HYDRO-POTENTIAL UTILIZATION OF COOLING WATER ON THE HYDRO-ELECTRIC POWER PLANT DALEŠICE

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Engineering solution of a surplus pressure head in a system of reversible machine unit's cooling water. Current technologies supplemented with Francis turbine or more precisely a centrifugal volute-type pump in turbine mode. It contains the layout for the basic extent of several various high-speeds with regard to maximum coverage of working conditions. Minimization of construction works on the structure of the cooling water inlet. Furthermore it includes an assignment of the annual power production.

Keywords: *renewable energy, turbine, cooling water*

1. Introduction

In practice there has been many technologies making use of cooling water with a surplus of a pressure head which is to no avail dissipated in the regulating valve on its beginning or ending. The point is e.g. cooling water inlet for the water power plant technology.

There is a possibility which would utilize the surplus energy effectively and change it into a high-grade and utilizable electric power again. Nowadays is the pressure through the use of throttle valve from the initial level about 8.5 bars regulated on the level of 4 bars.

The purpose of this work is to propose several varieties of hydraulic machines, engine arrangement and approximate estimation of generated energy with respect to minimal necessary investment in subsequent technology of cooling water.

2. Head and discharge of cooling water

For the proposal of proper hydraulic machine is necessary to know the hydrological conditions of the given locality, in this case the Pumped-Storage Power Plant (PSPP) Dalešice. It was resulted from the data provided by submitter, specifically from the technical report of OSC firm [1], handling log of WaterWorks (WW) Dalešice and operating data established in place. Further were used the homologous water turbine characteristics near to required high-speeds. The source parameter for the proposal had been a design spot containing values of design discharge and head. For operational results and also for easier comparison of particular varieties are the design parameters of individual alternates identical.

The design head was determinated from the statistics of upper water level in years 2000–2007 compared with the machine index plane of 287.2 m n.m. spot height. The design section or guaranteed section was chosen from the range of $H_{\max} = 49.14$ m and $H_{\min} = 38.6$ m. The design head was chosen from the range of $H_{\max}$, $H_{\min}$ as the arithmetic average of mean level of daily transfer cycle in the interval of last 8 years. There were

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performed the calculations of hydraulic losses in the newly projected waterway upstream DN300 up to the inlet before the filter of cooling water. The losses amount is 1.1 % from the disposable head on the turbine which presents the value of 0.65 m. In regard of fluctuating back-pressure and the range of heads on the V D is the value of friction head negligible.

The design discharge $Q_{\mathrm{N}} = 0.145\,\mathrm{m}^3\,\mathrm{s}^{-1}$ was established as the mean value of the discharge in the turbine running PSPP from data [2]. Another guaranteed discharge $Q_{\max}$ and $Q_{\min}$ results from technology requirement of filtration the cooling water and were assumed from the technical report [1]. The minimal guaranteed value of cooling water discharge $Q_{\min} = 0.08\,\mathrm{m}^3\,\mathrm{s}^{-1}$ does entirely not reflect the measuring lows of cooling water by reason of regulation limitations of Water Turbine (WT) on the value of $0.55\,Q_{\mathrm{N}}$. All the designed varieties provide the field for regulation of the discharges smaller than $Q_{\min}$, but beyond the limit of the guaranteed range with the appearance of pulsations in the draft tube of VT. The discharge $Q_{\max}$ is the state of opened labyrinth currently with filter washing which occurs only for a short term, so in the proposal it was not given such a big weight as to the operational spot N.

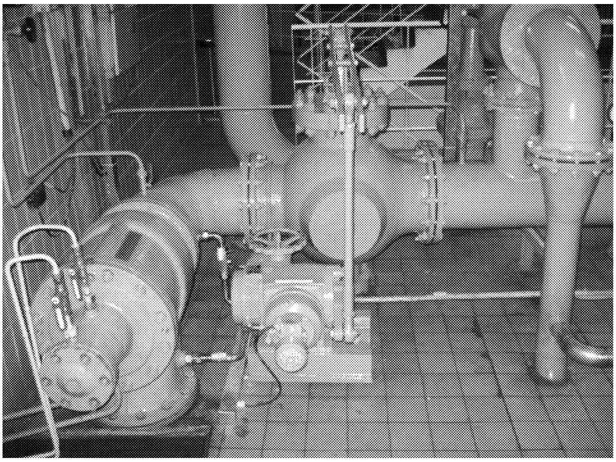

Fig.1

3. Operating area description

The design point N represents the average head and flow values that correspond to the standard operation of the turbo generator. Point A and B represents the head changes in the water power plant perhaps even the pressure rate changes in the cooling system for the flow value Q_{N}. By the border line points C–F there are defined the rest of working regimes. It means the transient conditions as filter and labyrinth washing. These regimes are in comparison with points A, B, N not more then 2 % of the whole operation time.

4. Design variations

4.1. Variation I. F90

For the Francis turbine (FT) design was used the model characteristic signed as F90, see Fig. 2. The design was done due regard for the present control valve function and optimized in light of produced energy.

Q_N [$m^3 s^{-1}$]	Q_{max} [$m^3 s^{-1}$]	Q_{min} [$m^3 s^{-1}$]	H_N [m]	H_{max} [m]	H_{min} [m]
0.145	0.208	0.087	45.00	49.14	38.60

Tab.1

Turbine model parameters:

Unit flow	$Q_{11\mathrm{opt}} = 0.151\,\mathrm{m^3\,s^{-1}}$,
Unit revolutions	$n_{11\mathrm{opt}} = 73\,\mathrm{min}^{-1}$,
Runner model diameter	$\varnothing D_\mathrm{M} = 0.382\,\mathrm{m}$,
Optimal efficiency	$\eta_\mathrm{opt} = 83.8\,\% = \eta_\mathrm{M}$.

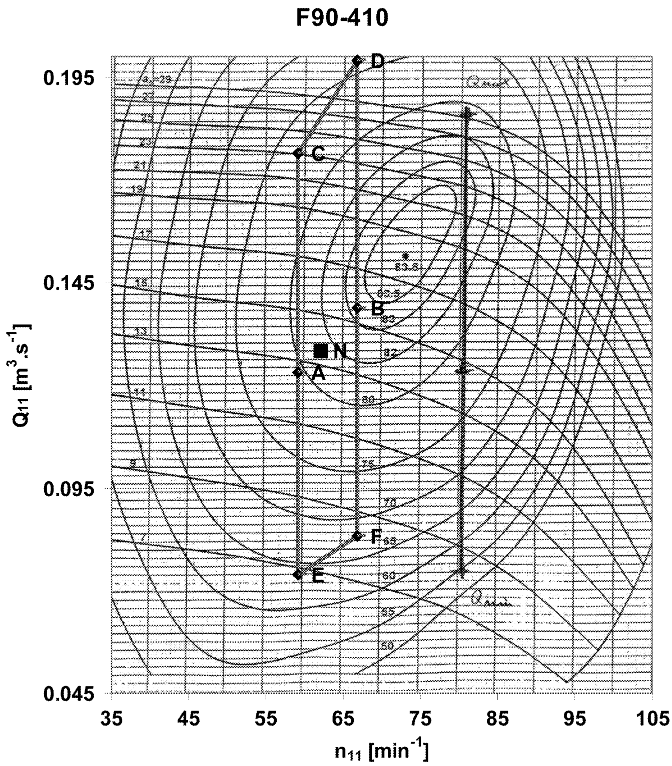

Fig.2

The FT runner diameter evaluation goes out of the design point value, see Tab. 1:

$$Q_{11\mathrm{N}} = \frac{Q_\mathrm{N}}{D^2\sqrt{H_\mathrm{N}}} \quad\Rightarrow\quad D = \sqrt{\frac{Q_\mathrm{N}}{Q_{11\mathrm{N}}\sqrt{H_\mathrm{N}}}}\ , \tag{1}$$

$$n_{11\mathrm{N}} = \frac{n\,D}{\sqrt{H_\mathrm{N}}} \quad\Rightarrow\quad n = \frac{n_{11\mathrm{N}}\sqrt{H_\mathrm{N}}}{D}\ . \tag{2}$$

The FT hydraulic efficiency recalculation from the model to the prototype according to Moody:

$$\eta_\mathrm{T} = 1 - (1-\eta_\mathrm{M})\left(\frac{D_\mathrm{M}}{D_\mathrm{T}}\right)^{0.2}\ . \tag{3}$$

The turbine delivery evaluation:

$$P_{\mathrm{T}i} = \varrho\, Q_i\, g\, H_i\, \eta_{\mathrm{T}i}\ . \tag{4}$$

Produced energy:

$$E = P_{\mathrm{G}}\, T_i\, \Delta t\ , \tag{5}$$

where T_i working time ratio, Δt av. annual operating time Turbine-Generator (TG) ($\Delta t = 1912$ hrs/year).

F90 – 410	
$\varnothing D$ [mm]	410
n [min^{-1}]	1013

Tab.2

Point	H [m]	Q [m^3 s^{-1}]	Q_{11} [m^3 s^{-1}]	n_{11} [min^{-1}]	η_{M} [%]	η_{T} [%]	P_{T} [kW]	η_{G} [%]	P_{G} [kW]	T [–]	E [MWh]
N	45.00	0.145	0.1286	61.92	81.1	81.4	52.07	95.2	49.55	0.50	47.362
A	49.14	0.145	0.1231	59.25	79.5	79.8	55.75	95.4	53.17	0.25	25.413
B	38.60	0.145	0.1388	66.85	83.2	83.5	45.83	94.6	43.35	0.25	20.719
C	49.14	0.208	0.1765	59.25	77.4	77.7	77.87	95.2	74.11	0.005	0.708
D	38.60	0.208	0.1992	66.85	73.4	73.7	58.02	95.5	55.39	0.005	0.529
E	49.14	0.087	0.0738	59.25	63.8	64.1	26.87	91.6	24.61	0.005	0.235
F	38.60	0.087	0.0833	66.85	67.5	67.8	22.32	90.7	20.25	0.005	0.194
Total produced energy E_{C}											95.2

Tab.3

The direct connection with the three phase generator with squirrel cage motor is proposed. The asynchronous generator parameters are given in the Table 4.

Type	power [kW]	n_{s}	n_{as}	n. curr. [A]	n. volt. [V]	η_{G} [%]	I [kg m^2]	m [kg]
1LG6 310-6AA	75	1000	1013	138	400	95.0	2.5	760

Tab.4

4.2. Variation II. F100

For the Francis turbine (FT) design was used the model characteristic signed as F100A, see Fig. 3. The design was done due regard for the present control valve function and optimized in light of produced energy.

Q_N [m^3 s^{-1}]	$Q_{\max}$ [m^3 s^{-1}]	$Q_{\min}$ [m^3 s^{-1}]	H_{N} [m]	$H_{\max}$ [m]	$H_{\min}$ [m]
0.145	0.220	0.087	45.00	49.14	38.60

Tab.5

Turbine model parameters:

Unit flow	$Q_{11\mathrm{opt}} = 0.205\,\mathrm{m^3\,s^{-1}}$,
Unit revolutions	$n_{11\mathrm{opt}} = 66\,\mathrm{min^{-1}}$,
Runner model diameter	$\varnothing D_\mathrm{M} = 0.4\,\mathrm{m}$,
Optimal efficiency	$\eta_\mathrm{opt} = 91.1\,\% = \eta_\mathrm{M}$.

F100A – 370	
$\varnothing D$ [mm]	370
n [min^{-1}]	1013

Tab.6

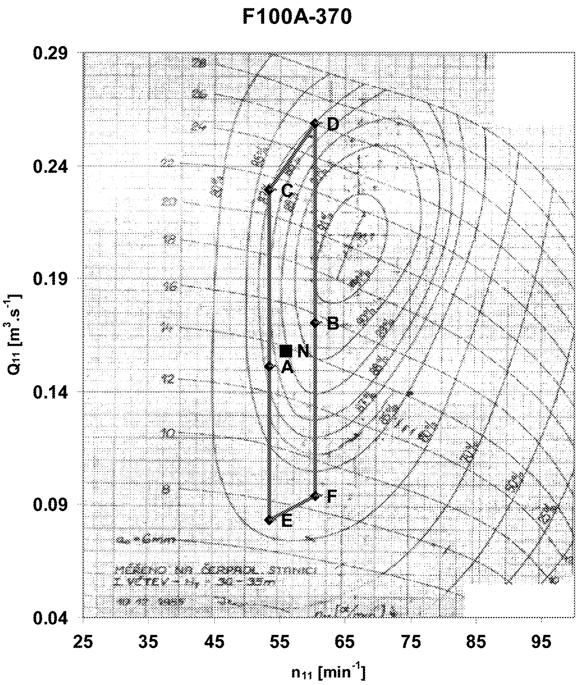

Fig.3

Point	H [m]	Q [m^3 s^{-1}]	Q_{11} [m^3 s^{-1}]	n_{11} [min^{-1}]	η_M [%]	η_T [%]	P_T [kW]	η_G [%]	P_G [kW]	T [–]	E [MWh]
N	45.00	0.145	0.1579	55.88	88.9	88.7	56.77	95.4	54.17	0.50	51.776
A	49.14	0.145	0.1511	53.47	87.5	87.3	61.02	95.5	58.30	0.25	27.864
B	38.60	0.145	0.1705	60.33	90.5	90.3	49.58	95.0	47.09	0.25	22.504
C	49.14	0.220	0.2292	53.47	87.0	86.8	92.05	94.5	87.02	0.005	0.832
D	38.60	0.220	0.2587	60.33	87.1	86.9	72.39	95.4	69.07	0.005	0.660
E	49.14	0.080	0.0831	53.47	81.0	80.8	31.06	92.4	28.71	0.005	0.274
F	38.60	0.080	0.0938	60.33	82.8	82.6	24.94	91.2	22.76	0.005	0.218
Total produced energy E_C											104.1

Tab.7

The direct connection with the three phase generator with squirrel cage motor is proposed. The asynchronous generator parameters are given in the Table 4.

4.3. Variation III. F130

For the Francis turbine (FT) design was used the model characteristic signed as F130, see Fig. 4. The design was done due regard for the present control valve function and optimized in light of produced energy.

Q_N [$\mathrm{m^3\,s^{-1}}$]	$Q_{\max}$ [$\mathrm{m^3\,s^{-1}}$]	$Q_{\min}$ [$\mathrm{m^3\,s^{-1}}$]	H_{N} [m]	$H_{\max}$ [m]	$H_{\min}$ [m]
0.145	0.220	0.087	45.00	49.14	38.60

Tab.8

Turbine model parameters:

Unit flow $Q_{11\mathrm{opt}} = 0.380\,\mathrm{m^3\,s^{-1}}$,
Unit revolutions $n_{11\mathrm{opt}} = 65\,\mathrm{min^{-1}}$,
Runner model diameter $\varnothing D_{\mathrm{M}} = 1.0\,\mathrm{m}$,
Optimal efficiency $\eta_{\mathrm{opt}} = 89.5\,\% = \eta_{\mathrm{M}}$.

F130 – 255	
$\varnothing D$ [mm]	255
n [$\mathrm{min^{-1}}$]	1520

Tab.9

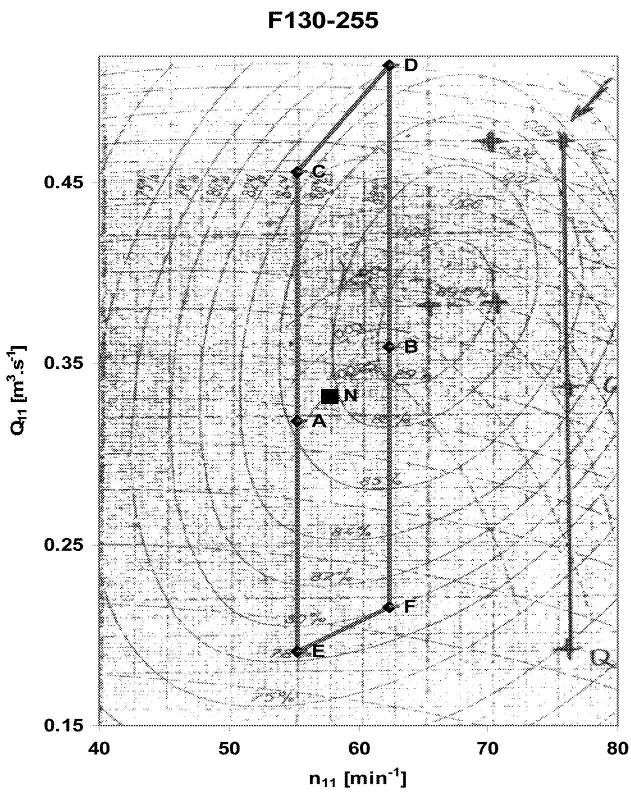

Fig.4

Point	H [m]	Q [$m^3 s^{-1}$]	Q_{11} [$m^3 s^{-1}$]	n_{11} [min^{-1}]	η_M [%]	η_T [%]	P_T [kW]	η_G [%]	P_G [kW]	T [–]	E [MWh]
N	45.00	0.145	0.3328	57.79	87.8	83.9	53.69	95.3	51.15	0.50	48.894
A	49.14	0.145	0.3184	55.29	86.2	82.3	57.53	95.5	54.91	0.25	26.243
B	38.60	0.145	0.3594	62.39	89.1	85.2	46.78	94.7	44.30	0.25	21.174
C	49.14	0.220	0.4568	55.29	83.2	79.3	79.51	95.1	75.61	0.005	0.723
D	38.60	0.220	0.5155	62.39	82.0	78.1	61.51	95.6	58.78	0.005	0.562
E	49.14	0.080	0.1911	55.29	78.3	74.4	31.20	92.4	28.84	0.005	0.276
F	38.60	0.080	0.2156	62.39	79.9	76.0	25.04	91.3	22.85	0.005	0.218
Total produced energy E_C											98.1

Tab.10

Type	power [kW]	n_s	n_{as}	n. curr. [A]	n. volt. [V]	η_G [%]	I [$kg\,m^2$]	m [kg]
1LG6 280-4AA	75	1500	1520	130	400	95.0	1.39	575

Tab.11

The direct connection with the three phase generator with squirrel cage motor is proposed. The asynchronous generator parameters are given in the Table 11.

4.4. Variation IV. Pump-Turbine (PT) (spiral one-stage pump)

For the primary calculation it holds:

$$Q_{\check{C}} = \eta\, Q_T \ , \tag{6}$$

$$H_{\check{C}} = \eta^2\, H_T \ . \tag{7}$$

The turbine (pump) revolutions are assumed as $1520\,min^{-1}$ respectively $1450\,min^{-1}$.

The index Č means the flow and head of the machine in the pump regime; the index T means the flow and head of the machine in the turbine regime.

For the design discharge $Q_T = 145\,l/s$ and head $H_T = 45\,m$, after the specific speed determination we get the value: $n_s = 122\,min^{-1}$.

$$n_s = 3.65\, \frac{n}{\sqrt{H}} \sqrt{\frac{Q}{\sqrt{H}}} \ . \tag{8}$$

This is the runner for that is possible to guess the hydraulic efficiency $\eta = 84.5\,\%$. That's the reason why the impeller parameters must meet the requirements:

$$H_{\check{C}} = 32.13\,m \qquad \text{and} \qquad Q_{\check{C}} = 122.5\,l\,s^{-1} \ .$$

For these parameters reaches the impeller the specific revolutions $n_s = 137.3\,min^{-1}$. That is the primary presumption commonly used for the one-stage spiral pump in the turbine regime [3].

From the one-stage pump catalog are the best matching models: Beta 36 and Meta 38.

The more accurate design discharge and head setting is done by the gradually impeller diameter reducing.

5. Turbine arrangement

It was determined, out of the measurement analysis [1] of the present state of the cooling water upstream waterway characteristic, that there is no disposing head for the water turbine installation. That was the reason for the parallel connection to the current cooling water technology selection. The newly projected branch is connected to the existing waterway DN300 in the place before the slide valve DN300. The vertical configuration was used because of the limiting spatial disposition, save service and minimized construction works.

Because of the planned installation of the machine set also for the rest of TG was the vertical configuration more convenient because of the smaller floor space occupation, Fig. 5 and 6. The horizontal configuration of the water turbine + generator needs approximately 2×1 m of area, that is not available for the rest of TG. The existing branch conservation with the control valve will subserve a security margin function for the Francis turbine failure.

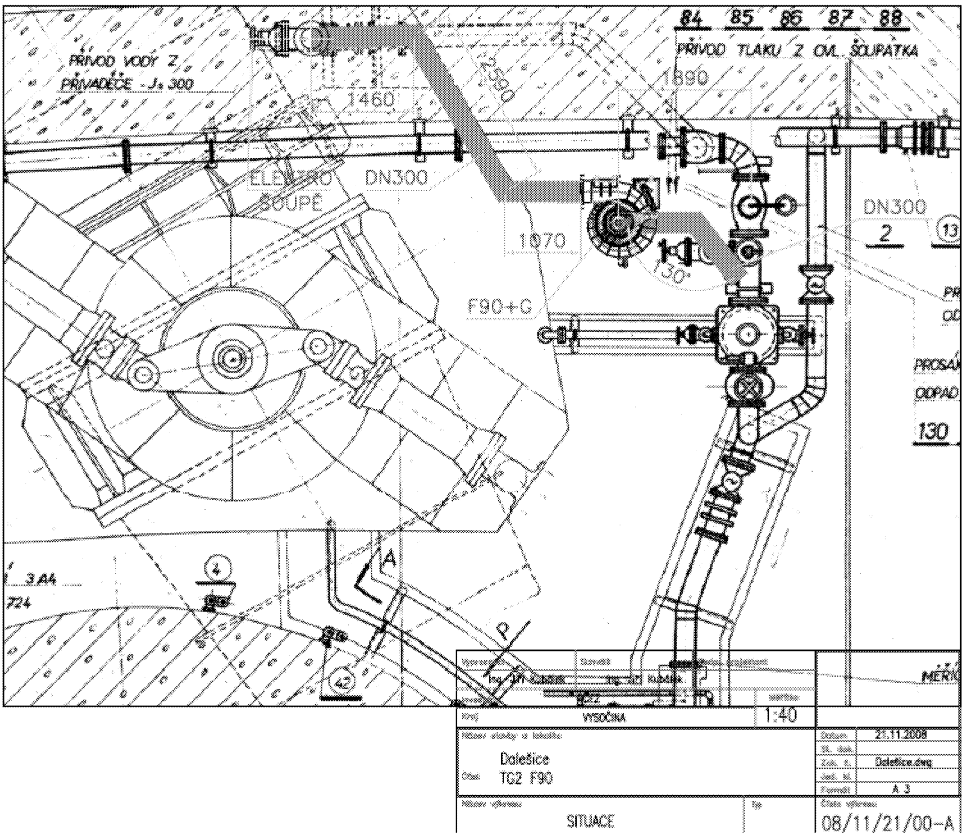

Fig.5

6. Conclusion

Out of the above mentioned variations follows that the potential utilization better fits the F100A. The operating points A, B, N are placed in the high hydraulic efficiency area between 87 % and 90 % in comparison with the rest variant. At the same time it provides sufficient space for the smaller heads regulation where it is close to the efficiency optimum. On the

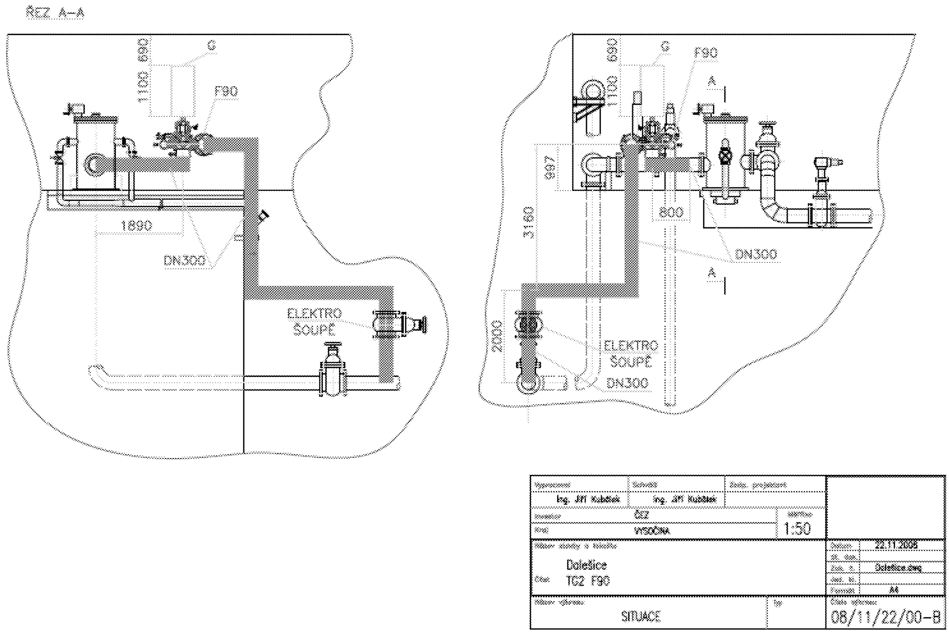

Fig.6

contrary the usable head growth and accompanying lower unit revolutions regulation is under possibilities. The F100A variant has the highest produced energy value $E_\mathrm{C} = 104\,\mathrm{MWh}$ for the model production distribution. It also offers the possibility of the machine set using out of the main TG cooperation.

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