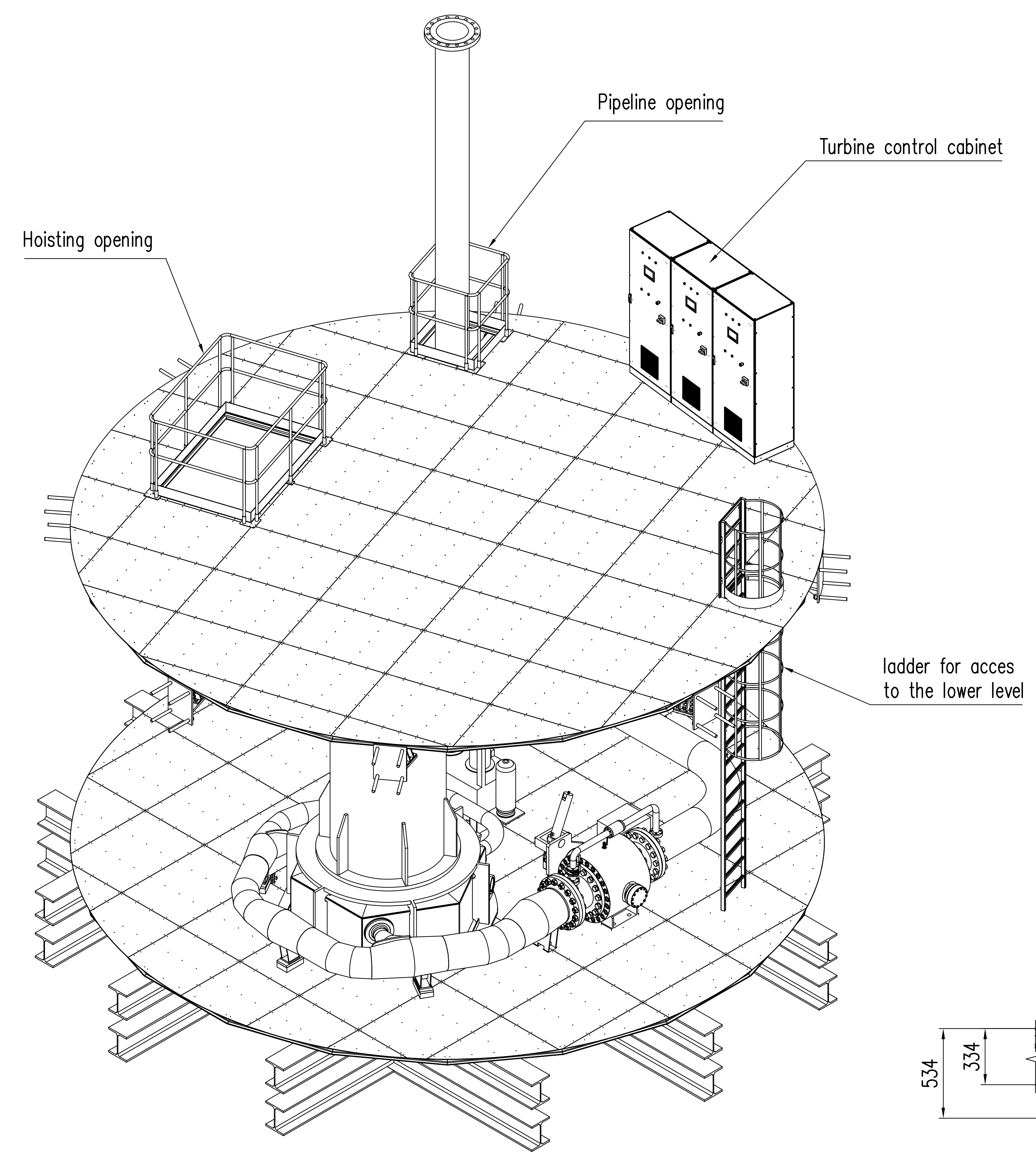
B–B



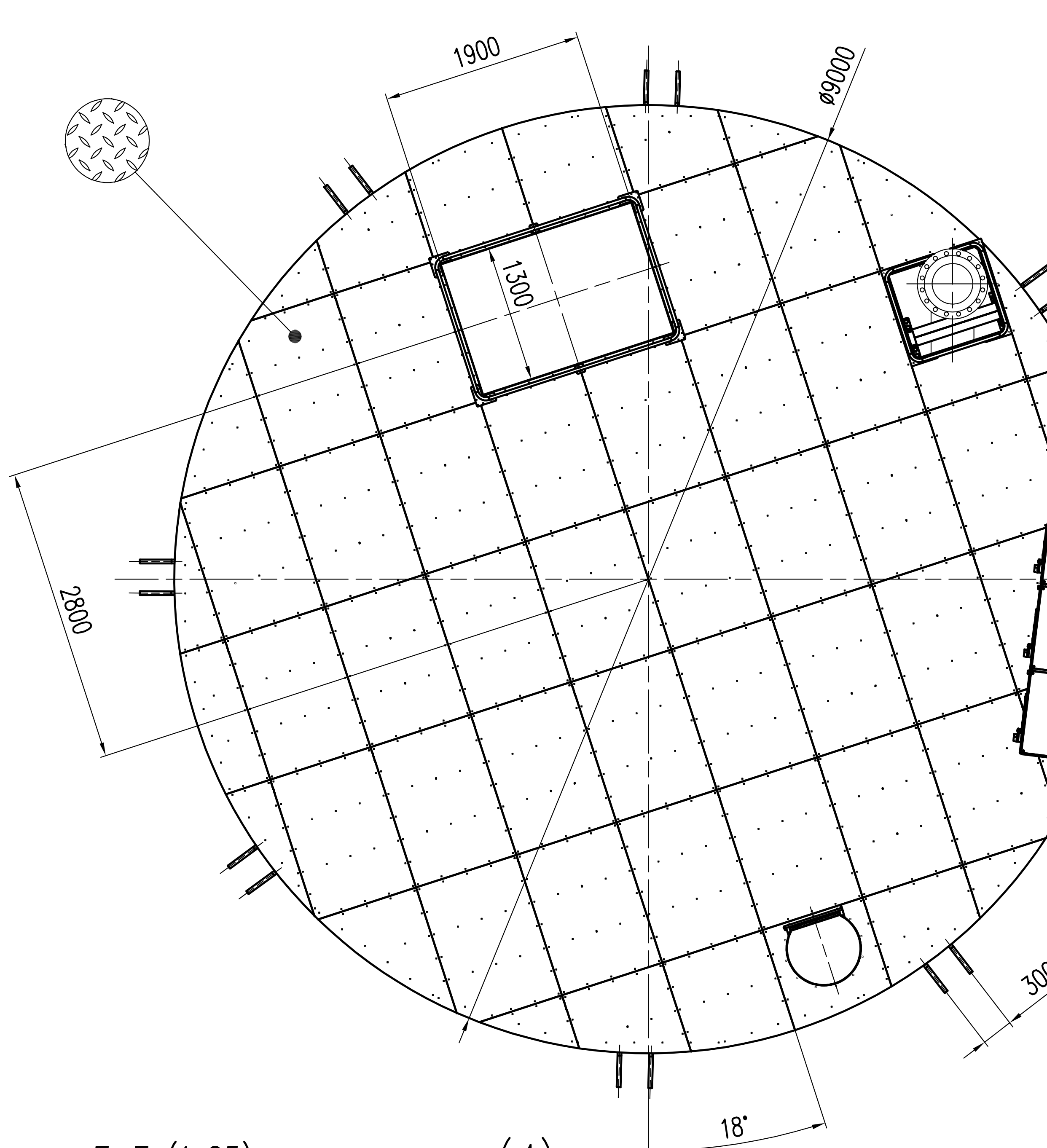
S1/S2 (1:25)



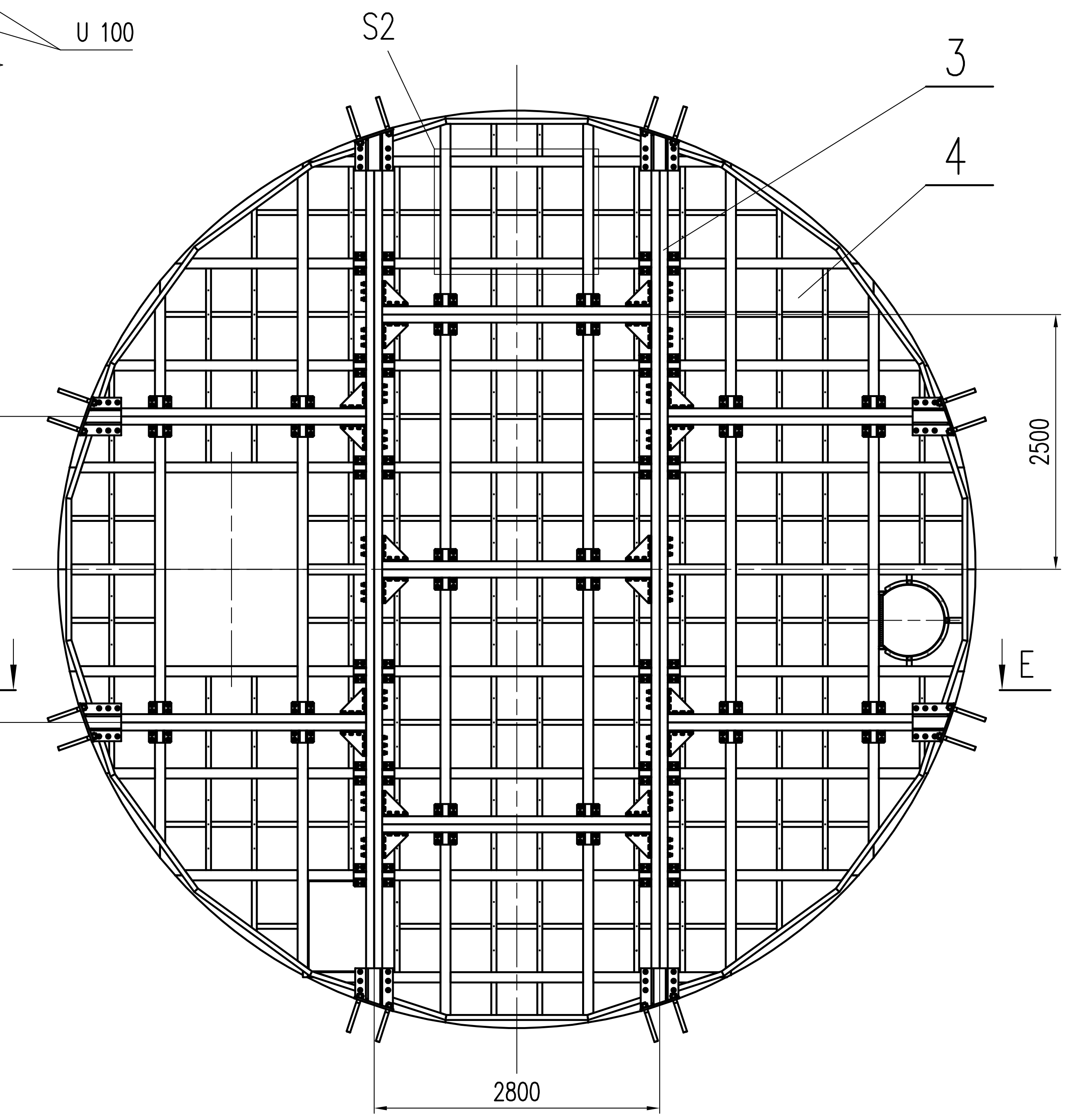
Isometric view



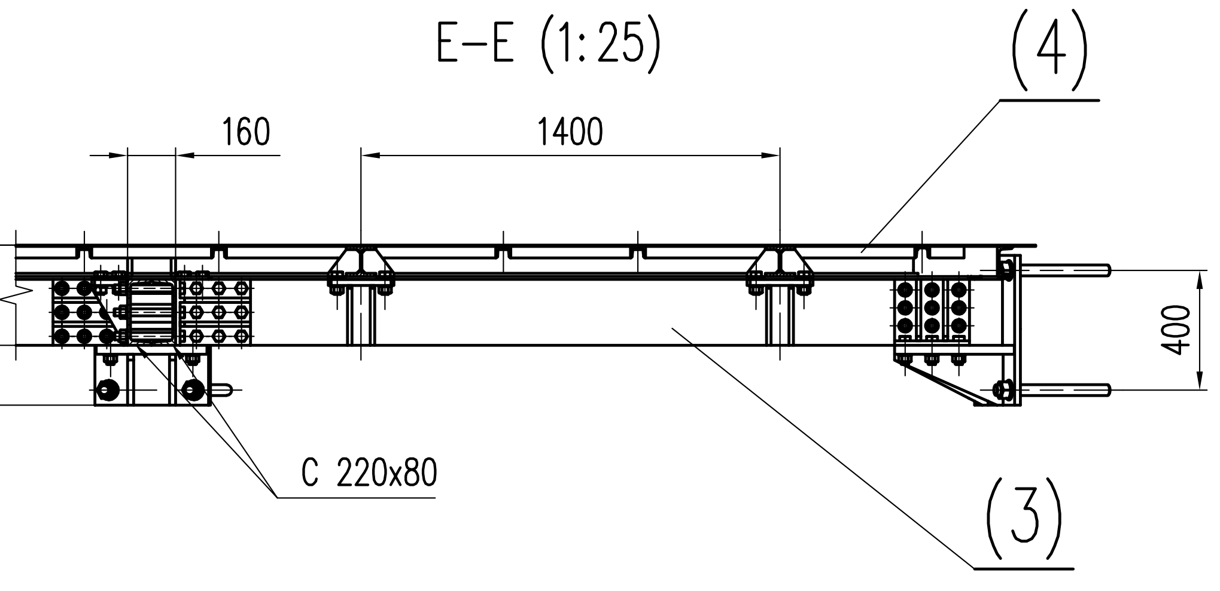
C–C



D–D



E–E (1:25)



Notes:

1. There are two variants of the assembly: variant I – shown on the sheet 1 with the lower level platform made of steel structure, and variant II – shown on the sheet 2 with the lower level platform made of concrete sections.
2. The mass of variant I assembly is 29855 kg. For variant II assembly, the mass is 9698 kg and does not include the mass of concrete sections.
3. The platform floor must be made of chequered steel plate for anti-slip surface.

INSTITUTE OF MINING TECHNOLOGY **KOMAG**	Task supervisor	BG	P. Matusiak	10.2025
	Designed by	BU	D. Rupik	10.2025
RFCS Project: HESS No.: 101112380	Verified by	BU	K. Kaczmarczyk	10.2025
	Approved by	BG	P. Matusiak	10.2025

Department | Surname | Signature | Date

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No.: W52.010PW04.en

Sheet/number of sheets 1/2

Scale	Mass (kg)	Material
1:50	notes	–

CAD editing:

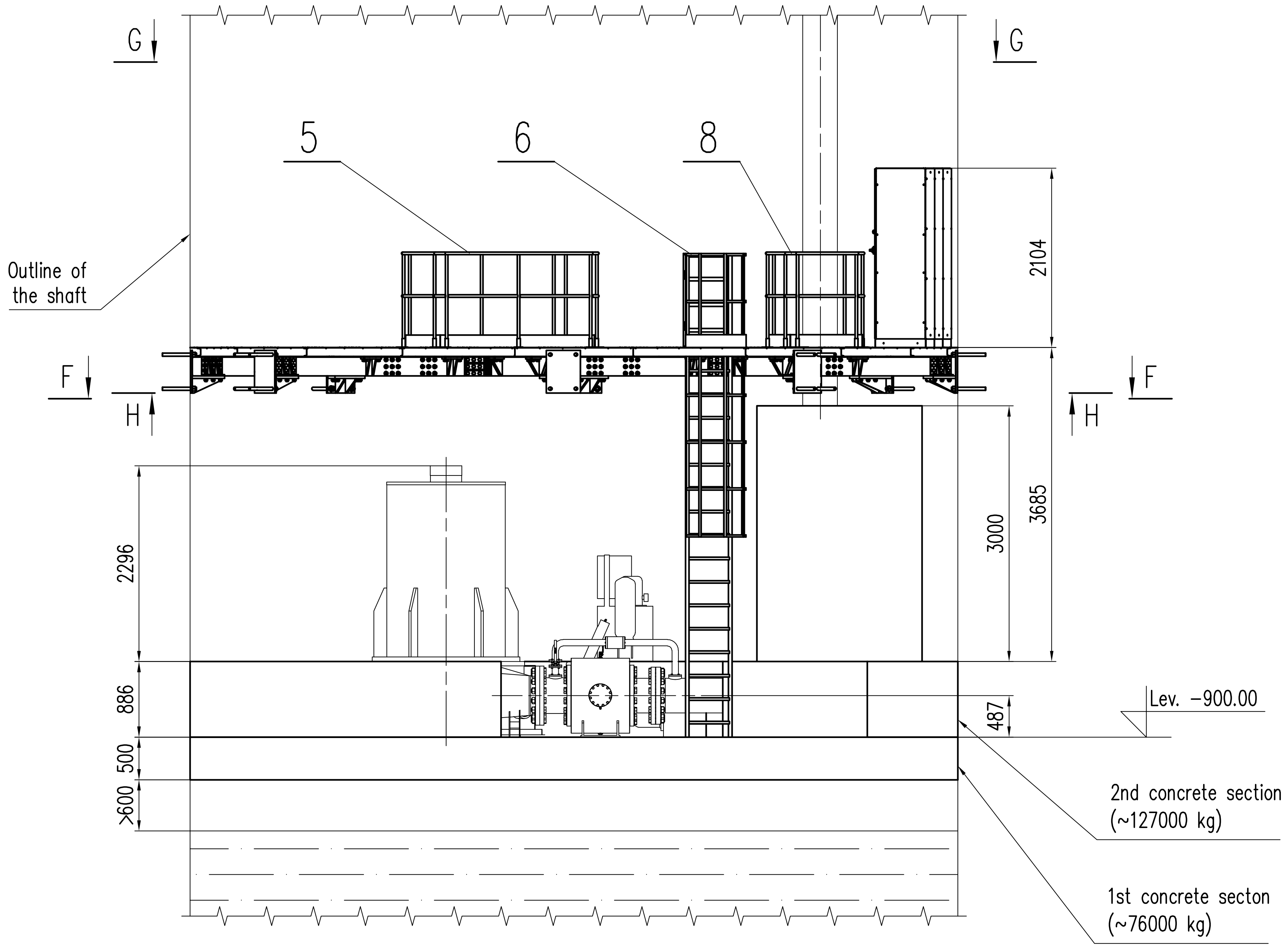
Drawing size A1

Add No. (or file)

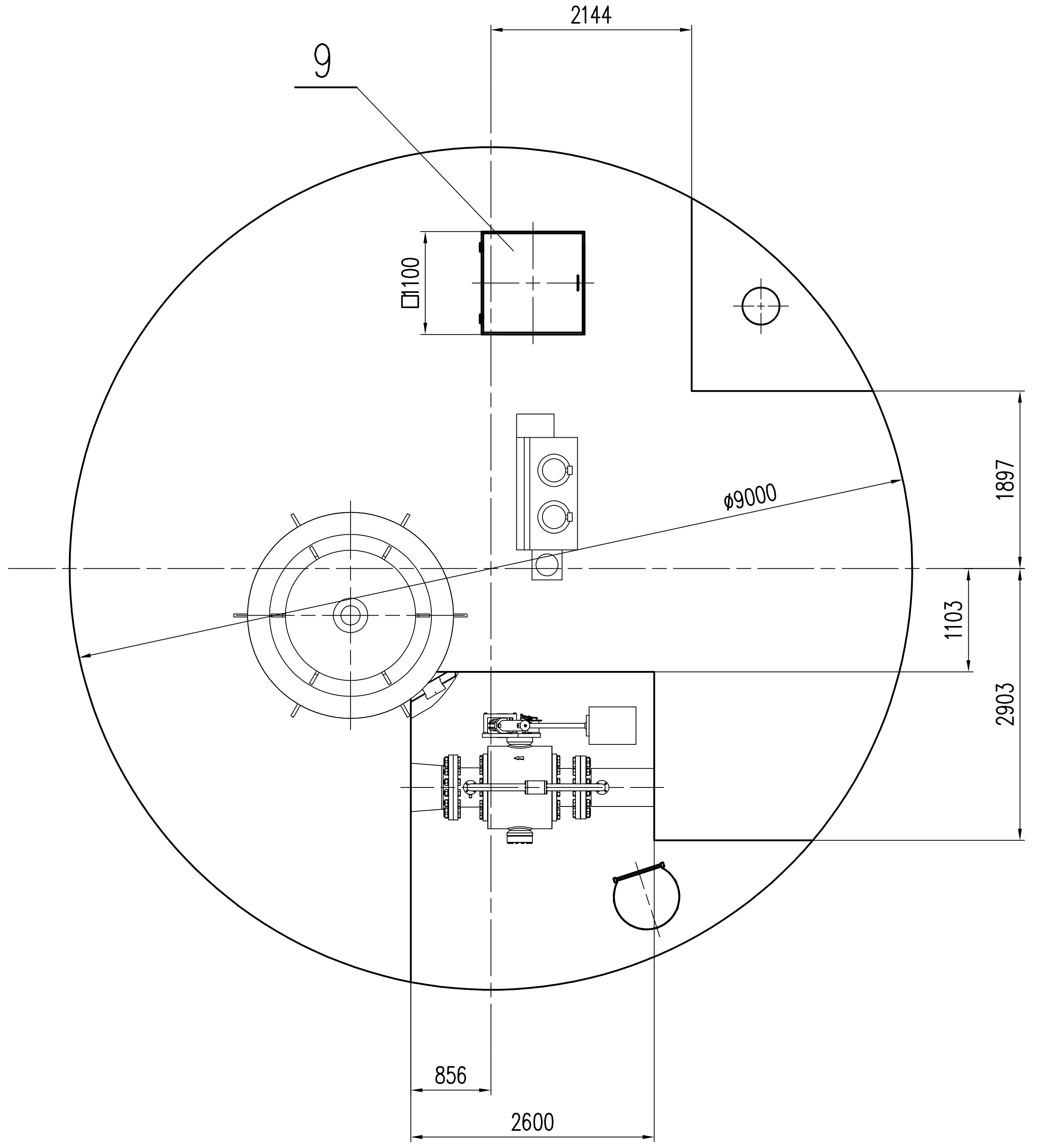
Item name:

Platform assembly

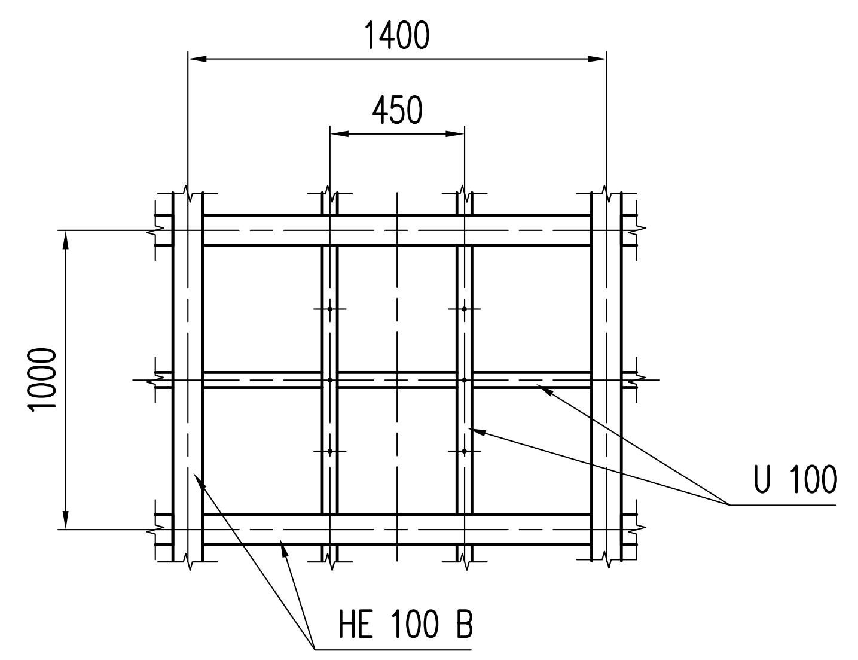
Variant II



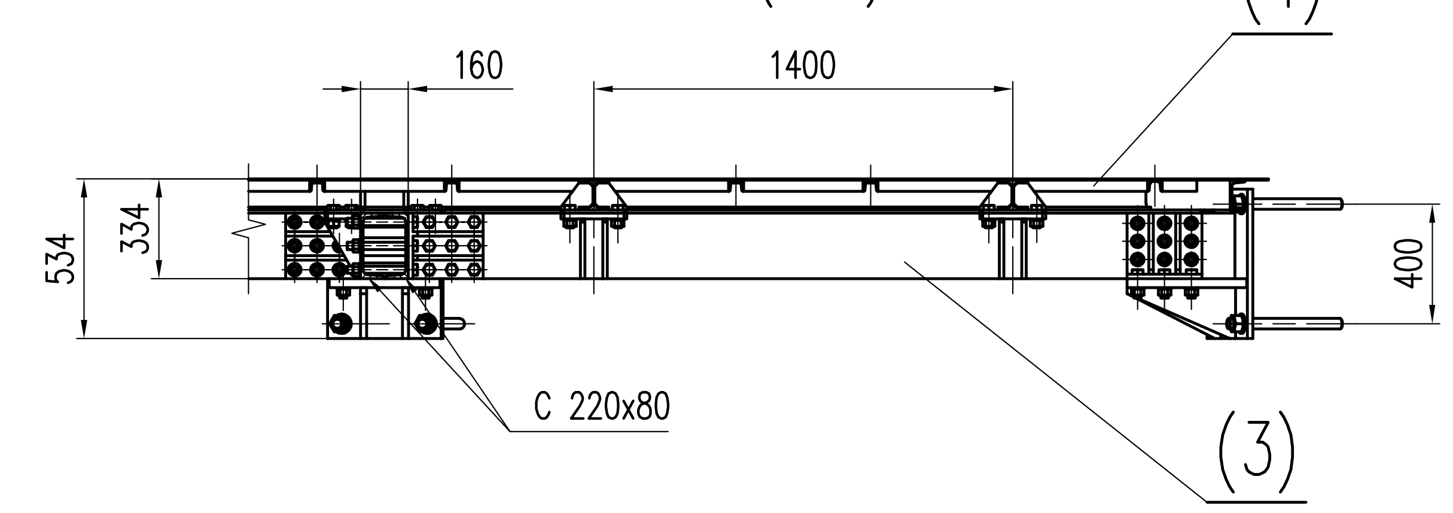
F–F



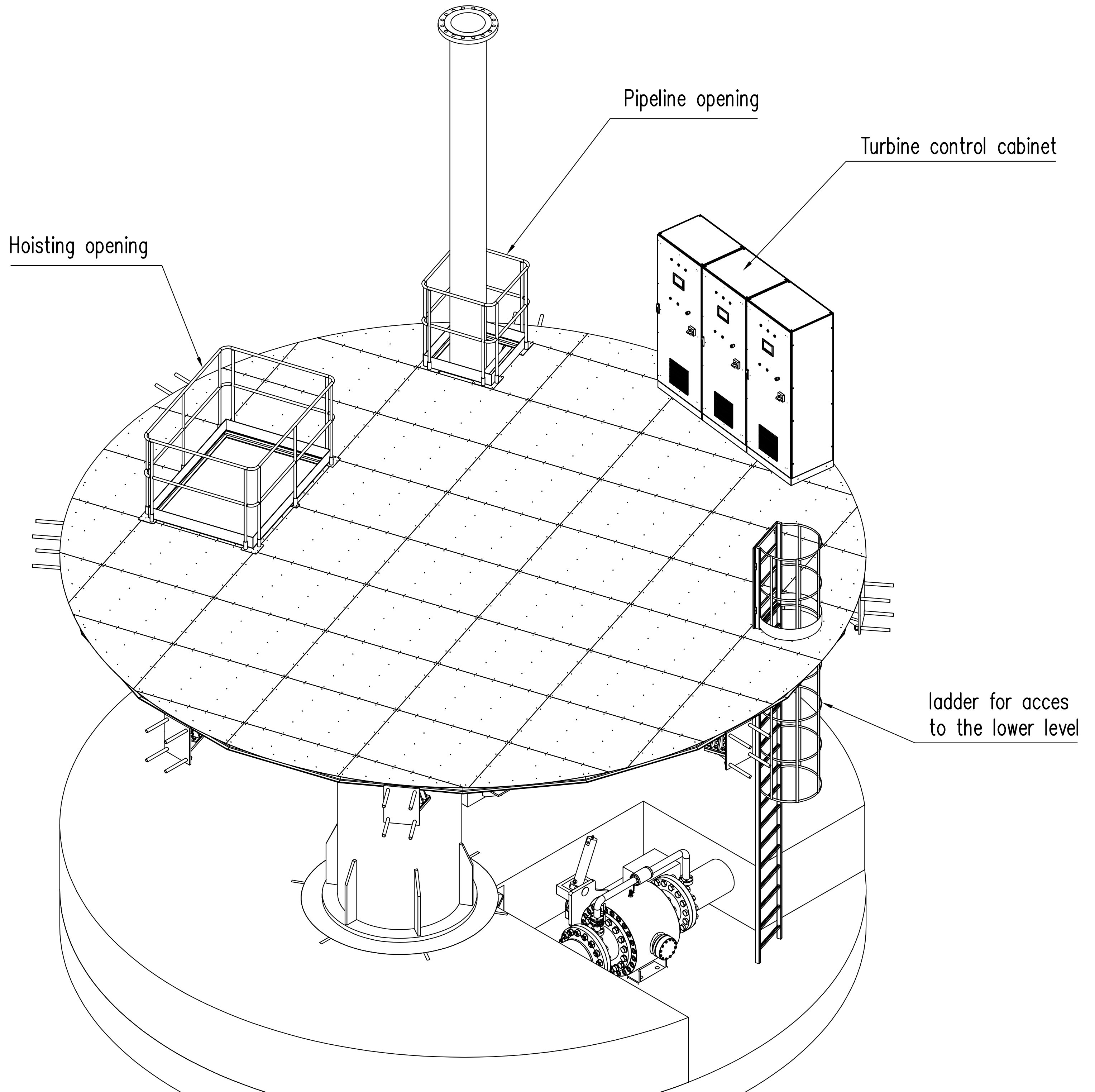
S3 (1:25)



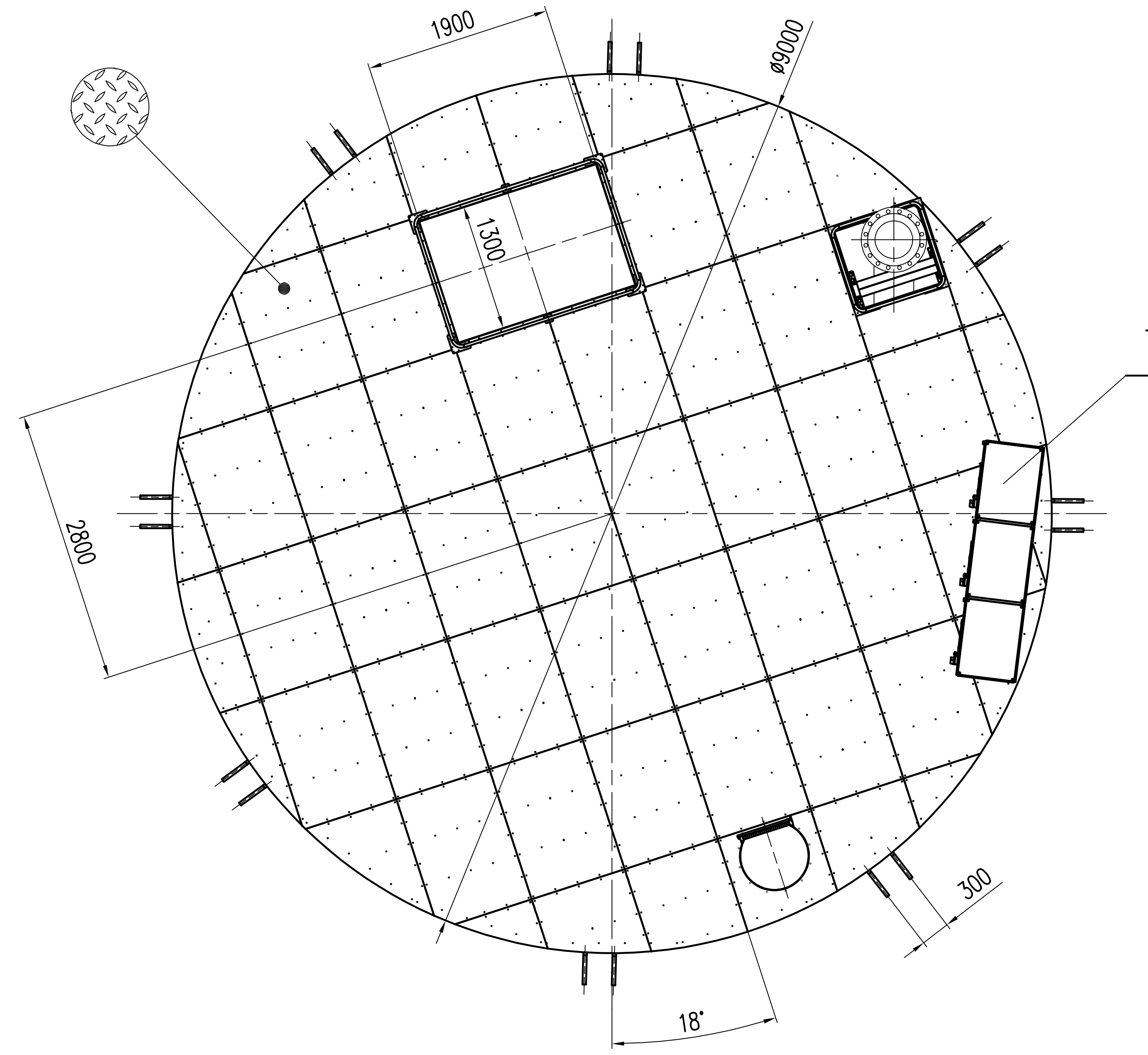
I–I (1:25)



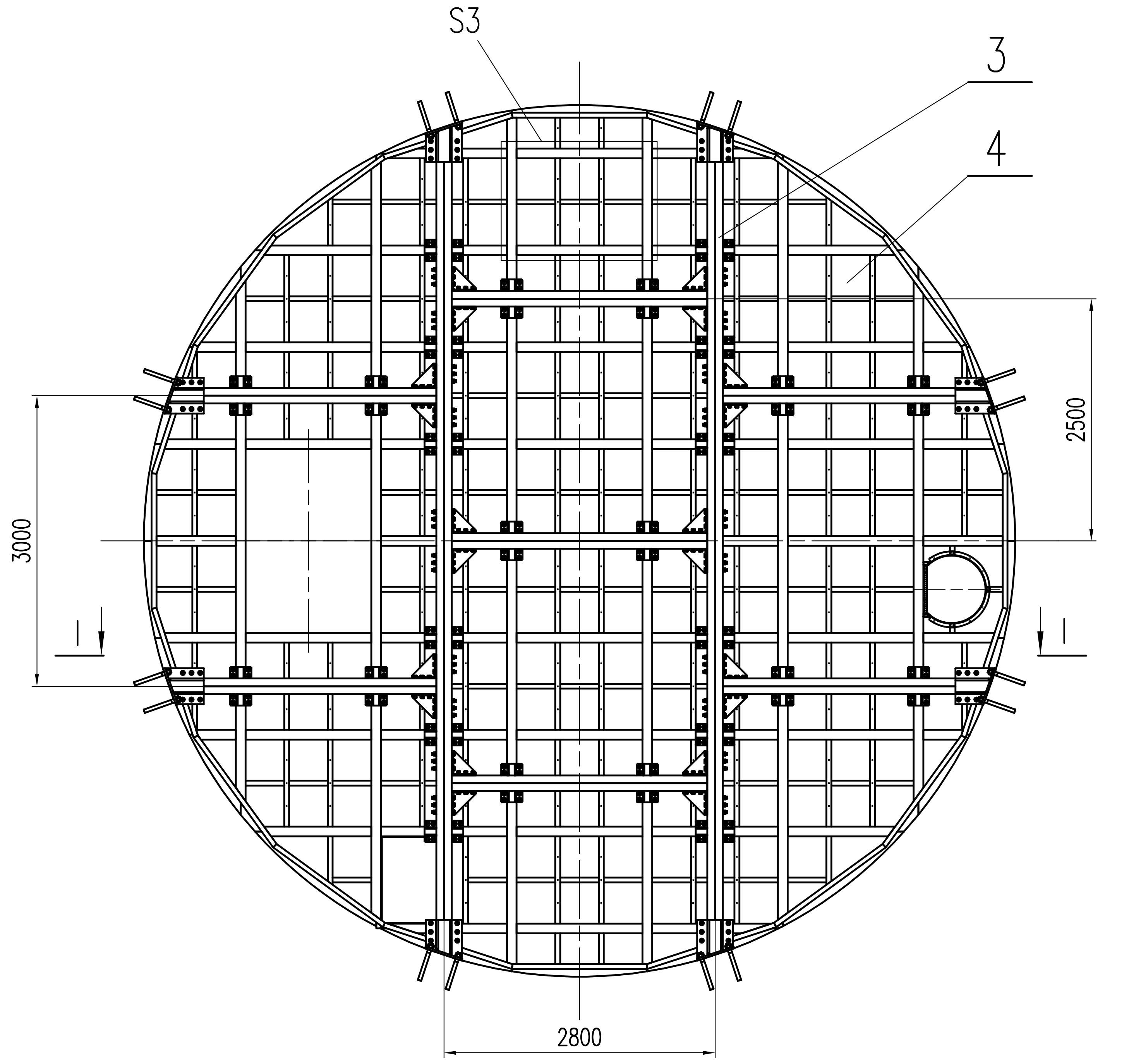
Isometric view



G–G



H–H



INSTITUTE OF MINING TECHNOLOGY **KOMAG**	Task supervisor	BG	P. Matusiak		10.2025
	Designed by	BU	D. Rupik		10.2025
RFCS Project: HESS No.: 101112380	Verified by	BU	K. Kaczmarczyk		10.2025
	Approved by	BG	P. Matusiak		10.2025

Department / Surname / Signature / Date

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No.: W52.010PW04.en

Sheet/number of sheets 2/2

Scale	Mass (kg)	Material
1:50	notes	–

CAD editing: Drawing size A1 | Add No. (or file)

Item name:

Platform assembly

WYKAZ CZĘŚCI / LIST OF COMPONENTS

KOMAG	Obiekt/Object: **Pumped Storage Hydroelectricity**	Nr rysunku/Drawing No: **W52.010PW.en**	Nr wykazu/List No: **W52.010PW-W.en**	
	Zespół/Unit: **Assembly**	RFCS Project: HESS No.: 101112380	*Information included in the list may not be used for any application without the KOMAG Institute written consent*	Ark. / l. ark. Sheet...of... 1/1

Poz./ Item	Szt. na komp./ pcs per unit	Nazwa/Name	Nr rys., norma, katalog/ Dwg. No, standard, catalogue	Masa/Mass (kg)		Materiał/Material	Uwagi/Remarks
				1 szt./ 1 piece	kompletu/set		
1	2	3	4	5a	5b	6	7
1	1	System of the hydro turbine and generator with the shaft reservoir	W52.010PW01.en		265285,0	-	
2	1	Water pumping system	W52.010PW02.en		-	-	
3	1	Upper reservoir	W52.010PW03,en		-	-	
4	1	Platform assembly	W52.010PW04.en		9698,0	-	
					274983,0		The mass refers to the structural part of the turbine–generator unit and the penstock

Opracował/Developed by: R. Baron	Data/Date: 10.2025	Weryfikował/Verified by: K. Kwaśny	Data/Date: 10.2025	Zatwierdził/Approved by: P. Matusiak	Data/Date: 10.2025

WYKAZ CZĘŚCI / LIST OF COMPONENTS

KOMAG	Obiekt/Object: **Pumped Storage Hydroelectricity**	Nr rysunku/Drawing No: **W52.010PW01.en**	Nr wykazu/List No: **W52.010PW-W01.en**	
	Zespół/Unit: **System of the hydro turbine and generator with the shaft reservoir**	RFCS Project: HESS No.: 101112380	*Information included in the list may not be used for any application without the KOMAG Institute written consent*	Ark. / l. ark. Sheet...of... 1/1

Poz./ Item	Szt. na komp./ pcs per unit	Nazwa/Name	Nr rys., norma, katalog/ Dwg. No, standard, catalogue	Masa/Mass (kg)		Materiał/Material	Uwagi/Remarks
				1 szt./ 1 piece	kompletu/set		
1	2	3	4	5a	5b	6	7
1	1	Penstock	W52.010PW01.01.en		245655,0	-	
2	1	Synchronous generator 5800 kW, 6300 V, 1500 rpm with vertical Pelton turbine type PV3i-790-160	VOIGHT Hydro GmbH & Co KG Division Small Hydro Budget offer No.: 021792		14500,0	-	
3	1	Ball valve DN400, PN100			4180,0	-	
4	1	Hydraulic power unit			950,0	-	
					265285,0		

Opracował/Developed by: K. Kwaśny	Data/Date: 08.2025	Weryfikował/Verified by: M. Bal	Data/Date: 08.2025	Zatwierdził/Approved by: P. Matusiak	Data/Date: 08.2025

WYKAZ CZĘŚCI / LIST OF COMPONENTS

KOMAG	Obiekt/Object: **Pumped Storage Hydroelectricity**	Nr rysunku/Drawing No: **W52.010PW01.01.en**	Nr wykazu/List No: **W52.010PW-W01.01.en**
	Zespół/Unit: **Penstock**	RFCS Project: HESS No.: 101112380	*Information included in the list may not be used for any application without the KOMAG Institute written consent* Ark. / l. ark. Sheet...of... 1/2

Poz./ Item	Szt. na komp./ pcs per unit	Nazwa/Name	Nr rys., norma, katalog/ Dwg. No, standard, catalogue	Masa/Mass (kg)		Materiał/Material	Uwagi/Remarks
				1 szt./ 1 piece	kompletu/set		
1	2	3	4	5a	5b	6	7
1	1	Supply pipe DN400, PN100	W52.010PW01.01.en		2260,0	-	
2	3	Support pipe I DN400, PN100	W52.010PW01.01.en	1273,0	3819,0	-	
3	1	Support pipe II DN400, PN100/63	W52.010PW01.01.en		1195,0	-	
4	2	Support pipe III DN400, PN63	W52.010PW01.01.en	1008,0	2016,0	-	
5	1	Support pipe IV DN400, PN63/40	W52.010PW01.01.en		976,0	-	
6	4	Support pipe V DN400, PN40	W52.010PW01.01.en	847,0	3388,0	-	
7	56	Pipe section I DN400, PN100, L=5500mm	W52.010PW01.01.en	1525,0	85400,0	-	
8	39	Pipe section II DN400, PN63, L=5900mm	W52.010PW01.01.en	1198,0	46722,0	-	
9	51	Pipe section III DN400, PN40, L=5900mm	W52.010PW01.01.en	1120,0	57120,0	-	
10	1	Pipe section IV DN400, PN40, L=5000mm	W52.010PW01.01.en		1346,0	-	
11	6	Support structure I	W52.010PW01.01.en	1975,0	11850,0	-	
12	5	Support structure II	W52.010PW01.01.en	1376,0	6880,0	-	
13	55	Guiding support	W52.010PW01.01.en	188,0	10340,0	-	
14	4	Stuffing box expansion joint DN400, PN100	W52.010PW01.01.en	1290,0	5160,0		

Opracował/Developed by: K. Kwaśny	Data/Date: 08.2025	Weryfikował/Verified by: M. Bal	Data/Date: 08.2025	Zatwierdził/Approved by: P. Matusiak	Data/Date: 08.2025

WYKAZ CZĘŚCI / LIST OF COMPONENTS

KOMAG	Obiekt/Object: **Pumped Storage Hydroelectricity**	Nr rysunku/Drawing No: **W52.010PW01.01.en**	Nr wykazu/List No: **W52.010PW-W01.01.en**	
	Zespół/Unit: **Penstock**	RFCS Project: HESS No.: 101112380	*Information included in the list may not be used for any application without the KOMAG Institute written consent*	Ark. / l. ark. Sheet...of... 2/2

Poz./ Item	Szt. na komp./ pcs per unit	Nazwa/Name	Nr rys., norma, katalog/ Dwg. No, standard, catalogue	Masa/Mass (kg)		Materiał/Material	Uwagi/Remarks
				1 szt./ 1 piece	kompletu/set		
1	2	3	4	5a	5b	6	7
15	3	Stuffing box expansion joint DN400, PN63	W52.010PW01.01.en	1097,0	3291,0	-	
16	4	Stuffing box expansion joint DN400, PN40	W52.010PW01.01.en	973,0	3892,0	-	
					245655,0		

Opracował/Developed by: K. Kwaśny	Data/Date: 08.2025	Weryfikował/Verified by: M. Bal	Data/Date: 08.2025	Zatwierdził/Approved by: P. Matusiak	Data/Date: 08.2025

WYKAZ CZĘŚCI / LIST OF COMPONENTS

KOMAG

Obiekt/Object:
Pumped Storage Hydroelectricity

Nr rysunku/Drawing No:
W52.010PW02.en

Nr wykazu/List No:
W52.010PW-W02.en

Zespół/Unit:
Water pumping system

RFCS Project: HESS
No.: 101112380

Information included in the list may not be used for any application without the KOMAG Institute written consent

Ark. / l. ark.
Sheet...of...
1/1

Poz./ Item	Szt. na komp./ pcs per unit	Nazwa/Name	Nr rys., norma, katalog/ Dwg. No, standard, catalogue	Masa/Mass (kg)		Materiał/Material	Uwagi/Remarks
				1 szt./ 1 piece	kompletu/set		
1	2	3	4	5a	5b	6	7
1	1	Existing dewatering pipeline	W52.010PW02.01.en		-	-	

Opracował/Developed by: K. Kaczmarczyk	Data/Date: 10.2025	Weryfikował/Verified by: D. Rupik	Data/Date: 10.2025	Zatwierdził/Approved by: P. Matusiak	Data/Date: 10.2025

WYKAZ CZĘŚCI / LIST OF VOMPONENTS

KOMAG	Obiekt/Object: **Pumped Storage Hydroelectricity**	Nr rysunku/Drawing No.: **W52.010PW04.en**	Nr wykazu/List No.: **W52.010PW-W04.en**
	Zespół/Unit: **Platform assembly**	RFCS Project: HESS No.: 101112380	*Information included in the list may not be used for any application without the KOMAG Institute written consent* — Ark. / l. ark. Sheet…of… 1/1

Poz./ Item	Szt. na komp./pcs per unit	Nazwa/Name	Nr rys., norma, katalog/ Dwg. No., standard, catalogue	Masa/Mass [kg] 1 szt./1 piece	Masa/Mass [kg] kompletu/set	Materiał/Material	Uwagi/Remarks
1	2	3	4	5a	5b	6	7
1	1	Lower platform support frame	W52.010PW04.en		15275,00	-	only occurs in variant I
2	1	Lower platform	W52.010PW04.en		4882,00	-	only occurs in variant I
3	1	Upper platform support frame	W52.010PW04.en		4245,00	-	
4	1	Upper platform	W52.010PW04.en		5175,00	-	
5	1	Hoisting opening guardrail	W52.010PW04.en		68,00	-	
6	1	Ladder	W52.010PW04.en		50,00	-	
7	1	Control assembly	W52.010PW04.en		90,00	-	
8	1	Pipeline opening guardrail	W52.010PW04.en		47,00	-	
9	1	Inspection hatch	W52.010PW04.en		23,00	-	
					29855,00		for varian I mass is 29855 kg/ for variant II mass is 9698 kg

Opracował/Developed by: D. Rupik	Data/Date: 10.2025	Weryfikował/Verified by: K. Kaczmarczyk	Data/Date: 10.2025	Zatwierdził/Approved by: P. Matusiak	Data/Date: 10.2025