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High-Performance Power Generation Equipment for the 21st Century

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❊ Electrostatic and Magnetic Induction Disk Dynamos

AESI Business Plan *Appendix B:*

Proof Sheet for Thermal Delivery Capacity of StarDrive Dynamo Utility Plants

This proof sheet is intended to confirm the quoted thermal delivery capacities of large-scale liquid-cooled "StarDrive Dynamo" utility plants, based on a detailed coolant flow rate analysis for the 20-ft. 256 MW EDF Generator and assuming an actual primary conduit inside diameter which is exactly ***1.888 times*** the minimum conduit inside diameter specification given in the Company's technical manual *StarDrive Engineering.* Since the device is linearly scalable, and the primary conduit cross section will increase by the square of linear increases in Dynamo diameter, it is therefore assured that the thermal delivery capacity of all the planned StarDrive utility plant models will be uniformly proportional to the square of changes in the diameter of the respective Dynamos relative to the said 20-ft. diameter unit with which this study is concerned.

It must be noted that the stated increase in primary conduit inside diameter is well within technical feasibility according to the design. The thermal efficiency of the liquid sodium coolant loops will be assumed to equal 85%. Thus, the net thermal delivery capacities quoted represent 85% of the theoretical maximums achievable at 100% thermal efficiency, considered reasonable overall in this preliminary "ideal" flow analysis. It is of course presumed that the multistage flash distillation system used in each case will function according to the manufacturer's specifications.

Each regional-scale Dynamo is rated in multiples of 60 MW, so the theoretical design baseline used in this analysis is a Dynamo model having a net rating of 30 MW. The pertinent values for sodium specific heat, density, and dynamic viscosity are taken from the Argonne National Laboratory study *Thermodynamic and Transport Properties of Sodium Liquid and Vapor* (1995) by J. K. Fink and L. Leibowitz.

Part 1: Calculate the total coolant mass flow (G) required to deliver the thermal equivalent of 30 MW to the primary heat exchanger, at 85% efficiency, given a mean coolant temperature of 500 °C and a working coolant temperature differential of $\Delta T = 250\,^{\circ}C$.

a) The liquid sodium coolant system taken at 100% efficiency must deliver 30,000,000 joules of thermal energy per second. Since 1 joule = 0.239 calories, 30 MW = 7.17×10^6 cal/sec.

b) If the thermal input system is 85% efficient overall, the coolant must be pumped through the primary heat exchanger at a rate equivalent to 35.294 MW in practice, or 8.4353×10^6 cal/sec.

c) Given the exchanger ΔT of 250 °C and a specific heat for sodium of 0.302 cal/g/°C at 500 °C, the required mass flow is:

$$G = \frac{8.4353 \times 10^6 \text{ cal/sec}}{250^{\circ}\text{C}\ (0.302 \text{ cal/g/}^{\circ}\text{C})} = 111.73 \text{ kg/sec.}$$

With a total of 72 primary conduits, the individual conduit mass flow $G_i = 1.552$ kg/sec.

In the 240 MW Dynamo plant it is assumed that a mass flow $G_i = (8 \times 1.552) = 12.416$ kg/sec is therefore required in each primary conduit. This is equal to 235.8 gallons per minute (gpm) per individual conduit, and to 16,977 gpm overall.

Part 2: Apply the coolant mass flow (G) value obtained above to complete flow calculations for the 240 MW Dynamo plant, using idealized specifications for the pressure and diameter of the mainline, 6 manifolds, and 36 conduits of each primary coolant loop.

a) A brief but complete description of the simplified theoretical basis for the flow rate and pressure differential computations to be undertaken follows below:

(i) The formula for incompressible fluid flow through a Venturi tube, based on the Bernoulli principle, is:

$$\frac{P_1}{\rho} + \frac{V_1^2}{2} + g z_1 = \frac{P_2}{\rho} + \frac{V_2^2}{2} + g z_2 + \frac{\Delta P_{1\text{-}2}}{\rho}$$

, where P = pressure,
ρ = fluid density,
V = flow velocity,
g = accel. due to gravity,
and z = geodetic height.

(ii) Assuming the pressure lost is negligible ($\Delta P_{1\text{-}2} = 0$), and that $g z_1 = g z_2$, then the velocities in the equation above may be used to define a *laminar* volumetric flow rate Q as follows:

$$Q = \pi D_1^2 V_1 / 4, \text{ and } Q = \pi D_2^2 V_2 / 4,$$

where D_1 = inlet pipe inside diameter,
and D_2 = discharge pipe inside dia.

(iii) It should be noted that these are simple flow cross section relationships wherein we have elected to input pipe diameters (instead of radii) for expediency. Solving the above equations for the respective velocities and substituting into the simplified Bernoulli formula yields:

$$\frac{2(P_1 - P_2)}{\rho} = \frac{16Q^2}{\pi^2}\left(\frac{1}{D_2^4} - \frac{1}{D_1^4}\right)$$

where $(P_1 - P_2)$ is the pressure drop throught the Venturi due to flow velocity increase.

(iv) We may then solve for an unknown flow rate Q through the Venturi junction, given only the two different pipe diameters, the pressure differential, and the fluid density, as follows:

$$Q = CE\,\frac{\pi D_2^4}{4}\sqrt{\frac{2(P_1 - P_2)}{\rho}} \quad , \text{ where } E = \sqrt{\frac{1}{1 - (D_2/D_1)^4}}$$

and C is the *coefficient of discharge* (typically 0.985 for $D_2/D_1 = 0.4$ to 0.7).

Given the flow rate thus calculated, the thermal mass flow is simply $G = \rho Q$, and the two flow velocities are $V_1 = 4Q/\pi D_1$, and $V_2 = 4Q/\pi D_2$. By selecting suitable mainline-to-manifold and manifold-to-conduit junction diameter ratios, and realistic pressure levels for each of the three different required piping diameters, it is desired to develop a fluid flow regime that is both laminar in nature and roughly linearly scalable for all sizes of StarDrive Dynamo utility plants in a range from 60 MW to 720 MW in thermal output capacity.

b) As stated at the end of Part 1 above, it is assumed that in the 240 MW Dynamo utility plant a mass flow G_i = (8 x 1.552) = 12.416 kg/sec is required in each primary conduit. To perform the calculations required to derive the following pressures and flow velocities, a mean density of 834.5 g/l at 500 °C was used for the liquid sodium coolant. It should be noted again that each of two (2) coolant mainlines feed six (6) manifolds, which in turn each serve six (6) conduits.

(i) The following values for piping diameters and pressures were used in this example:

- ► D1 = mainline ID = 6.8″ ; D2 = manifold ID = 2.85″ ; D3 = conduit ID = 1.18″.

  D2 / D1 = 0.419, so the *coefficient of discharge* C is approximately 0.985.

  D3 / D2 = 0.414, so the *coefficient of discharge* C is also approximately 0.985.

- ► P1 = mainline psi = 70.3 ; P2 = manifold psi = 41.7 ; P3 = conduit psi = 14.7.

(ii) For each manifold-to-conduit junction, the resulting primary conduit flow velocity derived was V3 = 21.12 m/sec, with a mass flow G_3 = 12.434 kg/sec., which is indeed marginally greater than that required. The corresponding single-conduit manifold flow velocity was 3.62 m/sec. *If we assume that the resultant total manifold flow velocity with six (6) conduits per manifold will be 6 times the single-conduit velocity*, the total manifold flow velocity V2 = 21.72 m/sec, which is also marginally greater than that of the individual primary conduits.

(iii) For each mainline-to-manifold junction, the resulting manifold flow velocity derived was V2 = 21.75 m/sec, with a coolant mass flow G_2 = 74.708 kg/sec., which is once again marginally greater than a required minimum value of 6 x 12.434 = 74.604 kg/sec. The corresponding single-manifold mainline flow velocity was 3.82 m/sec. *If we assume that the resultant total mainline flow velocity with six (6) manifolds per mainline will be 6 times the single-manifold velocity*, the total mainline flow velocity V1 = 22.92 m/sec, which again is marginally greater than that of the individual manifolds. The resulting mainline mass flow G_1 = 6 x 74.604 kg/sec, or 447.62 kg/sec. Since the total base coolant mass flow required in this "ideal" calculation is 72 x 12.416 kg/sec., or 893.95 95 kg/sec., and the calculated total mainline mass flow of $2G_1$ = 895.24 kg/sec., it can be seen that the complex coolant system as designed and analyzed above is adequate 'ideally' to provide a net thermal equivalent of 240 MW to the primary heat exchanger contemplated.

(iv) It is nevertheless necessary to calculate the Reynolds number associated with the fluid flow system described to verify that the flow characteristics therein will be laminar (non-turbulent). So, if Reynolds number $R = DV\rho/\mu$, where μ = dynamic viscosity = 2.37 x 10^4 Pa·s @ 500 °C:

with conduit flow V3 = 21.12 m/sec, the flow is laminar at R = 0.0223;

with manifold flow V2 = 21.75 m/sec, the flow is laminar at R = 0.0554;

and with mainline flow V1 = 22.92 m/sec, the flow is laminar at R = 0.1394.

Knowing that the StarDrive Dynamo mainline, manifold, and conduit flow cross sections will be reduced or increased uniformly by the square of changes in the Dynamo's design housing radius, it can readily be shown that *the system defined and modelled above is linearly scalable in the range of utility plant sizes from 60 MW to 720 MW in thermal output capacity (as intended) with only very minor adjustments to pressure or proportional piping diameter(s)*.