DOUBLE-ENDED AZIMUTH DRIVE FERRY

Piping System and Mechanical Calculations

Prepared for: NCDOT • Raleigh, North Carolina

Ref: 18026-200-505-1 Rev. - August 10, 2018

PREPARED BY

Elliott Bay Design Group – North Carolina, PLLC 5305 Shilshole Ave. NW, Ste. 100 Seattle, WA 98107

REVISIONS

GENERAL NOTES

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1 PURPOSE

This report documents the calculations used to devleop of the machinery and piping systems for the DOUBLE-ENDED AZIMUTH DRIVE FERRY, a design for the North Carolina Department of Transportation. The subject vessel is a 183 foot 7 inch long by 46 foot 10 inch wide by 10 foot 6 inch deep passenger and vehicle ferry intended for service within the Outer Banks of North Carolina, and associated rivers.

The calculations relating to each specific system are presented in separate appendices. Each appendix provides the following information:

- Description of the system
- Calculation procedures
- Given and assumed parameters
- Formulas and software used
- Calculation results
- References used in preparing the calculations

2 REGULATORY FRAMEWORK

The DOUBLE-ENDED AZIMUTH DRIVE FERRY will be inspected by the US Coast Guard under the provisions of 46 CFR Subchapter H. Piping and mechanical system designs shall comply with the applicable regulations.

Appendix A

Cooling System, Dwg. 18026-200-256-1

DESCRIPTION $\mathbf{1}$

This appendix documents the first principles calculations used in designing the cooling system. These calculations are used to identify steady state frictional losses throughout the piping system and to validate system design.

PROCEDURE 2°

Calculations are presented in the following sequence:

- Pipe size calculations
- Seawater cooling system pressure calculations
- Sea Chest Sizing
- Freshwater Cooling System Calculations

Frictional losses through the piping system are calculated by constructing a model using PIPE-FLO Professional software utilizing the Darcy-Weisbach method.

GIVEN AND ASSUMED PARAMETERS $\mathbf{3}$

- The seawater cooling systems will be constructed of Class 200 copper nickel pipe.
- Maximum summer water temperature is assumed to be 86°F.
- System elevations were estimated from [1] and all elevations are in reference to the vessel's baseline:

• Piping system lengths, routing, fittings, etc. are estimated based on [1] and [2].

3.1 Generator Cooling System

- Each main generator is supplied with an engine mounted sea water pump.
- Each main generator is supplied with engine mounted coolers for jacket water and separate circuit aftercooler.
- The system is designed for sea water supply to two generators, and the auxiliary seawater system, and ballast system with one sea chest closed. Under normal operation, both sea chests are intended to be open.
- The fluid medium is seawater with the following properties:

• Reference [3] provides pump data used to construct the engine cooling models.

3.2 Auxiliary Seawater Cooling System

 Auxiliary sweater cooling demands, estimated from preliminary vendor data, are as follows:

Item	Flow Rate	Pressure Drop
Machinery HVAC Chillers [4]	9 gpm x $4 = 36$ gpm	10 psi
Freshwater Cooling Heat Exchanger [5]	30 gpm x $1 = 30$ gpm	5.6 psi
Propulsion Drive Heat Exchanger	15 gpm $x = 20$ gpm	10 psi

Table 3-2: Auxiliary Seawater Demands

- Flowrate and pressure drop for propulsion drive heat exchangers estimated. Flowrate based on 34kW estimated heat rejection [6] and a seawater temperature rise of 10°F or less. A 10 psi pressure drop through the heat exchanger was assumed.
- The fluid medium is seawater with the following properties:

- Freshwater cooling heat exchangers are sized so that each can handle the full heat rejection of all four motors, allowing one heat exchanger to be out of service while vessel is operating.
- Seawater cooling branches are balanced with Hays Measurflo valves, which have a minimum operating pressure of 2 PSI.

3.3 Freshwater Cooling System

- The freshwater cooling system will be constructed with SCH 10S stainless steel pipe.
- The fluid medium is a 30% ethylene glycol solution with the following properties:

• Freshwater flowrate requirements are based on four (4) Ramme SW500_S_250_1241_B liquid cooled electric motors integrated provided with Schottel SCD 200 STP 150 thrusters. [7]

FORMULAS $\overline{\mathbf{4}}$

(not used)

5 **CALCULATIONS**

5.1 Pipe Size Calculations

Pipe sizes are based on the nominal velocity limits found in Marine Engineering, Chapter 20, Table 3, [8]. The following table shows the flow rates of cooling water occurring in the system, and the resulting pipe sizes.

5.2 Generator Cooling System Pressure Calculations

As shown in the attached pipeflo model of the seawater cooling piping, the calculated pressure drop of 27.7 feet H2O result in a flow rate of 74.2 gpm for the given pump.

5.3 Auxiliary Seawater Cooling System Pressure Calculations

As shown in the attached pipeflo model of the required pump capacity and head for the auxiliary seawater cooling system is 45.2 feet H2O at a flow rate of 96 gpm.

5.4 Freshwater Cooling System Pressure Calculations

As shown in the attached pipeflo model of the required pump capacity and head for the freshwater seawater cooling system is 71.4 feet H2O at a flow rate of 51.9 gpm.

REFERENCES 6

- [1] Elliott Bay Design Group, "NCDOT Double-Ended Azimuth Drive Ferry: Profiles and Arrangements," 18026-200-101-1, Seattle, WA.
- [2] Elliott Bay Design Group, "NCDOT Double-Ended Azimuth Drive Ferry: Cooling System Schematic," 18026-200-256-1, Seattle, WA.
- [3] CAT, C18 Auxiliary Pump Performance, Ref. EM 0327, June 21, 2018.
- [4] S. Brigham (2018, July 25), *RE: 18026 HVAC System for Switchboard Room,* email: Available e-mail: sab@flagshipmarine.com, 7/25/18.
- [5] Alfa Laval (MSI / Kevin Oakley), "Plate Heat Exchanger TL3-PFG," EBDG-NDOT Ferry TL3-PFG Propulsion Motor Cooling HX.pdf, Seattle, WA, 07/24/18.
- [6] F. Gonzalez (2018, Jun. 29), *RE: 18026 Z-Drive ferry switchboard room arrangement / questions,* email: felixgonzalez@epdltd.com, 06/29/18.
- [7] Ramme Electric Machines GMBH, "Techniche Datanblatt/Technical Data sheet SW500_S_250_1241_B," Osterweick, Germany, 07/24/18.
- [8] R. L. Harrington, Marine Engineering, Jersey City, NJ: SNAME, 1992.

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MAIN ENGINE SEA WATER PUMP CURVE

AUXILIARY SEA WATER SYSTEM PIPE-FLO MODEL

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1 x Tee - Flow Thru Branch

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FRESHWATER COOLING SYSTEM PIPE-FLO MODEL

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1 x Reducer - Contraction (2 in x 1.5 in - 3 in)
1 x Reducer - Enlargement (2 in x 1.25 in - 3 in)
1 x Swing Check - Angled
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SEA CHEST SIZING

Appendix B

Exhaust System, Dwg. 18026-200-259-1

1 DESCRIPTION

This appendix documents the calculations used in designing the diesel engine exhaust systems. These calculations estimate frictional losses through the exhaust piping to verify that system backpressure is below the engine manufacturers' published requirements.

2 PROCEDURE

Calculations are presented in the following sequence:

- Main generator exhaust piping ΔP calculations
- Emergency generator exhaust piping ΔP calculations

Frictional losses through the piping system are calculated by constructing a model using PIPE-FLO Professional software utilizing the Darcy-Weisbach method.

3 GIVEN AND ASSUMED PARAMETERS

• Main Generator exhaust gas characteristics are taken from [1] [2] as follows:

Emergency generator exhaust gas characteristics are taken from [3] as follows:

- For the purpose of calculating piping friction losses, exhaust gas pressure is assumed to be standard atmospheric pressure, 14.7 PSIA, plus half the maximum allowable backpressure value listed for the system analyzed.
- In calculating frictional losses through pipe, exhaust gas is assumed to have the same density and dynamic viscosity as air at assumed system pressure and stack temperature noted above for the system analyzed.
- System pipe lengths, routing and fittings are estimated based on the routing shown in [4].
- Selected main generator silencer is a 10 inch Harco 2458VRSA10 with estimated backpressure of 4.5 in H_2O . Backpressure estimated with vendor provided calculation tool [5].
- Selected emergency generator silencer is a 4 inch Harco VRS-4 SISO with estimated backpressure of 9.8 in H_2O . Backpressure estimated with vendor provided calculation tool [5]. In conjunction with the side-inlet, side-outlet silencer, the selected emergency generator spark arrestor is a Harco 5AA. Backpressure of spark arrestor estimated at 1 inH20 based on vendor guidance.

4 FORMULAS

(not used)

5 CALCULATIONS

5.1 Main generator Exhaust Piping ΔP **Calculation**

As shown in the attached PIPE-FLO results, the estimated main engine exhaust piping backpressure is approximately 29.4 in H₂O, 74% of the stated vendor maximum design value.

5.2 Emergency Generator Exhaust Piping ΔP **Calculation**

Exhaust gas volumetric flow rate is first determined using the given mass flow rate from the engine technical data [3].

As shown in the attached PIPE-FLO results, the estimated emergency generator exhaust piping backpressure is approximately 13.6 in H_2O , 23% of the stated vendor maximum.

6 VENDOR DATA

The following vendor provided data was used in the calculations

6.1 CAT C18 Generator rated 565 kW at 1,800 RPM

From [2]:

General Performance Data Top

From [1]:

Harco 2458VRSA10 backpressure calculated using [5]:

CALCULATION OF BACK PRESSURE OF SILENCER

Note: Exhaust Velocity should not exceed 9000 ft/min on all Critical and Super Critical applications and velocity should not exceed 12000 ft/min for the rest of the applications

6.2 CAT C4.4 Generator rated 66 kW at 1,800 RPM

From [3]:

EXHAUST SYSTEM

Exhaust Gas Data

Harco 1236 VRS4SI-SO backpressure calculated using [5]:

CALCULATION OF BACK PRESSURE OF SILENCER

Note: Exhaust Velocity should not exceed 9000 ft/min on all Critical and Super Critical applications and velocity should not exceed 12000 ft/min for the rest of the applications

7 REFERENCES

- [1] Caterpillar, "EM4133; C18 565 ekW at 1800 rpm Systems Data," May 24, 2018.
- [2] Caterpillar, EM4133; C18 565 ekW at 1800 rpm Performance Data, May 24, 2018.
- [3] Caterpillar, "LEHM0240-00; C4.4 ACERT Marine Generator Set Package Specifications," 2016.
- [4] Elliott Bay Design Group, "NCDOT Double Ended Azimuth Drive Ferry: Exhaust Arrangement; 18026-200-259-1," Seattle, WA.
- [5] Harco Manufacturing, "Silencer Back Pressure Calculator," [Online]. Available: http://harcomfg.com/. [Accessed July 2017].
- [6] USFS, "Spark Arrester Guide," 2017.
- [7] R. L. Harrington, Marine Engineering, Jersey City, NJ: SNAME, 1992.
- [8] USCG, "46 CFR, Chapter I, Subchapter F, 56.50-50," 7/21/2017.
- [9] Elliott Bay Design Group, "NCDOT Double Ended Azimuth Drive Ferry: Profiles and Arrangements," 18026-200-101-1, Seattle, WA.

MAIN GENERATOR EXHAUST MODEL

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EMERGENCY GENERATOR EXHAUST MODEL

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Appendix C

Fuel Oil Piping System, Dwg. 18026-200-261-1

DESCRIPTION $\mathbf{1}$

This appendix documents the calculations used in designing the fuel oil system. These calculations are used to identify steady state frictional losses throughout the piping system and to validate system design.

PROCEDURE 2°

Calculations are presented in the following sequence:

- Pipe size calculations
- Engine fuel pump supply pressure calculations
- Engine fuel pump return pressure

Frictional losses through the piping system are calculated by constructing a model using PIPE-FLO Professional software utilizing the Darcy-Weisbach method.

GIVEN AND ASSUMED PARAMETERS $\mathbf{3}$

- The fuel oil system is to be constructed of Schedule 40 carbon steel pipe.
- Maximum fuel flow rate to each generator is 200.2 g/hr = 3.34 gpm, from [1].
- The generators have the following fuel system design constraints [1]: *Table 1: Fuel System Design Constraints*

- Piping system lengths, routing, fittings, etc. are estimated based on the Profiles and Arrangements, and Fuel Oil Diagram [2] [3].
- Each generator fuel supply utilizes a Racor 751000MAXM duplex fuel filter with a pressure drop of 3.5 psi at a flowrate of 6.0 gpm, from [4].
- The system is normally arranged such that one generator consumes fuel from the nearest tank. The model assumes a worst case wherein two generators are utilizing one tank.
- Tank and engine elevations were estimated from [2] and all elevations are in reference to the vessel's baseline:

Table 2: Elevations

$\overline{\mathbf{4}}$ **FORMULAS**

(not used)

5° **CALCULATIONS**

5.1 Pipe Size Calculations

Pipe sizes are based on the nominal velocity limits found in Marine Engineering, Chapter 20, Table 3, [5]. The following table shows the flow rates of fuel occurring in the system, and the resulting pipe sizes.

Pipe Segment	Flow	Pipe	Schedule	d , ID	Design Velocity		
	Rate	Size			Nominal	Limit	
	gpm	(NPS)		\overline{m}	$\mathrm{ft/s}$	$\mathrm{ft/s}$	ft/s
Fill Rate	50	$\overline{2}$	SCH 80	1.939	$2.0 \sqrt{d}$ 2.78	15	5.43
Supply Main (2 Generators)	6.68	$\overline{2}$	SCH 40	2.067	$5.0 \sqrt{d}$	7.19 20	0.64
Return Main (2 Generators)	6.68	$\mathbf{2}$	SCH 40	2.067	$5.0 \sqrt{d}$	7.19 20	0.64
Generator Supply	3.34	3/4	SCH 40	0.824	$5.0 \sqrt{d}$ 4.54	20	2.01
Generator Return	3.34	3/4	SCH 40	0.824	0.91 $1.0 \sqrt{d}$	4	2.01

Table 3: Fuel Oil System Pipe Sizes and Velocities

5.2 Engine Fuel Suction Pressure Calculations

The attached system model confirms that the piping systems are compatible with the generator's fuel supply allowable line restriction requirements. As listed above, the maximum fuel line restriction is 4.37 psi. The attached model shows the calculated pressure loss to be 2.6 psi.

5.3 Engine Fuel Return Pressure Calculations

The attached model of the fuel system's return piping confirms that the backpressure at the generators does not exceed the listed maximum value of 3.93psi. The return line restriction at the generator was found to be 0.77 psi.

6 **VENDOR DATA**

6.1 CAT c18 Generator Rated 565 kW at 1,800 RPM

From [1]:

6.2 Racor 751000 MAXM

From [4]:

Performance Graphs -These results are from controlled laboratory tests. Field results may vary by application.

$\overline{7}$ **REFERENCES**

- [1] Caterpillar, "C18 565 kW at 1800 RPM Systems Data," EM4133, May 24, 2018.
- [2] Elliott Bay Design Group, "NCDOT Double Ended Azimuth Drive Ferry: Profiles and Arrangements," 18026-200-101-1, Seattle, WA.
- [3] Elliott Bay Design Group, "NCDOT Double Ended Azimuth Drive Ferry: Fuel Oil Diagram," 18026-200-261-1, Seattle, WA.
- [4] Parker Hannifin Corp, "Racor Products: Parts, Service and Technical Information (Marine Turbine Series)".
- [5] R. L. Harrington, Marine Engineering, Jersey City, NJ: SNAME, 1992.

FUEL OIL SUPPLY PIPE-FLO MODEL

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FUEL OIL RETURN PIPE-FLO MODEL

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Appendix D

Machinery Ventilation, Dwg. 18026-200-513-1

DESCRIPTION $\mathbf{1}$

This appendix documents the calculations used in designing the machinery ventilation system. The required ventilation rate was calculated for each machinery or void space in the hold, and ducting calculations performed to determine required fan performance. Louver sizes for the emergency generator room are also determined.

$\overline{2}$ **PROCEDURE**

Calculations are presented in the following sequence:

3 **REGULATORY FRAMEWORK**

The DOUBLE-ENDED AZIMUTH DRIVE FERRY will be inspected by the US Coast Guard under the provisions of 46 CFR Subchapter H.

GIVEN AND ASSUMED PARAMETERS $\overline{\mathbf{4}}$

- A maximum outdoor air temperature of 95°F is assumed for hold ventilation calculations. The maximum expected dry bulb temperature over a 20-year period is 95.2 °F. Per ASHRAE climatic data [3], the 0.4% outdoor air design temperature is 88.1 \degree F, which represents a temperature that is exceeded fewer than 40 hours per year on average.
- The Engine Room, Voids, and Thruster Rooms will be fitted with flow-through ventilation systems utilizing 100% outdoor air.
- Compartment volumes are estimated based on the arrangement shown in [4]
- The following ship service generator parameters at 100% MCR are used in the calculations [5].

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• The following parameters from [7] are used to size louvers for the emergency generator:

- Engine Room ventilation calculation assumes:
	- o Maximum temperature rise of 25 °F
	- o Two ship service generators operating at full power.
- Heat radiated from hot exhaust piping is estimated using data from [9], which assumes 1 3/4 inches to 2 3/4 inches of insulation and a 610°F temperature difference between exhaust and ambient.
- Thruster Room ventilation calculation assumes:
	- o Maximum space temperature of 104°F (40°C)
	- o Estimated heat rejection of 4.6 kW per thruster with 30% margin added.
- The Switchboard Room will be fitted with a dedicated air conditioning system, which maintains the Switchboard Room at 95°F at the outdoor air conditions given above, with the propulsion plant operating at full power. This machinery space HVAC system will also serve the EOS. The EOS will be maintained at 74°F.
- The Switchboard Room HVAC System is a chilled water system, consisting of three fan coil units, two serving the switchboard room, and one serving the EOS. Vendor estimated total demand flow is 30 GPM. [8] The fan coils are served by four equally sized seawater cooled chiller sized so that the Switchboard room can be cooled with one chiller out of operation.
- Chilled water piping will be Type K copper pipe.

CONCLUSIONS $5¹$

5.1 Engine Room

A minimum ventilation supply rate of 17,900cfm and ventilation exhaust rate of 15,400 cfm are required to provide cooling and combustion air in the engine room.

Four supply fans, each providing 4,500 cfm, will supply air to the engine room separate ventilation supply plenums on the main deck. Air will exhaust through the exhaust uptake to the 01 deck. The calculated total pressure loss is 2.45 inches $H₂O$. The corresponding static pressure for a 15-inch supply fan is 1.7 inches H_2O .

5.2 Voids

A minimum ventilation rate of 800 cfm is required to maintain six air changes per hour in each void.

One supply fan, providing 1,000 cfm, will supply air to each void via a ventilation supply plenum on the main deck. Air will exhaust through the exhaust plenums on the main deck. The calculated total pressure is 1.30 inches H_2O . The corresponding static pressure for a 12-inch supply fan is 1.21 inches $H₂O$.

5.3 Thruster Rooms

A minimum ventilation rate of 2000 cfm is required to provide cooling air in each thruster room.

One supply fan, providing 2,000 cfm, will supply air to each thruster room via a ventilation supply plenum on the main deck. Air will exhaust through the exhaust plenums on the main deck. The calculated total pressure is 1.3 inches $H₂O$. The corresponding static pressure for a 12-inch supply fan is 1.15 inches H2O.

5.4 Emergency Generator Room

Louvers 39 inches wide by 60 inches tall were selected to keep the emergency generator room restriction below 0.5 inches H_2O .

5.5 Switchboard Room

With the anticipated propulsion electrical equipment, the switchboard room requires $94,980$ BTUH of cooling. The switchboard room will be fitted with two equally sized fan coils, each rated at 48,000 BTUH. As calculated in Appendix E, the EOS requires 25,600 BTUH of cooling. A 36,000 BTUH fan coil is required for the EOS. Chilled water will be supplied by four chillers, each rated at 36,000 BTUH.

The estimated head for each chilled water pump is 94 ft TDH at 30gpm. See the attached PIPE-FLO model for additional details.

REFERENCES

- [1] The Society of Naval Architects and Marine Engineers, "Technical and Research Bulletin 4- 16: Calculations for Merchant Ship Heating, Ventilation and Air Conditioning Design," New York, NY, 08/1980.
- [2] R. L. Harrington, Marine Engineering, Jersey City, NJ: SNAME, 1992.
- [3] ASHRAE, Climatic Design Condtions Hattaras Billy Mitchel AP, NC, USA, ASHRAE, 2009.
- [4] Elliott Bay Design Group, "Double-Ended Azimuth Drive Ferry: Profiles and Deck Arrangements," 18026-200-101-1, Rev -, Seattle, WA, 2018.
- [5] Caterpillar, "EM4133; C18 565kW at 1800 rpm Performance Data," May 24, 2018.
- [6] Caterpillar, LEHM02040-00 C4.4 ACERT Marine Generator Set Package, 2016.
- [7] International Organization for Standardization, "ISO 8861: Shipbuilding Engine-room ventilation in diesel-engined ships - Design requirements and basis of calculations," 1988.

ENGINE ROOM

Engine Room Ventilation

Approach

The minimum required airflow to the Engine Room is calculated based on i) airflow necessary to dissipate heat given off by running equipment, and ii) minimum airflow of 6 air change per hour into the space.

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Engine Room Total Pressure Calculation *Input Variables*

SWITCHBOARD ROOM

Switchboard Room Supply Fan Total Pressure Calculation *Input Variables*

Segment total pressure 7.35

Switchboard Room AHU Ducting *Input Variables*

VOIDS

Void Ventilation

Approach

The minimum required airflow to the Void is calculated based on i) airflow necessary to dissipate heat given off by running equipment, and ii) minimum airflow of 6 air change per hour into the space.

Void Total Pressure Calculation *Input Variables*

THRUSTER ROOMS

Thruster Room Ventilation

Approach

The minimum required airflow to the Thruster Room is calculated based on i) airflow necessary to dissipate heat given off by running equipment, and ii) minimum airflow of 6 air change per hour into the space.

Thruster Room Total Pressure Calculation *Input Variables*

EMERGENCY GENERATOR ROOM

CHILLED WATER PIPING

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Appendix E

Accomodations HVAC

DESCRIPTION $\mathbf{1}$

This appendix presents calculations performed to estimate heating, ventilation, and air conditioning requirements for the DOUBLE-ENDED AZIMUTH DRIVE FERRY design. The resulting heating and cooling loads were used to estimate electrical requirements for the vessel.

PROCEDURE $\overline{2}$

Heating and cooling load calculations were performed using the procedures outlined in [1], modified as follows:

- Passenger Lounge ventilation rate calculated using requirements from [2].
- Low-e glass is specified for the Crew Lounge and Passenger lounge. Glass solar factors (GSF) for these spaces were modified for geographic location and use of low-e coated glass using the RLF method from [3].

3 **REGULATORY FRAMEWORK**

The DOUBLE-ENDED AZIMUTH DRIVE ferries will be US Coast Guard under the provisions of 46 CFR Subchapter H.

GIVEN AND ASSUMED PARAMETERS $\overline{\mathbf{4}}$

- The Pilothouse, Crew Lounge, and Passenger Lounge will be air-conditioned using aircooled split heat pump units.
- The Pilothouse, Crew Lounge, and Passenger Lounge will utilize split heat pumps as primary heat, with electric strip heaters for backup
- The EOS will be air conditioned using a chilled water fan coil as part of the machinery space HVAC system. A heater in the EOS fan coil will provide heat for the space.
- The hold spaces of existing vessels are not heated, except for the Engine Room. The Engine Room will be fitted with two (2) five-kilowatt unit heaters per NCDOT request. A third 5 kilowatt heater will be installed in the Switchboard Room.
- Single pane windows are assumed for the Pilothouse. Low-e coated dual pane insulating glass is assumed for exterior windows in the Crew Lounge and Passenger Lounge. Dual pane A-60 windows are assumed in the EOS.
- Bulkhead insulation in air-conditioned or heated passenger and crew spaces is assumed to have a maximum overall heat transfer coefficient of $U = 0.12 B T U/hr/ft^2/°F$. This corresponds to 3 inches of fiberglass insulation plus 1" stiffener wrap on unlined decks or bulkhead or 2" of fiberglass insulation plus 1" stiffener wrap on decks with furred sheet metal linings, Table 18, Type 92 or Table 17 Type 55 in [4] respectively.
- The following environmental conditions, taken from [5] and [6] were used to determine the HVAC loads:

• Space areas estimated from [7].

CONCLUSIONS 5°

Calculations are presented below. Heating and cooling loads for each space are estimated as follows:

REFERENCES

- [1] The Society of Naval Architects and Marine Engineers, "Technical and Research Bulletin 4- 16: Calculations for Merchant Ship Heating, Ventilation and Air Conditioning Design," New York, NY, 12/2015.
- [2] ASHRAE, ASHRAE Standard 62.1-2016 Ventilation for Acceptable Indoor Air Quality, Atlanta, GA: ASHRAE, 2016.
- [3] AHSRAE, 2009 ASHRAE Handbook: Fundamentals, Atlanta, GA, 2009.
- [4] The Society of Naval Architects and Marine Engineers, Technical and Research Bulletin 4-7: Thermal Insulation Report, New York, NY, 1963.
- [5] AHSRAE, *ASHRAE Handbook Fundamentals, Hatteras Billy Mitchell AP, NC, USA WMO#723139,* 2009.
- [6] NOAA, "Station HCGN7 USCG Station Hatteras, NC Climatic Summary Plots for Sea Temperature," 24 Nov 2015. [Online]. Available: http://www.ndbc.noaa.gov/view_climplot.php?station=hcgn7&meas=st.

[7] EBDG - NC, PLLC, "Double-Ended Azimuth Drive Ferry: Profiles and Deck Arrangements," 18026-200-101-1, Seattle, WA, 2018.

GLASS SOLAR FACTOR FOR LOW-E WINDOWS

Calculation of fenestration load using RLF method from ASHRAE Fundamentals, Ch 17

 CF_{fin} = U(Δt -0.46*DR) + PXI x SGHC x IAC X FF_s $q_{fin} = A \times C F_{fin}$

Given/Assumed

1) Glass u value taken from SNAME T&R 4-16

2) Cooling design temperature, To =95F, Ti = 74

3) DR from ASHRAE data for Hatteras Billy Mitchell AP

4) Latitude is 35.2 degrees, using Peak Irradiance from ASHRAE Fundamentals, Table 10, 35 degrees

5) Assume no interior shading, $IAC = 1$

6) Assume no exterior shading (no shading modifications to PXI)

7) SHGC assumes use of Low-e coated window glass with a SHGC of less than .47

8) FF^s taken from ASHRAE Fundamentas Table 13, single family detached column (see descriptions on page 17.1)

9) Worst combination of PXI and FFs, West Facing, used for calculation

 $Cf_{\text{fen}} = 71.67$ BTUH/sf fenestration cooling load

Use Cf_{fen}= 75 BTUH/sf

ROOM LOAD CALCULATIONS

*** This calculation sheet assumes sun on forward and port sides**

*** This calculation sheet assumes sun on starboard side only**

*** This calculation sheet assumes sun on starboard side only**

Appendix F

Fire Main System, Dwg. 18026-200-521-1

DESCRIPTION $\mathbf{1}$

This appendix documents the calculations used in designing the fire main piping system. These calculations establish pump capacity in accordance with regulatory requirements and minimum pipe sizes based upon nominal velocity limits. Estimated losses through the system piping are calculated to establish the total dynamic head (TDH) and net positive suction head (NPSH) requirements for the fire pumps.

$\overline{2}$ **PROCEDURE**

Calculations are presented in the following sequence:

- Fire main pipe nominal velocity calculations
- Fire pump TDH and NPSH calculations

Minimum fire pump capacity, pressure, and nozzle size is based upon the requirements found in

46 CFR Subchapter H, [1].

Frictional losses through the piping system are calculated by constructing a model using PIPE-FLO Professional software utilizing the Darcy-Weisbach method.

GIVEN AND ASSUMED PARAMETERS $3¹$

- Fire main system is to be constructed of Class 200 copper nickel pipe.
- Piping system lengths, routing, fittings, etc. are estimated based on the Profiles and Arrangements, and Fire Main System Schematic [2] [3].
- In accordance with [1], the required pressure at the two most remote fire hydrants is 50 psi and the required nozzle orifice size is 5/8".
- The theoretical discharge from a 5/8" orifice at 50 psi is assumed to be 82 gpm, or 164 gpm for two nozzles.
- To maintain fleet commonality, the client prefers the fire pump be a Goulds $3796 \times 2 \times 2 \cdot 10$, 3550 rpm, 8.5 inch impeller.
- The following elevations above baseline are assumed for the system calculation

• The fluid medium is seawater with the following properties:

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FORMULAS $\overline{\mathbf{4}}$

(not used)

5 **CALCULATIONS**

5.1 Pipe Size Calculations

Pipe sizes are based on the nominal velocity limits found in Marine Engineering, Chapter 20, Table 3, [4]. The following table shows the flow rates of fuel occurring in the system, and the resulting pipe sizes.

Pipe Segment	Flow	Pipe	Schedule	d, ID	Design Velocity		
	Rate	Size			Nominal	Limit	
	gpm	(NPS)		m	$\mathrm{ft/s}$	$\mathrm{ft/s}$	ft/s
Firemain Suction	164	3	CL 200	3.310	$3.0 \sqrt{d}$ 5.46	15	6.11
Firemain Discharge	164	3	CL 200	3.310	$5.0 \sqrt{d}$ 9.10	15	6.11
Firemain Branch (1 hydrant)	82	11/2	CL 200	1.756	$5.0 \sqrt{d}$ 6.63	15	10.86
Firemain Branch (2 hydrants)	164	$\mathbf{2}$	CL 200	2.209	$5.0 \sqrt{d}$ 7.43	15	13.73
Firemain Overboard	164	3	CL 200	3.310	$5.0 \sqrt{d}$ 9.10	15	6.11

Table 5-1: Nominal Pipe Velocity

Note that the velocity limit of 12 feet per second is exceeded in a 2" line with a 164 gpm flowrate, and it is likely this limit will be exceeded in some cases with the client's preferred fire pump. This is acceptable; however, as the fire system is infrequently used and minimal pipe wear is anticipated over the life of the vessel.

5.2 Fire Pump TDH and NPSHa Calculation

From the enclosed system model, the minimum fire main pump operating point is 165 gpm at a total dynamic head of 182 feet H2O while discharging from the two most remote fire stations with a pitot pressure of 50 psig. The system provides NPSH of 31.5 feet H2O. The client's preferred pump is also modeled. This pump will provide a 192 gpm at 237 feet H2O while discharging from the two most remote fire stations; this exceeds the minimum regulatory requirements.

REFERENCES 6

- [1] USCG, "46 CFR, Chapter I, Subchapter H, Part 76 Fire Protection Equipment," 5/16/2018.
- [2] Elliott Bay Design Group, "NCDOT Double Ended Azimuth Drive Ferry: Profiles and Arrangements," 18026-200-101-1, Seattle, WA.
- [3] Elliott Bay Design Group, "NCDOT Double Ended Azimuth Drive Ferry: Fire Main System Schematic," 18026-200-521-1, Seattle, WA.

[4] R. L. Harrington, Marine Engineering, Jersey City, NJ: SNAME, 1992.

FIRE MAIN PIPE-FLO MODEL

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NCDOT DOUBLE ENDED AZIMUTH DRIVE VERRY 8/10/18

NCDOT DOUBLE ENDED AZIMUTH DRIVE VERRY 8/10/18

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Appendix G

Sanitary Drain System, Dwg. 18026-200-528-1

$\mathbf{1}$ **DESCRIPTION**

This appendix documents the calculations used in selecting the zero discharge pump for the sanitary drains system. These calculations are used to establish minimum pipe sizes based upon nominal velocity limits and to identify steady state frictional losses to establish the total dynamic head (TDH) and net positive suction head (NPSH) requirements for the zero discharge pumps.

$\overline{2}$ **PROCEDURE**

Calculations are presented in the following sequence:

- Pipe size calculations
- Zero discharge pump TDH and NPSH calculations

Frictional losses through the piping system are calculated by constructing a model using PIPE-FLO Professional software utilizing the Darcy Weisbach method.

$3¹$ **GIVEN AND ASSUMED PARAMETERS**

- The zero discharge system is to be constructed of Schedule 80 CPVC pipe.
- Piping system lengths, routing, fittings, etc. are estimated based on the Profiles and Arrangements, and Sanitary Drains Schematic, [1] [2].
- A pressure of 10 psig is assumed at the zero discharge connection on the Main Deck.
- The zero discharge tanks are 2 feet above baseline, the pump suction and discharge are 2.5 feet above baseline, and the zero discharge connection on the Main Deck is assumed to be 11 feet above baseline, [1].
- The fluid used for all calculations is fresh water with the following properties:

- A single pump shall be capable of emptying the two 500 gallon tanks in 10 minutes or less, so the minimum flowrate is 100 gpm.
- To maintain fleet commonality the client prefers an MP Pumps Flomax 10 $2x2$, 3450 rpm with a 5.5 inch impeller for the zero discharge pump.

$\overline{\mathbf{4}}$ **FORMULAS**

(not used)

$\overline{\mathbf{5}}$ **CALCULATIONS**

5.1 Pipe Size Calculations

Pipe sizes are based on the nominal velocity limits found in Marine Engineering, Chapter 20, Table 3, [3]. The following table shows the flow rates of zero discharge effluent occurring in the system, and the resulting pipe sizes.

Pipe Segment	Flow	Pipe	Schedule	d, ID	Design Velocity		
	Rate	Size			Nominal	Limit	
	gpm	(NPS)		m	$\mathrm{ft/s}$	ft/s	ft/s
Pump Suction	100		SCH 80	1.939	$3.0 \sqrt{d}$ 4.18	15	10.86
Pump Discharge	100 ₁		SCH 80	1.939	$5.0 \sqrt{d}$ 6.96	15	10.86

Table 5-1: Zero Discharge System Pipe Sizes and Velocities

While the expected velocities exceed the nominal velocities , the expected velocities are still well below the maxium limit.

5.2 Zero Discharge Pump TDH and NPSH Calculation

From the enclosed system model, the zero discharge pump shall be sized for 105 gpm at 75 feet H2O TDH. The system has 30.5 NPSH available. The system model considers the client's preferred pump; this operating point is on the pump curve of the client's preferred pump.

REFERENCES 6

- [1] Elliott Bay Design Group, "NCDOT Double Ended Azimuth Drive Ferry: Profiles and Arrangements," 18026-200-101-1, Seattle, WA.
- [2] Elliott Bay Design Group, "NCDOT Double Ended Azimuth Drive Ferry: Sanitary Drains Schematic," 18026-200-528-1, Seattle, WA.
- [3] R. L. Harrington, Marine Engineering, Jersey City, NJ: SNAME, 1992.

ZERO DISCHARGE PIPE-FLO MODEL

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NCDOT DOUBLE ENDED AZIMUTH DRIVE FERRY 8/10/18

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1 x Swing Check - Vertical

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Appendix H

Bilge System, Dwg. 18026-200-529-1
DESCRIPTION $\mathbf{1}$

This appendix documents the calculations used in designing the bilge piping system. These calculations establish minimum bilge pipe sizes and required pump capacity in accordance with regulatory requirements. Estimated losses through the bilge system are calculated to establish the total dynamic head (TDH) and net positive suction head (NPSH) requirements for the bilge pump.

PROCEDURE $\mathbf{2}$

Calculations are presented in the following sequence:

- Bilge pipe size and pump capacity calculations
- Bilge pipe nominal velocity calculations
- Bilge pump TDH and NPSH calculation

Bilge pipe size and capacity are based on the regulatory requirements found in [1].

Frictional losses through the piping system are calculated by constructing a model using PIPE-FLO Professional software utilizing the Darcy Weisbach method.

GIVEN AND ASSUMED PARAMETERS $3¹$

- Bilge system is to be constructed of Schedule 80 carbon steel pipe.
- Piping system lengths, routing, fittings, etc. are estimated based on the Profiles and Arrangements, and Bilge and Ballast System Schematic [2] [3].
- The fluid medium is seawater with the following properties:

The following elevations above baseline are assumed for the system calculation

$\overline{\mathbf{4}}$ **FORMULAS**

(not used)

$\overline{\mathbf{5}}$ **CALCULATIONS**

5.1 Bilge Pipe Size and Pump Capacity Calculation

From the attached spreadsheet, the minimum internal bilge main diameter is 3.02 inches ± 0.25 inches. 3 inch schedule 80 steel pipe has an inside diameter of 2.9 inches, and is minimum acceptable size for the bilge main. The minimum bilge piping diameter for the hull compartments ranges from 2.0 inches to 2.45 inches. The acceptable minimum NPS pipe size ranges from 2 inches to 2.5 inches schedule 80 steel pipe.

The minimum bilge pump capacity to maintain a nominal velocity of 400 feet per minute in the bilge main is 149 gpm.

5.2 Bilge Pipe Nominal Velocity Calculation

Pipe sizes are checked against the nominal velocity limits found in Marine Engineering, Chapter 20, Table 3, [4]. The following table shows the flow rates of bilge water occurring in the system.

Pipe Segment	Flow	Pipe	Schedule	d, ID	Design Velocity		
	Rate	Size			Nominal	Limit	
	gpm	(NPS)		\overline{m}	ft/s	ft/s	ft/s
Bilge Main	149		SCH 80	2.900	3.0 Vd 5.11	15	7.24
Engine Room Bilge Branch	149	21/2	SCH 80	2.323	$3.0 \sqrt{d}$ 4.57	15	11.28
All Other Bilge Branches	125 ₁	$\overline{\mathbf{2}}$	SCH 80	1.939	4.18 3.0 Vd	15	13.58
Overboard Discharge	149		SCH 80	2.900	$5.0 \sqrt{d}$ 8.51	15	7.24

Table 5-1: Nominal Pipe Velocity

Note that it is necessary to throttle the pump discharge when pumping individual compartments beyond the engine room.

5.3 Bilge Pump TDH and NPSH Calculation

From the enclosed system model, the bilge pump is required to produce about 40 feet H2O TDH at the required flowrate of 148 gpm. Calculated NPSH available for the bilge system ranges from about 6 feet to 12 feet in the SWBD room and engine room, respectively.

Note that it is necessary to throttle the pump discharge when pumping an individual compartment beyond the Engine Room to prevent pump cavitation.

REFERENCES 6

- [1] USCG, "46 CFR, Chapter I, Subchapter F, 56.50-50," 5/16/18.
- [2] Elliott Bay Design Group, "NCDOT Z-Drive Ferry: Profiles and Arrangements," 18026-200- 101-1, Seattle, WA.
- [3] Elliott Bay Design Group, "NCDOT Z-Drive Ferry: Bilge and Ballast System Schematic,"

18026-200-529-1, Seattle, WA.

[4] R. L. Harrington, Marine Engineering, Jersey City, NJ: SNAME, 1992.

BILGE PIPE SIZING AND PUMP CAPACITY BILGE SYSTEM SIZING CALCULATIONS

APPROACH

Size bilge pipes and bilge pumps per the requirements of 46CFR Subchapter F, 56.50-50.

ASSUMPTIONS

1. Unit System used in this calculation: US.

2. Bilge piping is schedule 80.

 $B =$

CALCULATION OF PIPESIZEFOR SUCTION TO EACH MAIN BILGEPUMP 46CFR 56.50-50(d)(1)

CALCULATION OF PIPESIZEFOR SUCTION OF EACH BRANCH 46CFR 56.50-50(d)(2)

$$
d = 1 + \sqrt{\frac{c \times (B + D)}{1500}}
$$

BILGE SYSTEM PIPE-FLO MODEL

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Appendix I

Ballast System, Dwg. 18026-200-529-1

$\mathbf{1}$ **DESCRIPTION**

This appendix documents the calculations used in designing the ballast piping system. These calculations establish ballast pipe sizes and required pump capacity based upon a desired ballast loading and unloading rate. Estimated losses through the ballast system are calculated to establish the total dynamic head (TDH) and net positive suction head (NPSH) requirements for the bilge pump.

PROCEDURE $\mathbf{2}$

Calculations are presented in the following sequence:

- Ballast pipe nominal velocity calculations
- Ballast pump TDH and NPSH calculation

Frictional losses through the piping system are calculated by constructing a model using PIPE-FLO Professional software utilizing the Darcy Weisbach method.

$3¹$ **GIVEN AND ASSUMED PARAMETERS**

- Ballast system is to be constructed of copper nickel class 200.
- Piping system lengths, routing, fittings, etc. are estimated based on the Profiles and Arrangements, and Bilge and Ballast System Schematic [1] [2].
- The fluid medium is seawater with the following properties:

The following elevations above baseline are assumed for the system calculation

- Desired ballast loading rate is 400 gpm split between two pumps. To calculate the worst case loading condition, the ballast tanks are approaching full capacity.
- Desired ballast unloading rate is 400 gpm split between two pumps. To calculate the worst case unloading condition, the ballast tanks are approaching empty.
- Desired ballast trimming rate is 200 gpm with one pump. To calculate the worst case trimming condition, the source tank is approaching empty and the destination tank is approaching full capacity.

$\overline{\mathbf{4}}$ **FORMULAS**

(not used)

$\overline{\mathbf{5}}$ **CALCULATIONS**

5.1 Ballast Pipe Size and Nominal Velocity Calculation

Pipe sizes are based on the nominal velocity limits found in Marine Engineering, Chapter 20, Table 3, [3]. The following table shows the flow rates of sea water occurring in the system, and the resulting pipe sizes.

Pipe Segment	Flow	Pipe	Schedule	d, ID	Design Velocity			
	Rate	Size			Nominal		Limit	
	gpm	(NPS)		\mathbf{m}	$\mathrm{ft/s}$		ft/s	$\mathrm{ft/s}$
Ballast Pump Suction (1 pump)	200		CL 200	4.282	3.0 Vd	6.21	12	4.46
Ballast Pump Discharge (1 pump)	200	4	CL 200	4.282	$5.0 \sqrt{d}$	10.35	12	4.46
Ballast Loading (1 tank)	100	$\mathbf{3}$	CL200	3.310	$5.0 \sqrt{d}$	9.10	12	3.73
Ballast Unloading (1 tank)	100	3	CL 200	3.310	3.0 Vd	5.46	12	3.73
Ballast Main Suction (2 pumps)	400	4	CL200	4.282	3.0 Vd	6.21	12	8.91
Ballast Main Discharge (2 pumps)	400		CL 200	4.282	$5.0 \sqrt{d}$	10.35	12	8.91

Table 5-1: Nominal Pipe Velocity

5.2 Ballast Pump TDH and NPSH Calculations

Three different conditions are considered in the attached system model.

- Loading ballast water with 2 pumps in operation
- Unloading ballast water with 2 pumps in operation
- Trimming ballast water with 1 pump in operation

For loading ballast water, the enclosed system model predicts that each ballast pump is required to produce 13.4 feet H2O TDH at the required flowrate of 200 gpm. Calculated NPSH available for each ballast pump in this condition is approximately 32 feet H2O.

For unloading ballast water, the enclosed system model predicts that each ballast pump is required to produce 17 feet H2O TDH at the required flowrate of 200 gpm. Calculated NPSH available for each ballast pump in this condition is approximately 28 feet H2O.

For trimming ballast water from one end to the other, the enclosed system model predicts that the ballast pump is required to produce 17.3 feet H2O TDH at the required flowrate of 200 gpm. Calculated NPSH available for the ballast pump in this condition is approximately 28 feet H2O.

The Ballast pumps are selected for 200 gpm at 20 feet H2O TDH.

6 **REFERENCES**

- [1] Elliott Bay Design Group, "NCDOT Z-Drive Ferry: Profiles and Arrangements," 18026-200- 101-1, Seattle, WA.
- [2] Elliott Bay Design Group, "NCDOT Z-Drive Ferry: Bilge and Ballast System Schematic," 18026-200-529-1, Seattle, WA.
- [3] R. L. Harrington, Marine Engineering, Jersey City, NJ: SNAME, 1992.

BALLAST SYSTEM PIPE-FLO MODEL

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Appendix J

Lube Oil and Waste Oil System, Dwg. 18026-200-529-2

$\mathbf{1}$ **DESCRIPTION**

This appendix documents the calculations used in designing the lube oil and waste oil system. These calculations are used to identify steady state frictional losses and to verify compliance with manufacturer-stated performance limits.

PROCEDURE $\overline{2}$

Calculations are presented in the following sequence:

- Pipe size calculations
- Frictional loss and pump ΔP calculations

Frictional losses through the piping system are calculated by constructing a model using PIPE-FLO Professional software utilizing the Darcy-Weisbach method.

GIVEN AND ASSUMED PARAMETERS $\overline{\mathbf{3}}$

- The Waste Oil system is assumed to be constructed of Schedule 80 carbon steel pipe.
- Piping system lengths, routing, fittings, etc. are estimated based on the Profiles and Arrangements, and Lube Oil and Waste Oil Schematic [1] [2].
- A pressure of 10 psig is assumed at the waste oil discharge on the Main Deck.
- The waste oil tank is 2 feet above baseline, the pump suction and discharge are 4 feet above baseline, and the waste oil discharge on the Main Deck is assumed to be 12 feet above baseline, [1].
- The fluid used for all calculations is SAE 30 Lube Oil with the following properties:

FORMULAS $\overline{\mathbf{4}}$

(not used)

5 **CALCULATIONS**

5.1 Pipe Size Calculations

Pipe sizes are based on the nominal velocity limits found in Marine Engineering, Chapter 20, Table 3, [3]. The following table shows the flow rates of lube oil occurring in the system, and the resulting pipe sizes.

Pipe Segment	Flow	Pipe	Schedule	d, ID	Design Velocity			
	Rate	Size			Nominal		Limit	
	gpm	(NPS)		m	$\mathrm{ft/s}$		ft/s	ft/s
Pump Suction		1/2	SCH 80	.500 ¹	$1.0 \sqrt{d}$	1.22		2.90
Pump Discharge			SCH 80	.500	$2.0 \sqrt{d}$	2.45	h	2.90

Table 5-1: Waste Oil System Pipe Sizes and Velocities

5.2 Frictional Loss and Pump ΔP Calculation

From the enclosed system model, the waste oil pump must provide at a minimum 20 psig when operating at a flow rate of 16 gpm.

6 **REFERENCES**

- [1] Elliott Bay Design Group, "NCDOT Double Ended Azimuth Drive Ferry: Profiles and Arrangements," 18026-200-101-1, Seattle, WA.
- [2] Elliott Bay Design Group, "NCDOT Double Ended Azimuth Drive Ferry: Lube Oil and Waste Oil Piping Schematic," 18026-200-529-2, Seattle, WA.
- [3] R. L. Harrington, Marine Engineering, Jersey City, NJ: SNAME, 1992.

WASTE OIL SYSTEM PIPE-FLO MODEL

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2 x Tee - Flow Thru Branch
1 x Tee - Flow Thru Run

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Appendix K

Potable and Sanitary Water System, Dwg. 18026-200-533-1

DESCRIPTION $\mathbf{1}$

This appendix documents the calculations used in designing the potable water system. These calculations were used to determine flow demand and pipe sizes for the potable water piping and to size the potable water pressure tank and pump.

PROCEDURE $\overline{2}$

Calculations are presented in the following sequence:

- Demand water supply calculation
- Potable water tank sizing calculation
- Potable water pump requirements

System sizing is based on the guidance found in Appendix A of [1] and estimated usage factors.

Water system pressure tank is sized in accordance with [2] .

3 **GIVEN AND ASSUMED PARAMETERS**

- The potable water system is Schedule 40 at the tank connection and Copper Seamless Hard Drawn Type K beyond the tank shut off valve.
- The system is supplied with two pumps. One pump will pressurize the system in normal operation and the second pump will be on standby. Pump will cycle on at 40 psig and off at 60 psig.
- Water closets flush using 1.28 gallons per flush, low-volume, flush-o-meter valves.
- Fixture count, system pipe lengths, routing and fittings are estimated based on the Profiles and Deck Arrangements [3] and Potable and Sanitary Water Piping Schematic [4].
- The highest potable and sanitary water outlet is on the bridge deck at the window washing system, 38 feet ABL.
- The pump inlet and outlet elevation is 2.5 feet.
- Required residual pressure at the highest outlet is 15 psi.
- The water tank is assumed to have 5 feet of water.
- Friction losses from the outlet of the pump to the highest outlet of the system are assumed to be 20 psi.

FORMULAS $\overline{\mathbf{4}}$

The following formulas, taken from [2], are used to estimate the size of the potable water pressure tank:

> Eq 1: Supplemental Drawdown (gal) = (Peak demand (gpm) − pump capacity(gpm)) \times Peak Demand Time (min)

Eq 2: Total Pressure Tank Volume $=$ Minimum Drawdown + Supplemental Drawdown Acceptance Factor

Eq 3: Acceptance Factor =
$$
1 - \frac{P1 \left(\text{tank precharge} \right) + 14.7}{P2 \left(\text{cutoff} \right) + 14.7}
$$

CALCULATIONS $5⁵$

5.1 Demand Water Supply Calculations

Table 5-1: Water Supply

Demand flow from Chart A-3, Line 1 is 40 gpm, from [1]. However, varying the usage factors to simulate different system loads results in instantaneous demands ranging from 8 to 16 gpm. A pump sized for flows in this range with a maximum head of 50 psi coupled with a suitable pressure tank to prevent pump cycling will suffice.

5.2 Pressure Tank Sizing Calculation

Per [2], Table IV.1.2, the minimum draw-down for an 8 gpm pump is 8 gallons. The minimum pressure tank volume is calculated as follows:

	Item	Qty		Note / Reference
(1)	Pump Capacity		8 gpm	
(2)	Minimum Drawdown		gallons	
(4)	Peak Demand Estimation		16 gpm	
(5)	Peak Demand Time		0.08 minutes	
(6)	P1 pressure tank precharge		40 psi	
(7)	P ₂ cutout pressure		60 psi	
(9)	Supplemental Drawdown		0.64 gallons	Eq. 1: $[(4)-(1)]*(5)$
(10)	Total Required Drawdown		8.64 gallons	Eq. 2: $(2) + (9)$
(11)	Acceptance Factor	0.27		Eq. 3: 1 - [[$(6) + 14.7$]/[$(7) + 14.7$]]
(12)	Total Calculated Tank Size		32 gallons	Eq. 4 $(10)/(11)$

Table 5-2: Pressure Tank Sizing

5.3 Friction Loss

In order to determine the pump head requirements, the piping system between the potable water supply tank and potable water pressure tank are modeled using Pipe-Flo Professional 15 utilizing the Darcy-Weisbach friction loss method.

The pressure tank is set to 50 psi to represent 15 psi at the highest potable water outlet and 20 psi friction losses in the piping.

Based upon the attached model, at a flow rate of 16 gpm the velocity in the water suction line is 3.0 ft/sec, head loss is 1.5 ft, and the pump total head requirement is 113.5 ft.

The potable water pump will operate between 40 and 60 psi against a pressure tank, and should be selected with a shutoff head exceeding 60 psi and an NPSHr well below the NPSHa of the system for the flow rate at 40 psi. System NPSHa at various flow rates are as follows:

Flow Rate, (gpm)	System NPSHa, ft	Velocity, ft/s						
	38.23	1.05						
	37.94	2.11						
ר ו	37.49	3.16						
16	36.86	4.22						
	36.08	5.27						

Table 5-3: System NPSHa

REFERENCES 6

- [1] IAPMO/ANSI UPC 1 2009: Uniform Plumbing Code, Ontario, CA: International Association of Plumbing and Mechanical Officials, 04/2009.
- [2] Water Systems Council, Wellcare Information For You About Sizing a Pressure Tank, Washington, DC.
- [3] EBDG, "18026-200-101-1 Profiles and Deck Arrangements," 2018.
- [4] EBDG, "18026-200-533-1 Potable and Sanitary Water Piping Schematic," 2018.

Potable and Sanitary Water Pipe-Flo Model

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