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FINAL REPORT

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ABSTRACT

The goal of this project is to create a heat exchanger that can effectively remove heat from a nuclear power plant's reactor core. Transferring 840 MWt of energy from the nuclear reactor to the turbine is our goal. When building this application, a number of elements were taken into account, and nuclear power plants were investigated. Heat exchanger types employed in industrial heat processes were analysed as a result of literature searches, and shell and tube heat exchangers were chosen. First, 316 stainless steel was selected as the material choice and then the design was made by changing the tube arrangements in two different configurations. The diameter length was chosen as the parameter to be altered on these designs in order to compare them. Following the recording of these comparisons in a table, a choice was taken by weighing the benefits and drawbacks, and the results were thoroughly discussed in the conclusion section.

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NOMENCLATURE

T: Temperature (K)

Cp: Specific heat (kj/kg K)

m: kilogram (kg)

ṁ: Mass flow rate (kg/h)

k: Thermal conductivity (W/m K)

q: Heat transfer rate (kj/s)

V: velocity (m/s)

C: Thermal conductance (kj/kg s)

A: Cross sectional area (m2)

D: Diameter (m)

h: Convection coefficient (W/m K)

CL: Tube layout constant

CTP: Tube count calculation constant

Pt: Pitch ratio

B: Baffle (m)

C: Clearance (m)

L: Length (m)

ΔP: Pressure (bar)

ε: effectiveness

ρ: density (kg/m3)

μ: viscosity (Ns/m2)

1. INRTODUCTION

The modern world will need power that is both efficient and plentiful in the years to come. As industry and people's standard of life continue to grow, conventional power sources like coal, oil, and gas are coming under pressure. The capacity of these sources will soon be exceeded by the world's increasing demand. The usage of nuclear energy is unavoidable in a nation when other power generating alternatives are inadequate. One of the most amazing things about nuclear power is the maximum amount of energy that can be liberated from a little mass of active materials. A nuclear plant consists of a nuclear reactor, heat exchanger, feed pump, condenser, turbine, generator, and coolant circulating pump. In a reactor, heat is generated by the fission process. The primary circuit's coolant is heated before it enters the heat exchanger by managing the heat. Here, the heat exchanger's input water is heated by the hot coolant via heat transfer to create steam. Steam from the heat exchanger will enter the turbine to power the turbine blades and generate energy. The feed pump returns the water to the heat exchanger after the steam has finished its transformation in the condenser. After cooling in the heat exchanger, the hot coolant is recirculated into the reactor via the coolant circulation pump. The cycle is continuously performed to produce power. Furthermore, it is obvious that a growing population leads to a significant rise in the need for energy. Therefore, energy efficiency is a critical issue in the modern world. The heat exchanger designed in this project will be used to remove heat from a nuclear power plant's reactor core. Moreover, there are many different types of power plant reactors.

In this project, the sodium-cooled fast reactor (SFR) was chosen as a result of comparisons.

Sodium has great transport and heat transfer capabilities at high temperatures. High heat release rates, or high-power densities as reactors are known, are made possible by several characteristics.

A high-power density and the benefit of low-pressure operation are made possible by the sodium-cooled fast reactor (SFR), which operates in the fast neutron spectrum and uses liquid sodium as a coolant.

High power density and low coolant volume percentage are made possible by the SFR's use of liquid sodium as the reactor coolant. Although corrosion is prevented by the oxygen-free atmosphere, a sealed coolant system is necessary due to sodium's chemical reactivity with air and water.

To sum up, the required mathematical procedures were followed in order to design the sodium-cooled fast reactor (SFR) a Shell and Tube Heat Exchanger with a high energy efficiency that satisfies effective heat transfer adapted to the particular needs of the application. This was done by optimizing the parameters of tube diameter and number of tubes.

2. THEORY

In this section, the equations used in the calculations of the report are listed below.

Mean film temperature,

$$T_m = \frac{T_i + T_o}{2} \tag{1}$$

Specific heat capacity,

$$\frac{C_p}{M} = a + bT + cT^2 + dT^3 \tag{2}$$

Energy balance,

$$E_{\text{in}} - E_{\text{out}} + E_g = E_{\text{st}} \tag{3}$$

Heat transfer rate,

$$q = \dot{m} * C_p * \Delta T \tag{4}$$

$$q = L * \pi * D_o * U * \Delta T_{lm} * N_t \tag{5}$$

$$q = \dot{m} * C_p * \Delta T + hfg * m \tag{6}$$

Maximum possible heat rate,

$$\mathrm{C} = \dot{m} * Cp \tag{7}$$

$$\mathrm{q} = C * (T_i - T_o) \tag{8}$$

$$q_{max} = C_{min} * (T_{h,i} - T_{c,i}) \tag{9}$$

Effectiveness,

$$\varepsilon = \frac{q}{q_{max}} \tag{10}$$

Heat capacity ratio,

$$C_r = \frac{C_{min}}{C_{max}} \tag{11}$$

NTU,

$$NTU = \frac{-1}{Cr - 1} * \ln\left(\frac{\varepsilon - 1}{\varepsilon * Cr - 1}\right) \qquad Cr < 1\ (Counterflow) \tag{12}$$

$$E = \frac{\frac{2}{\varepsilon} - (1 + C_r)}{(1 + C_r^2)^{0.5}} \tag{13}$$

$$NTU = \frac{U * A_s}{C_{min}} \tag{14}$$

Log mean temperature difference,

$$\Delta T_{lm} = \frac{(T_{h,o} - T_{c,i}) - (T_{h,i} - T_{c,o})}{\ln(\frac{(T_{h,o} - T_{c,i})}{(T_{h,i} - T_{c,o})})} \tag{15}$$

Cross-sectional area of one tube,

$$A_t = \frac{\pi * D_i^2}{4} \tag{16}$$

Velocity,

$$V = \frac{\dot{m}}{\rho A} \tag{17}$$

Reynold's Number,

$$Re_D = \frac{\rho * V * D}{\mu} \tag{18}$$

Nusselt Number,

$$Nu_D = 5 + 0.025 * {Re_D}^{0.8} \, Pr^{0.4} \tag{19}$$

$$Nu_D = C_1 * C_2 * Re_D^m * \mathrm{Pr}^{0.36} * (\frac{\mathrm{Pr}}{P_s})^{\frac{1}{2}} \tag{20}$$

$$Nu_D = \frac{h * D}{k} \tag{21}$$

Overall heat transfer coefficient,

$$\frac{1}{U} = \frac{1}{(\frac{1}{h_i}) + (\frac{\mathrm{D}_0 * \ln(\frac{D_0}{D_i})}{2 * k}) + (\frac{1}{h_o})} \tag{22}$$

Surface area,

$$A_{\text{surface}} = \pi * D_o * N_t * L \tag{23}$$

Pitch size,

$$P_t = D_0 * 1.25 \tag{24}$$

$$S_T = P_t \tag{25}$$

$$S_L = (\frac{3^{\frac{1}{2}}}{2})P_t \tag{26}$$

$$S_D = (S_L^2 + (\frac{S_T^2}{2}))^{\frac{1}{2}} \tag{27}$$

Baffle spacing (size),

$$B = 0.5 * D_s \tag{28}$$

Number of baffles,

$$N_{\text{baffle}} = \frac{L}{B} - 1 \tag{29}$$

The outlet temperature,

$$\frac{\Delta T_o}{\Delta T_i} = \frac{T_s - T_{c,o}}{T_s - T_{c,i}} = \exp\left(\frac{-\pi D N \bar{h}}{\rho V N_t S_t c_p}\right) \tag{30}$$

Pressure drops,

$$\Delta P = f * \frac{L}{2 * D} \rho * V^2 \tag{31}$$

3. DESIGN SPECIFICS

In the design of shell and tube heat exchangers, the configuration of shell and tube passes significantly impacts performance. For instance, a setup with 1 shell pass and 2 tube passes offers a good compromise between heat transfer efficiency and pressure drop, making it suitable for moderate heat loads and flow rates due to its lower complexity and cost. Conversely, a configuration with 1 shell pass and 4 tube passes enhances thermal performance by increasing the fluid's exposure to the heat exchange process, though it leads to higher pressure drops and increased pumping requirements. Selecting the correct number of passes is crucial to achieving the desired heat transfer coefficient and maintaining the required tube side velocity. Typically, the number of passes ranges from 1 to 16, with a common selection range between 1.2 to 4. For this design project, a configuration of 1 tube pass and 1 shell pass has been chosen to meet specific thermal and flow requirements.

3.1. Sodium-Cooled Fast Reactor (SFR)

Sodium-Cooled Fast Reactors are preferred due to sodium's thermal conductivity and high boiling point. This allows efficient heat transfer at low pressures. This reduces the risk of high pressure-related failures and ensures plant safety. Sodium allows high reactor outlet temperatures to be achieved, which leads to the formation of highly heated steam that increases the efficiency of the Rankine cycle. In addition, sodium does not cause energy losses during the scattering of neutrons. This feature allows more neutrons to be released per fission during scattering and increases reactor efficiency. [1]

3.2. Steam Turbine

Steam turbines are used to convert the heat of steam under high temperature and pressure into mechanical energy. The steam turbine transfers this work to generators, allowing usable energy to be produced. The use of turbines for power cycles in nuclear power plants is necessary for the transition from thermal energy to mechanical energy.

3.3. Shell-and-Tube Heat Exchanger

Shell and tube exchangers are highly efficient devices used to transfer heat between two fluids. These devices consist of a tube bundle inside a cylindrical shell. This structure provides a heat transfer surface that increases thermal efficiency.

These devices, which can operate effectively in the extreme environments typical of nuclear power plants, offer the ability to manage pressures across the shell and tube sides, thus ensuring efficient heat transfer between the primary sodium loop and the secondary water/steam loop. The design minimizes contamination risks by isolating the radioactive primary coolant from the non-radioactive secondary side. Leak detection systems provide security under changing conditions [2]

3.4. Number of Shell and Tube Passes

In the design of shell and tube heat exchangers, the configuration of shell and tube passes are crucial for the performance. For example, a setup with 1 shell pass and 2 tube passes offers a good value between heat transfer efficiency and pressure drop, making it appropriate for moderate heat loads and flow rates due to its lower complexity and cost. But, with a configuration with 1 shell pass and 4 tube passes improves thermal performance by increasing the fluid’s exposure to the heat exchange process, but it leads to higher pressure drops and increased pumping requirements. Selecting the correct number of passes is crucial to achieving the desired heat transfer coefficient and maintaining the required tube side velocity. Typically, the number of passes ranges from 1 to 8 or more, the most common selections are 1, 2 or 4. For this design project, a configuration of 1 tube pass and 1 shell pass has been chosen to meet specific thermal and flow requirements. [3] [7]

3.5. Flow Type

Counterflow configuration is commonly used for its superior heat transfer efficiency, which is primarily due to its minimized thermal stress and continuous temperature gradient along the heat exchanger's length. For these reasons for this heat exchanger counter-flow is used. In this arrangement, hot and cold fluids move in opposite directions, which not only increases the heat transfer rate but also requires the least amount of space for effective heat transfer, allowing the system to reach the inlet fluid temperature. This design uses counter-flow configuration to use these advantages, especially in the energy-intensive environment of a nuclear power plant, where high thermal efficiency is crucial for optimal performance. [4]

3.6. Tube Arrangements

The tube arrangement type is very important in shell and tube heat exchangers. For this parameter, triangular arrangement and square arrangement are generally used depending on the type of fluid to be processed and system requirements.

- Triangular Pitch:

Triangular pitch arrangement increases the heat transfer surface area by allowing more tubes to be placed in a shell diameter. This arrangement causes the liquid to create turbulence and increases heat efficiency. However, the narrow distance between the tubes is disadvantageous in terms of cleaning. It can cause problems especially in cases of calcification and sediment formation.

- Square Pitch:

In the square pitch arrangement, the presence of wide tube spaces provides ease of cleaning and reduces maintenance costs due to low pressure losses. However, in this arrangement, the heat transfer surface area is lower due to the low number of tubes, which reduces efficiency.

Studies have shown that the triangular arrangement provides much more heat transfer efficiency than the square arrangement in case of diameter increase. This ratio is 3% for the triangular and 2.8% for the square arrangement in case of 0.005m shell diameter increase. While the square arrangement is preferred in low pressure systems because it allows larger shell diameters, the triangular arrangement is suitable for use when higher pressure and heat efficiency are required. [5]

3.7. Baffles

Baffles are used in the shell to direct the fluid stream over the tubes, increasing fluid velocity and thus the rate of transfer. They also support the tubes. Single segmental baffles are the most common form used in exchangers and for this exchanger single segmental baffles are used. The distance between the tubes, or baffle spacing, should be between 20%–50% of inner diameter of the shell. Higher turbulence is caused by a decrease in baffle spacing, which raises the heat transfer coefficient. The segmental baffle cut represents the proportion of the baffle diameter. The baffle cuts range between 15–45%. The most optimum baffle cut should be chosen between 20 and 25% for an adequate and fair pressure drop and heat transfer coefficient. The chosen values of baffle cuts are 25% and baffle spacing is 0.5 in this project. [6]

3.8. Material Selection

316 stainless steel was selected for this project due to its corrosion resistance, high mechanical strength, and ability to withstand high pressures. Material selection in heat exchanger design is

very important in terms of heat efficiency and regular operation of the system. The selected material must be resistant to high temperature gradients and must be able to withstand parameters such as pressure and temperature. Factors such as corrosion, material fatigue, creep, and fracture resistance are also parameters considered in material selection. In order for the selected material to have a long life, it must be resistant to corrosion and not react with sodium. 316 stainless steel was selected for this project because it meets these characteristics. [5] [7]

3.9. Parameters

The parameters are optimized to balance pressure drop and heat transfer efficiency. Larger diameters reduce pressure drops but may decrease turbulence, while smaller diameters enhance turbulence but can lead to higher pressure drops. For the first correlation the tube count stayed at 10000 and the diameters used were 0.03526, 0.039 and 0.04526. In the second correlation the tube diameter stayed constant and the chosen tube numbers were 9800, 10000 and 10200. In the last correlation everything stayed constant and Shell diameters were 4.8, 5 and 6.

4. DESIGN AND CALCULATION

Calculations,

For internal flow, from equation (1) film temperature for sodium is:

$$T_m = \frac{772K + 633\text{K}}{2} = 702.5\ K$$

Table 1: Thermal Proporties of Sodium at 702.5 K

Proporties	Values
Density kg/m^3	902.6
Dinamic Viscosite Ns/m^2	3.43*10^{-10}
Kinematic Viscosite	3.096*10^{-7}
Prandtl Number	0.0048
Thermal Conductivity W/mK	70.08
Mass Flow Rate kg/s	4684.5

From equation (4), to calculate mass flow rate of sodium,

$$840 * 10^3 \frac{kj}{s} = \dot{m} * 1.29 \frac{kj}{kg * K} * (499\text{C}° - 360\text{C}°)$$

$$\dot{m} = 4684.5 \frac{kg}{s}$$

From equation (17) to calculate velocity of sodium from mass flow rate,

$$V = \frac{4684.5 \frac{kg}{s}}{902.6 \frac{kg}{m^3} * 0.0016 m^2} = 0.322 \frac{m}{s}$$

For external flow, from equation (1) film temperature for water is:

$$T_m = \frac{323K + 523\text{K}}{2} = 423\ K$$

Table 2: Thermal Proporties of Water at 423 K

Proporties	Values
Density kg/m^3	0.916
Dinamic Viscosite Ns/m^2	181.4*10^{-6}
Kinematic Viscosite	198.035*10^{-6}

Prandtl Number	1.139
Thermal Conductivity W/mK	0.0006871
Mass Flow Rate kg/s	318

For heat capacity of vapor from equation (2)

$$\frac{C_p}{18.015} = 32.24 + (0.1923 * 10^{-2} * 423) + (1.055 * 10^{-5} * (423^2)) + (-3.595 * 10^{-9} * (423^3))$$

$$C_p = 1.92 \frac{kj}{kg*K}$$

From equation (6) to calculate mass flow rate of water with phase change

$$840 * 10^3 \frac{kj}{s} = \dot{m} * 1.92 \frac{kj}{kg * K} * (499C° - 360C°) + 2257.5 \frac{kj}{kg} * \dot{m}$$

$$\dot{m} = 318 \frac{kg}{s}$$

From equation (17) to calculate velocity of water from mass flow rate,

$$V = \frac{318 \frac{kg}{s}}{0.916 \frac{kg}{m^3} * 2.006 m^2} = 172.98\, m/s$$

From equation (7),

$$C_{Na} = 4684.5 \frac{kg}{s} * 1.29 \frac{kj}{kg * K} = 6043.134 \frac{kj}{s * K}$$

From equation (8),

$$q_{Na} = 6043.134 \frac{kj}{s * K} * (139K) = 840.000\, kj$$

$$q_{Na} = q_{water}$$

$$C_{H20} * (200C° - 50C°) = q_{Na}$$

$$C_{H20} = 4200 \frac{kj}{s * K}$$

Hence, as the result shot that $C_{H20} < C_{Na}$, $C_{H20} = C_{min}$

$C_{min} = C_{H20}$ and $C_{max} = C_{Na}$ and Cr is, from (11)

$$C_r = \frac{4200 \frac{kj}{s * K}}{6043.134 \frac{kj}{s * K}} = 0.695$$

Now, q_{max} should be calculate with using C_{min} so from equation (9)

$$q_{max} = 4200 \frac{kj}{s * K} * (400C° - 50C°) = 1885790.2\, kj$$

Now effectiveness can be calculated from equation (10),

$$\varepsilon = \frac{840000}{1885790.2} = 0.445$$

The heat transfer rate is divided by the highest feasible heat transfer rate to determine effectiveness. NTU is computed for one shell pass and one tube pass. The ideal surface area is later found using the log mean temperature difference. NTU formula is taken from Table 11.4 and equation 11.29b for counterflow and Cr < 1, from Incropera.

Now. From equation (12) and (13),

$$NTU = \frac{-1}{0.695 - 1} * \ln\left(\frac{0.445 - 1}{0.445 * 0.695 - 1}\right) = \ 0.718$$

$$E = \frac{\frac{2}{0.445} - (1 + 0.695)}{(1 + 0.695^2)^{0.5}} = 2.295$$

Obtained Reynold's number is between 3000 and 5 x 106 and it shows that the flow is fully developed turbulent flow. Friction factor is computed by using the friction factor tables from Geankoplis.

So now we can calculate pressure drops for Na and H2O from equation (31)

$$\Delta P_{\text{Na}} = 0.0043 * \frac{28.95m}{2 * 0.04526m} * 902.6\ \frac{kg}{m^3} * \left(0.322 \frac{m}{s}\right)^2 = 130.3\ Pa$$

$$\Delta P_{\text{H20}} = 0.0121 * \frac{28.95m}{2 * 5m} * 0.916\ \frac{kg}{m^3} * \left(172.98 \frac{m}{s}\right)^2 = 992\ Pa$$

After determining the velocity Reynold Number calculated with;

$$Re_D = \frac{\rho * V * D}{\mu}$$

$$Re_D = 47158.78$$

When examining the Reynolds Number, it was determined that the flow is turbulent. Since the Prandtl Number is less than 0.1, the Seban-Shimazaki correlation was used.

For calculate the Nusselt number;

$$Nu_D = 5 + 0.025 * Re_D{}^{0.8}\ Pr^{0.4}$$

$$5 + 0.025 * 47158^{0.8}\ 0.0048^{0.4} = 21.19$$

To calculate convection coefficients;

$$Nu_D = \frac{h_i * D_i}{k} \qquad >> \qquad h_i = \frac{Nu_D * D_i}{k}$$

$$h_i = 32812.61308 \frac{W}{m^2\,K}$$

Since multiple tubes are used and there is crossflow over these tubes, the equation found in Incropera 7.64 was used.

When calculating the Reynolds number, the maximum velocity must be determined. The maximum velocity is derived based on the S_D value, which is calculated using S_L and S_T.

In our calculation, S_T was determined using the pitch ratio with the formula:

Then, S_L was calculated using a different equation. From these values, we ultimately calculated the maximum velocity.

$$S_T = D_0 \times 1.25 = P_t$$

$$S_L = \left(\frac{3^{\frac{1}{2}}}{2}\right) P_t = 0.052$$

$$S_D = \left(S_L^2 + \left(\frac{S_t}{2}\right)^2\right)^{\frac{1}{2}} = 0.067$$

$$V_{max} = \frac{S_T}{2(S_D - D_0)} * V = 432.4499\ m/s$$

Reynold Number calculated with;

$$Re_D = \frac{\rho * V_{max} * D}{\mu}$$

$$Re_D = 104883.56$$

The Reynolds number falls between 2000 and 2×10^6, and the Prandtl number is 1.19. Therefore, the equation found in Incropera 7.64 was used;

$$Nu_D = C_1 * C_2 * Re_D^m * \mathrm{Pr}^{0.36} * \left(\frac{\mathrm{Pr}}{Pr_s}\right)^{\frac{1}{2}}$$

The constants C_1, C_2, and m were obtained from the tables, and;

Table of Parameters Taken from the Table and Calculated Parameters

Table 3: Table of Parameters Taken from the Table and Calculated Parameters

S_L	0.051	N_L	5000
S_T	0.060	C_1	0.4
S_D	0.060	C_2	1

S_T/S_L	1.154	m	0.6

TABLE 7.7 Constants of Equation 7.64 for the tube bank in cross flow [15]

Configuration	$Re_{D,\max}$	C	m
Aligned	$10–10^2$	0.80	0.40
Staggered	$10–10^2$	0.90	0.40
Aligned	$10^2–10^3$	Approximate as a single (isolated) cylinder	
Staggered	$10^2–10^3$		
Aligned ($S_T/S_L > 0.7$)[a]	$10^3–2 \times 10^5$	0.27	0.63
Staggered ($S_T/S_L < 2$)	$10^3–2 \times 10^5$	$0.35(S_T/S_L)^{1/5}$	0.60
Staggered ($S_T/S_L > 2$)	$10^3–2 \times 10^5$	0.40	0.60
Aligned	$2 \times 10^5–2 \times 10^6$	0.021	0.84
Staggered	$2 \times 10^5–2 \times 10^6$	0.022	0.84

[a]For $S_T/S_L < 0.7$, heat transfer is inefficient and aligned tubes should not be used.

$$Nu_D = 831.889$$

$$h_o = \frac{Nu_D * D_0}{k} = 11.9007\ W/m^2K$$

After calculating the inner convection coefficient and the outer convection coefficient, the overall heat transfer coefficient can be determined.

$$\frac{1}{U} = \frac{1}{\left(\frac{1}{h_{sodium}}\right) + \left(\frac{D_0 * \ln\left(\frac{D_0}{D_i}\right)}{2 * k_{stainless\ steel}}\right) + \left(\frac{1}{h_{water}}\right)}$$

Overall heat transfer coefficient can determined;

$$U = 34.86\ W/m^2K$$

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$$

$$\Delta T_{lm} = \frac{(360-50)-(499-250)}{\ln\big((360-50)-(499-250)\big)}$$

$$\Delta T_{lm} = 278.387$$

$$q = \pi * D_0 * N_t * \Delta T_{lm} * U * L$$

We know the q value is 840 MW. Based on this, we can calculate the length (L).

$$L = \frac{q}{\pi * D_0 * N_t * \Delta T_{lm} * U}$$

L = 28.9599 m

To calculate baffle spacing (size),

$$B = 0.5 * D_s$$

$$B = 0.5 * 4.8m$$

$$B = 2.4m$$

Number of baffles,

$$N_{\text{baffle}} = \frac{L}{B} - 1$$

$$N_{\text{baffle}} = \frac{28.9559}{2.4} - 1$$

$$N_{\text{baffle}} = 11$$

Table 4: Results for Varying Number of Tubes with Tube Diameter of 0.04526 m and Shell Diameter of 4.8

Outside Diameter, m	0.04803	0.04803	0.04803
Inside Diameter, m	0.04526	0.04526	0.04526
Number of Tubes	10000	10200	9800
Velocity of Tubes, m/s	0.322	0.3162	0.32917
Velocity of Shell, m/s	172.98	206.009	149.078
Reynolds Number Tube Side	47158.78	46234.09	48121.2
Reynolds Number Shell Side	104883.356	124910.119	90391.02
Nusselt Number of Tube Side	32812.61308	20.93699	21.4552
Nusselt Number of Shell Side	2455.016	2726.396	2245.461
Convection Coefficient of Tube Side, W/m^2K	32812.61	32418.57033	33221.10
Convection Coefficient of Shell Side, W/m^2K	35.12	39.002	32.122
Friction Factor of Tube Side	0.004	0.004	0.004
Friction Factor of Shell Side	0.0121	0.0121	0.0121
Overall Heat Transfer Coefficient, W/m^2K	34.86759	38.69052	31.91
Shell Diameter, m	4.8	4.8	4.8
Shell Area, m^2	18.095	18.095	18.095
Length, m	28.9559	25.58	32.284
Pitch Size, m	0.060	0.060	0.060
Number of Baffles	11	10	12
Pressure Drop for Tube, Pa	120.1841	102.0619	139.5228
Pressure Drop for Shell, Pa	1000.32	1253.54	828.376

Table 5: Results for Varying Tube Diameter Values with 10000 Tubes and Shell Diameter of 4.8

Outside Diameter, m	0.04803	0.04803	0.04803
Inside Diameter, m	0.04526	0.03526	0.039
Number of Tubes	10000	10000	10000
Velocity of Tubes, m/s	0.322	0.53	0.4344
Velocity of Shell, m/s	172.98	41.6712	56.4521
Reynolds Number Tube Side	47158.78	60533.36	54728.36
Reynolds Number Shell Side	104883.356	25266.5949	34228.7303
Nusselt Number of Tube Side	32812.61308	24.77117	23.23923
Nusselt Number of Shell Side	2455.016	1045.101	1253.905
Convection Coefficient of Tube Side, W/m^2K	32812.61	49233.22288	41759.10737
Convection Coefficient of Shell Side, W/m^2K	35.12	14.95	17.94
Friction Factor of Tube Side	0.004	0.004	0.004
Friction Factor of Shell Side	0.0121	0.0121	0.0121
Overall Heat Transfer Coefficient, W/m^2K	34.86759	14.744	17.73
Shell Diameter, m	4.8	4.8	4.8
Shell Area, m^2	18.095	18.095	18.095
Length, m	28.9559	68.475	56.93
Pitch Size, m	0.060	0.060	0.060
Number of Baffles	11	28	23
Pressure Drop for Tube, Pa	120.1841	990.3851	497.41
Pressure Drop for Shell, Pa	1000.32	137.283	209.472

Table 6: Results for Shell Diameter Values with 10000 Tubes and Tube Diameter of 0.04526

Outside Diameter, m	0.04803	0.04803	0.04803
Inside Diameter, m	0.04526	0.04526	0.04526
Number of Tubes	10000	10000	10000
Velocity of Tubes, m/s	0.322	0.322	0.322
Velocity of Shell, m/s	172.98	97.89	28.48

Reynolds Number Tube Side	47158.78	47158.78	47158.78
Reynolds Number Shell Side	104883.356	59355.8642	17273.9517
Nusselt Number of Tube Side	32812.61308	21.19	21.19
Nusselt Number of Shell Side	2455.016	1744.651	831.8889
Convection Coefficient of Tube Side, W/m^2K	32812.61	32812.62	32812.61
Convection Coefficient of Shell Side, W/m^2K	35.12	24.96	11.9007
Friction Factor of Tube Side	0.004	0.004	0.004
Friction Factor of Shell Side	0.0121	0.0121	0.0121
Overall Heat Transfer Coefficient, W/m^2K	34.86759	24.83	11.87
Shell Diameter, m	4.8	5	6
Shell Area, m^2	18.095	19.64	28.74
Length, m	28.9559	40.6	85.04
Pitch Size, m	0.060	0.060	0.060
Number of Baffles	11	15	28
Pressure Drop for Tube, Pa	120.1841	168.7667	352.9902
Pressure Drop for Shell, Pa	1000.32	431.881	63.75

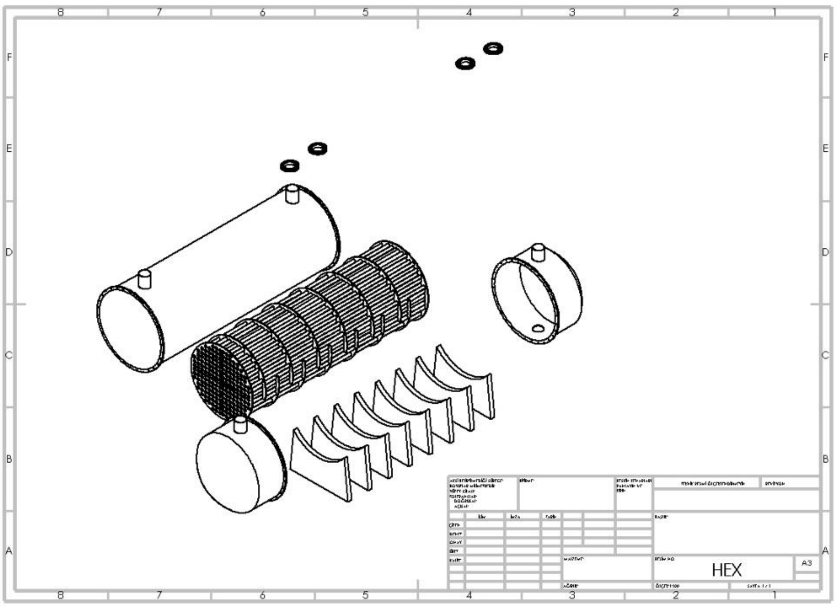


Figure 1: Exploded View of the Heat Exchanger

5. RESULTS AND DISCUSSION

This project concentrated on the design of a shell and tube heat exchanger that would optimize the heat transfer performance under certain operating conditions. The heat exchanger was required to handle two very different kinds of fluids: sodium, which served as the primary heat carrier, and water, which was the secondary coolant. The project team specified a sodium inlet temperature of 702.5 K, a pressure calculated to give a mass flow rate of 4684.5 kg/s, and a Prandtl number of 0.0048 indicative of a fluid with very high thermal conductivity. Water, on the other hand, was specified to come into the heat exchanger at a temperature of 423 K and at a flow rate of 318 kg/s. To avoid any phase change problems, the heat exchanger was designed to keep water in a liquid state along the entire length of the heat exchanger. The project team calculated that the outlet temperature of the sodium coming out of the heat exchanger would be 616 K. Water, on the other hand, was specified to leave the heat exchanger at a temperature that was optimized for what the downstream turbine required. Initial computations entailed working out the thermophysical properties of the fluids, like density, dynamic viscosity, and kinematic viscosity, from their mean film temperatures. Once the conditions for entry and exit were established, we selected the materials, paying special attention to the thermal and chemical properties of the fluids involved. For the shell, we chose Inconel 625 stainless steel because it has a high thermal conductivity, which is absolutely necessary when dealing with sodium, and because sodium's highly corrosive nature demands an equally tough material. For the tube, we selected AISI type 316L stainless steel because of its proven durability against another highly corrosive material water at elevated temperatures. To maximize the unit's thermal efficiency, we employed a counterflow configuration, which serves up an optimal energy exchange between sodium and water. Finally, we varied a whole bunch of parameters tube diameter, number of tubes, shell diameter, and the like to see what happened to the heat transfer efficiency and the pressure drops. We studied three configurations for varying shell and tube diameters. From the results, we determined that using 10,000 tubes with a tube diameter of 0.04526 m gave us the best combination of pressure drop and area for heat transfer. The overall heat transfer coefficient was 34.87 W/m^2K, and the flows were both turbulent, as confirmed by the Reynolds numbers of 47158.78 for the shell side and 104883.356 for the tube side. Further design considerations involved the tube pitch and baffle spacing to optimize flow dynamics. A 30° triangular pitch with a pitch ratio of 1.25 was selected to enhance the tube density within the shell and to ensure a higher heat transfer rate. The baffling was set at 0.5 times the shell diameter. This promoted an optimal shell-side sodium flow velocity of 46.228 m/s in the shell. The shell-side flow was confirmed to be turbulent with a Reynolds number greater than 2300, which is essential for effective heat transfer. The total shell area of 18.095 m^2 and a calculated heat exchanger length of 28.956 m and gave the heat exchanger something like "robust performance." Pressure drops were evaluated, and the results indicated an acceptable pressure drop of 120.1841 Pa for the tube side and 99969.7 Pa for the shell side, ensuring that the system is reliable under the supposed steady-state conditions that we designed it for. Among the many designs that we tried out, one configuration was chosen as the final design because it delivered the greatest effectiveness 46% while still using a minimal amount of surface area for heat transfer and maximizing the overall heat transfer coefficient.

6. CONCLUSION

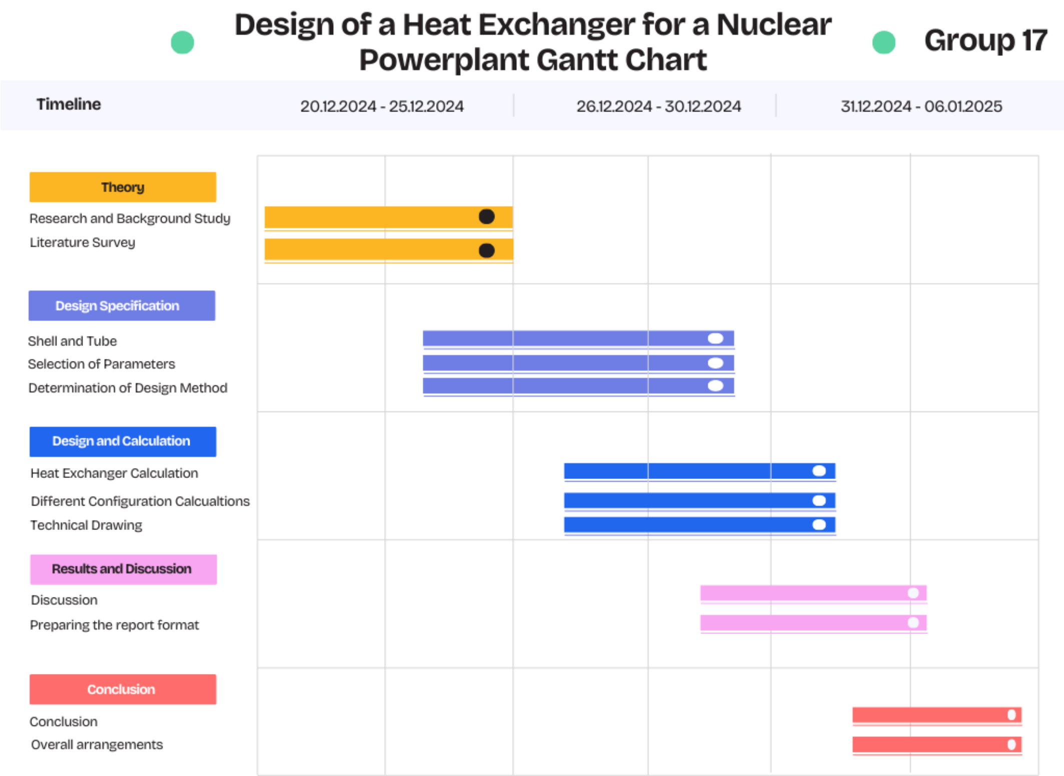
The project's goal is to design a heat exchanger that can effectively extract heat from a nuclear power plant's reactor core. The aim was to transfer 840 MWt of energy from the sodium-cooled fast reactor. Initially, a literature survey was conducted to gather comprehensive information on heat exchangers, heat exchanger designs and calculations. The research revealed that shell-and-tube heat exchangers are the most used and most efficient in the industry due to their well-known limitations, ease of application, and adaptability to various industrial processes.

Numerous considerations were made when developing this application, and nuclear power plants were examined. Sodium was selected for this project as the fluid that gets heated by taking the heat from the nuclear reactor. The heat exchanger we designed has an entry temperature of 499°C for this hot fluid, and an output temperature of 360°C. A heat exchanger was designed for this project when the data was analyzed.

The design and calculation process underscored the intricacy of heat exchanger construction, where multiple parameters, such as Reynolds number, are influenced by initial conditions. This complexity necessitates the use of tools like Excel for calculations, as it allows dynamic updates and recalculations of data when parameters and conditions are modified.

The heat exchanger's shell and tube design was chosen. To satisfy certain thermal and flow requirements, a configuration consisting of one tube pass and one shell pass has been selected for this design project. Sodium has excellent transport and heat transfer capabilities at high temperatures. High heat release rates, or high-power densities as reactors are known, are made possible by several characteristics. Because of its excellent mechanical strength, resistance to corrosion, and capacity to tolerate high pressures, 316 stainless steel was chosen for this project. The choice of material in heat exchanger design is crucial for both the system's regular functioning and heat efficiency. After choosing 316 stainless steel as the primary material, the design was created by rearranging the tubes in two distinct ways. To compare these designs, the diameter length was selected as the parameter to be changed. In addition, by discussing the results the final values met the specified parameters but also exhibits an effectiveness of 44%. In summary, the values we selected and calculated were quite efficient.

7. PROJECT PLANNING



Task	Person
Research and Background Study	Everyone
Literature Survey	Everyone
Shell and Tube	İrem Turhan, Baranalp Çinar, Sinem Börekcioğlu
Selection of Parameters	Everyone
Determination of Design Method	Kaan Gürsel, Tayfun Özel
Heat Exchanger Calculations	Everyone
Different Configurations Calculations	Everyone
Technical Drawing	Kaan Gürsel, Tayfun Özel, Berat Balmış, Gamze Uyaroğlu
Discussion	Everyone
Preparing the Report Format	Berat Balmış, Gamze Uyaroğlu
Result and Discussion	Everyone
Conclusion	İrem Turhan, Baranalp Çinar, Sinem Börekcioğlu

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9. APPENDIX

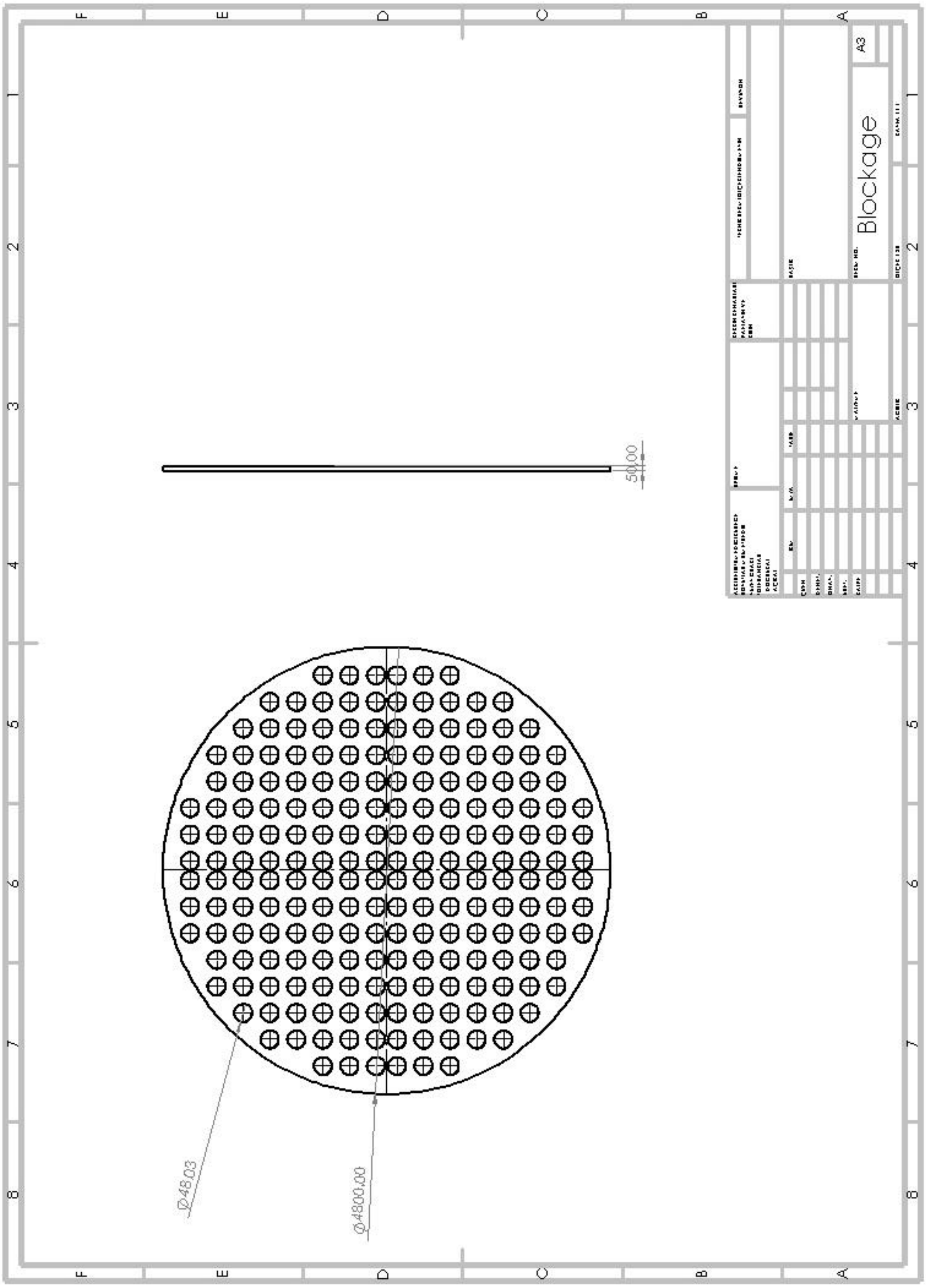

Figure 2: Figure of the Blockage

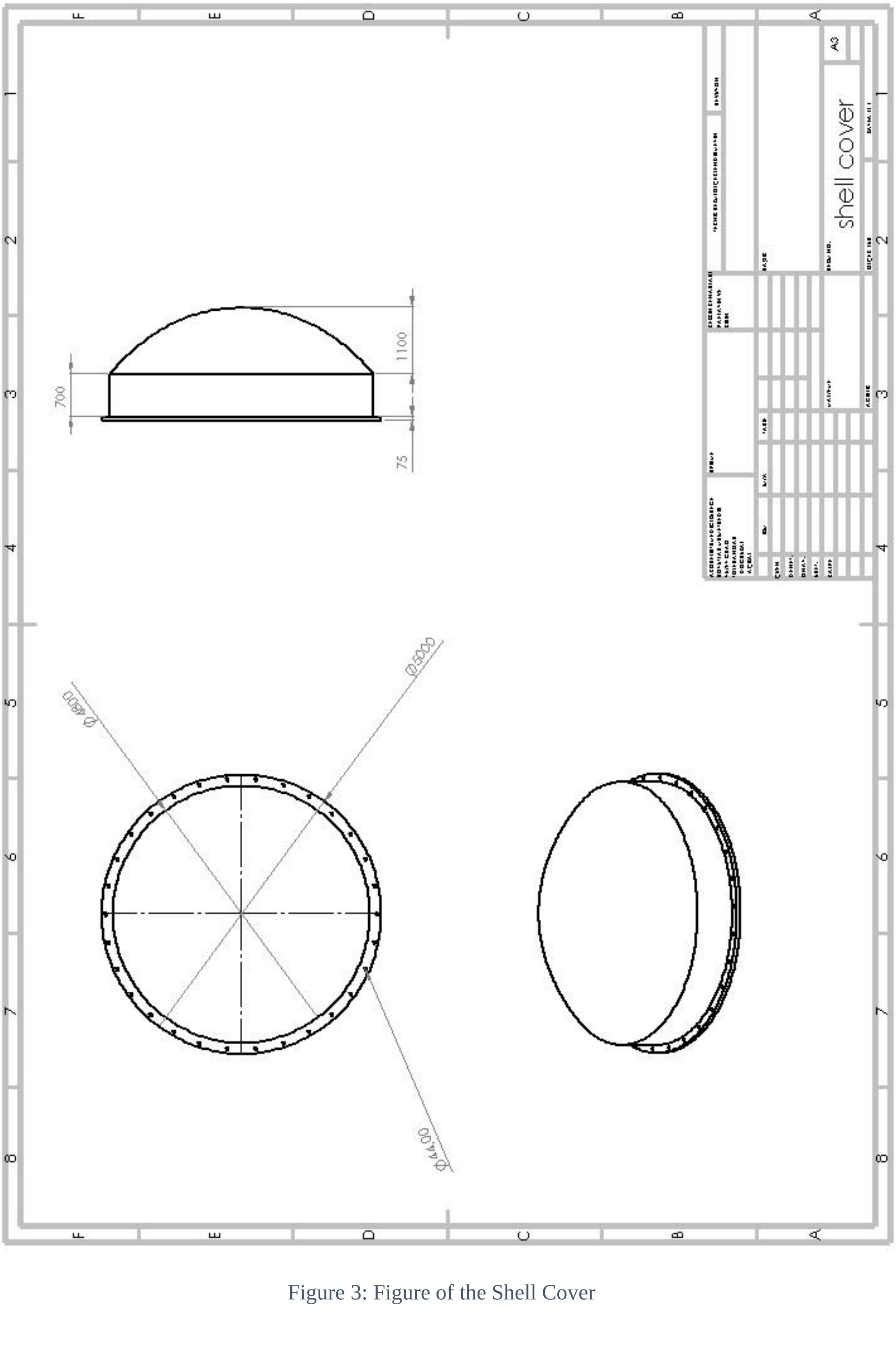

Figure 3: Figure of the Shell Cover

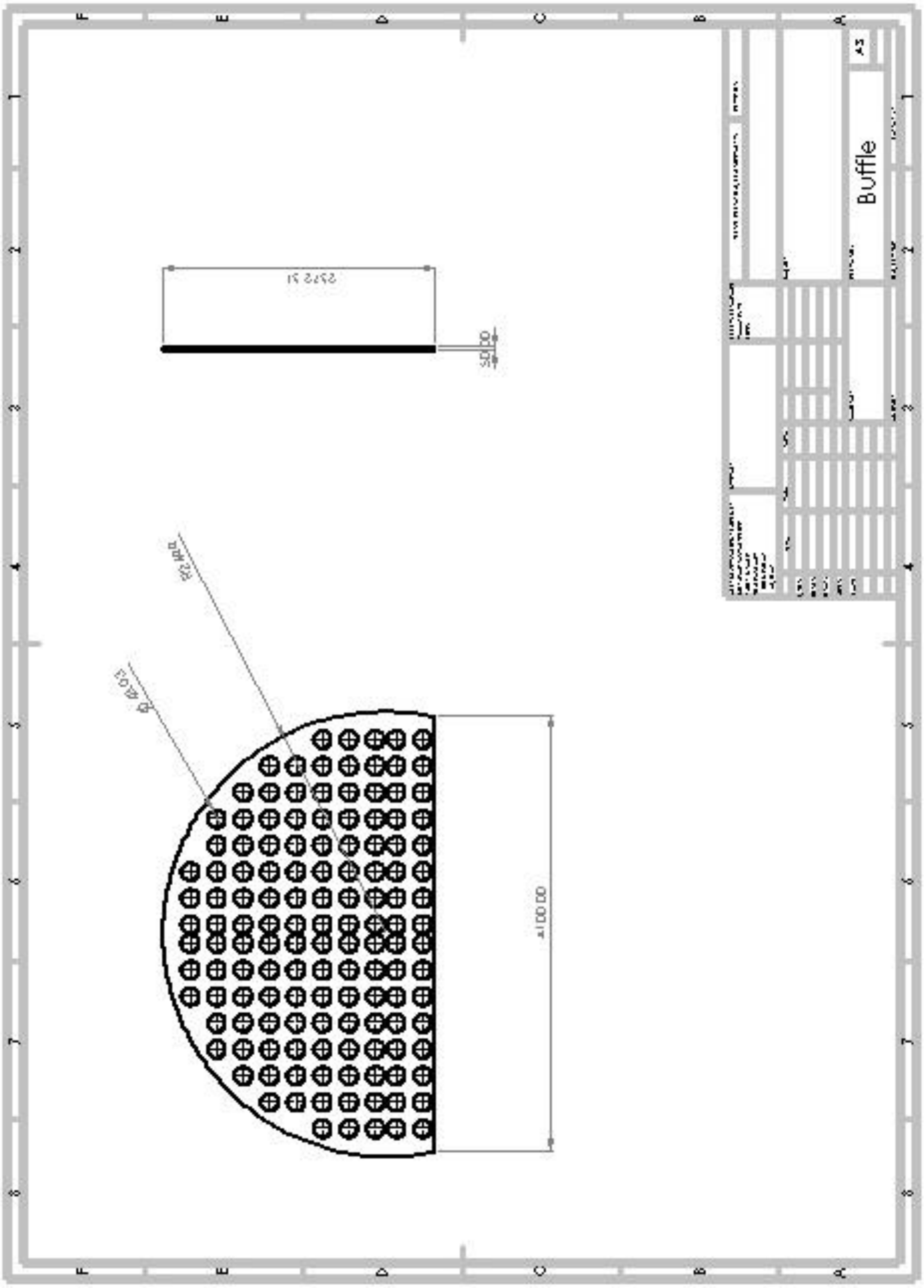

Figure 4: Figure of the Buffle

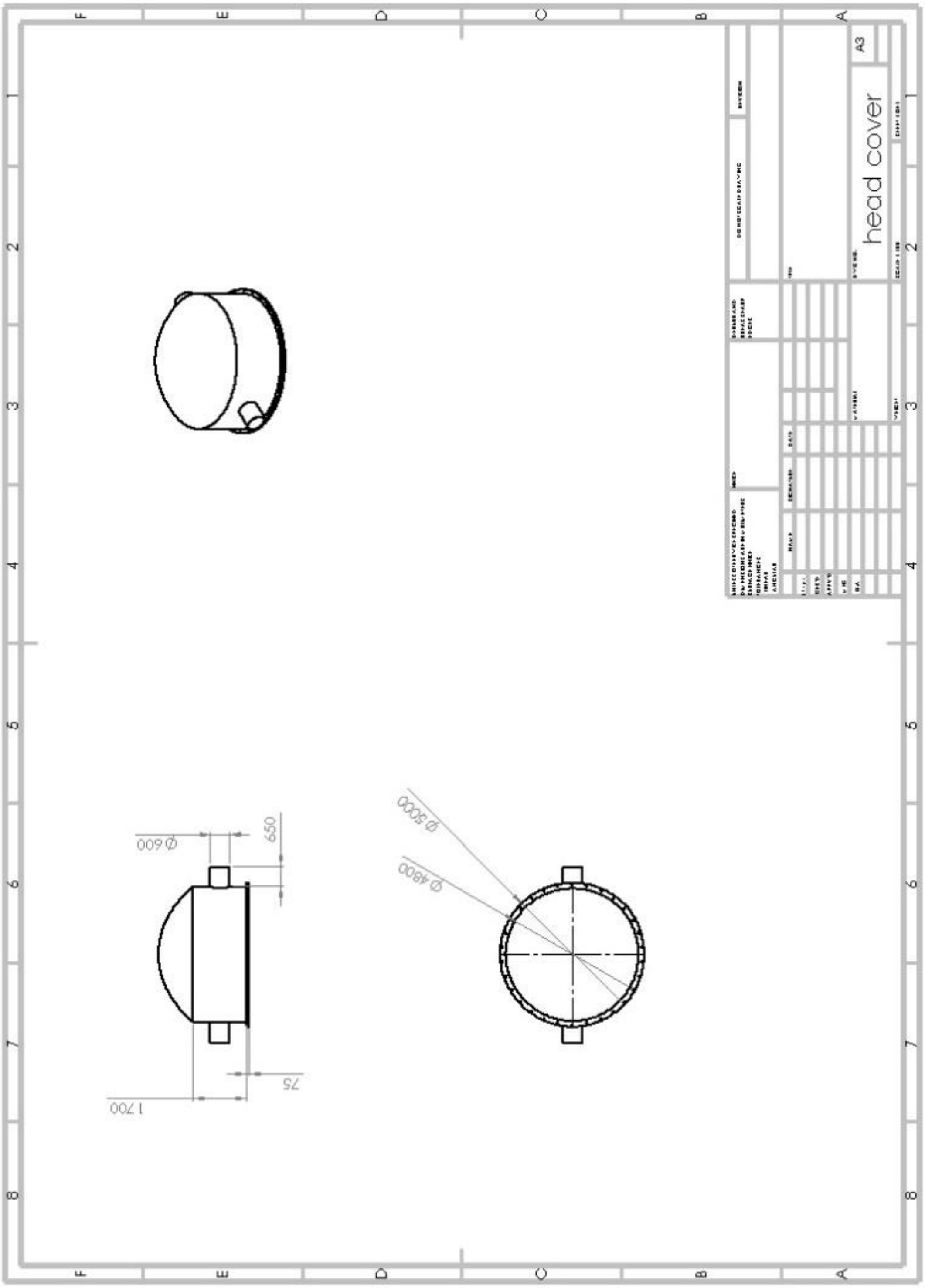

Figure 5: figure of the Head Cover

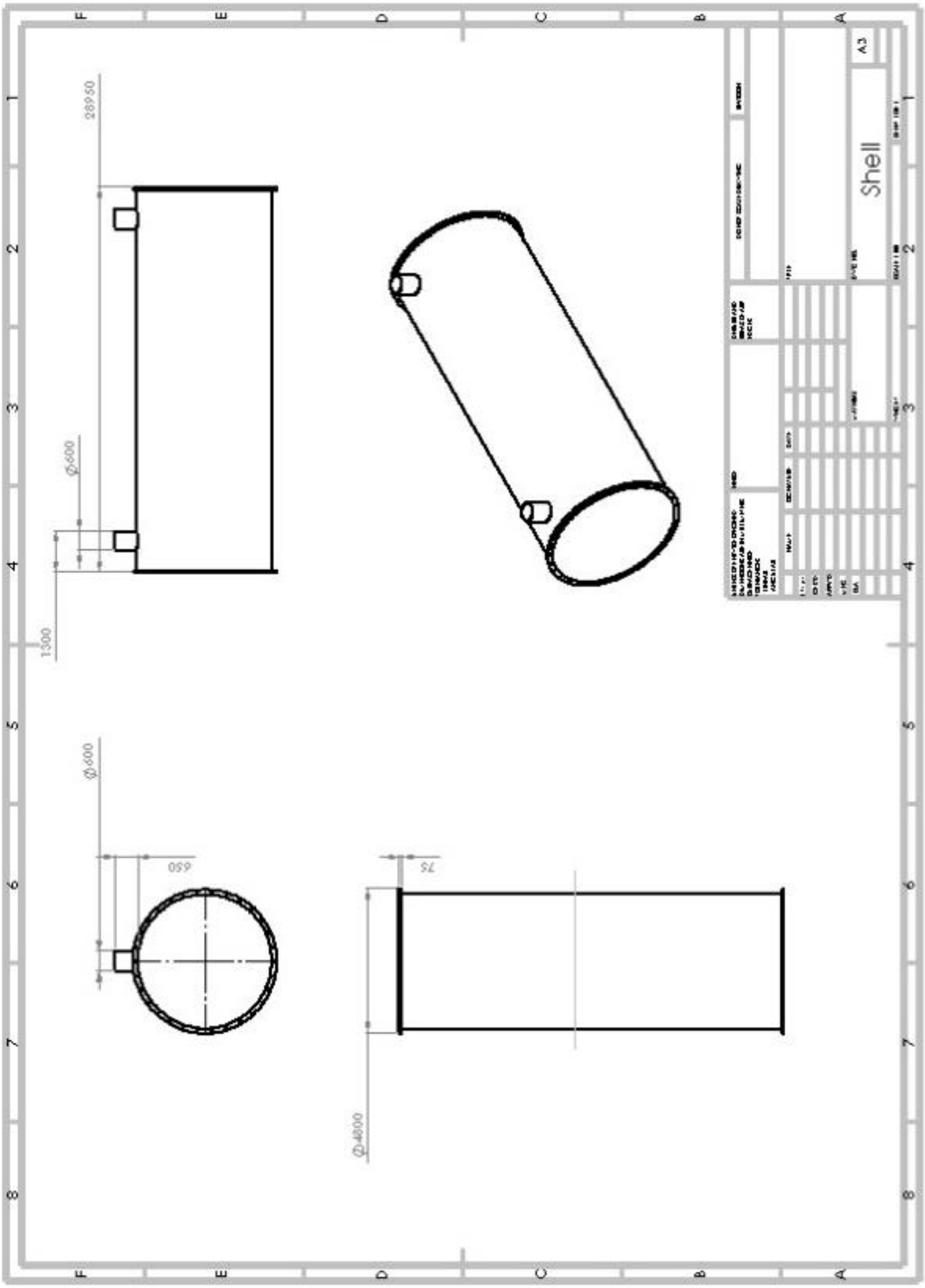

Figure 6: Figure of the Shell

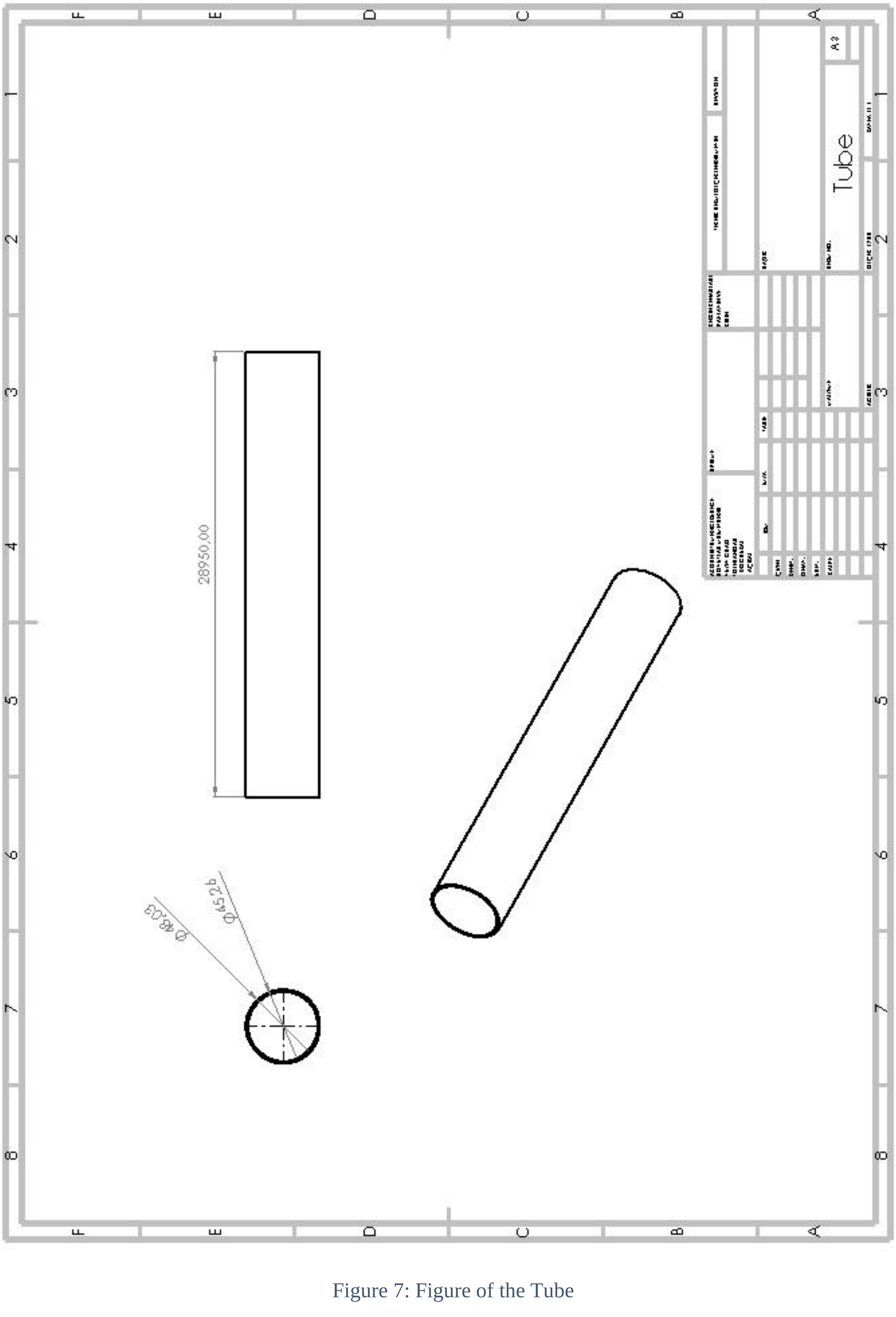

Figure 7: Figure of the Tube

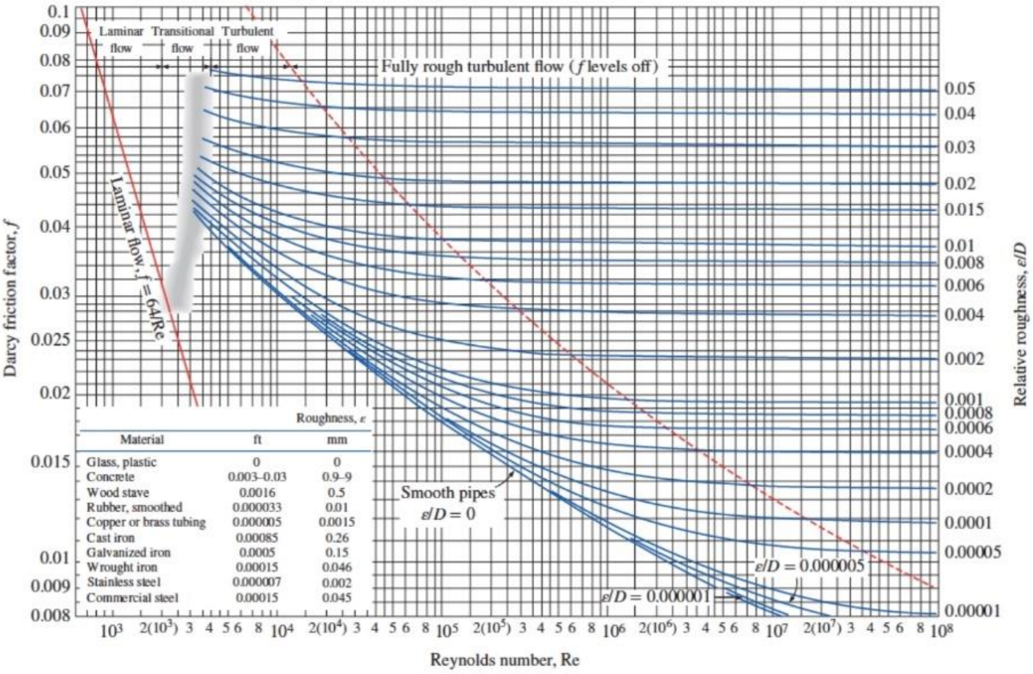

FIGURE A–12

The Moody chart for the friction factor for fully developed flow in circular pipes for use in the head loss relation $h_L = f\frac{L}{D}\frac{V^2}{2g}$. Friction factors in the turbulent flow are evaluated from the Colebrook equation $\frac{1}{\sqrt{f}} = -2\log_{10}\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right)$

Figure 8: Dancy Friction Factor