Large Marine Engine Technology Evaluation

Final Report

United States
Environmental Protection
Agency

Large Marine Engine Technology Evaluation

Final Report

Office of Transportation and Air Quality
U.S. Environmental Protection Agency

Prepared for EPA by
Southwest Research Institute (SwRI)
EPA Contract 68HERC20D0014
Task Order 68HERC24F0014

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United States
Environmental Protection
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LARGE MARINE ENGINE TECHNOLOGY EVALUATION

EPA CONTRACT 68HERC20D0014
TASK ORDER 68HERC24F0432
OMB CLEARANCE NUMBER 2030-0005

FINAL REPORT

SwRI® Project Number 03.28987

Prepared for:

U.S. Environmental Protection Agency

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February 28, 2025

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TABLE OF CONTENTS

Page

LIST OF FIGURES

LIST OF TABLES

Table **Page**

LIST OF ACRONYMS

ABS	Ammonium Bisulfate
ACD	Auxiliary Control Device
AIS	Automatic Identification System
AFR	Air to Fuel Ratio
ANR	Ammonia to NO_X Ratio
ASC	Ammonia Slip Catalyst
AT	Aftertreatment
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
CAPEX	Capital Expense
CARB	California Air Resources Board
CFR	Code of Federal Regulations
CO	Carbon Monoxide
CO2	Carbon Dioxide
COV	Coefficient of Variation
CWF	Carbon Weight Fraction
DCU	Decomposition Unit
DEF	Diesel Exhaust Fluid
deSO_X	Desulfation
DOC	Diesel Oxidation Catalyst
DPF	Diesel Particulate Filter
ECA	Emission Control Area
ECM	Electronic Control Module
EEVO	Early Exhaust Valve Opening
EGR	Exhaust Gas Recirculation
EGT	Exhaust Gas Temperature
EHC	Electrically Heated Catalyst
EO	Engine Out
EPA	Environmental Protection Agency
HD	Heavy Duty
IMO	International Marine Organization
IMT	Intake Manifold Temperature
MARPOL	International Convention for the Prevention of Pollution from Ships
MCR	Maximum Continuous Rating
MY	Model Year
NOAA	National Oceanic and Atmospheric Administration
NO_X	Oxides of Nitrogen
NTE	Not-to-Exceed
OGV	Ocean-going Vessel
OPEX	Operating Expense
PGM	Platinum Group Metals
PM	Particulate Matter
RAM	Random Access Memory
SCR	Selective Catalytic Reduction (ammonia-based)
SOI	Start of Injection

SwRI	Southwest Research Institute
TDC	Top dead center
THC	Total Hydrocarbons
TM	Thermal Management
ULSD	Ultra Low Sulfur Diesel
USCG	United State Coast Guard
VCR	Variable Compression Ratio
VVA	Variable Valve Actuation

ABSTRACT

MARPOL Annex IV Regulation 13 requires Tier III NO_X reduction technology to be used within NO_X Emission Control Area (ECA). Many Tier III Category 3 (≥ 30L per cylinder) engines utilize selective catalytic reduction (SCR) to reduce NO_X. The E2 and E3 propulsion test cycles only test engine emissions down to 25% power and the program allows Administrations to approve use of auxiliary control devices (ACDs) that permit disengaging the SCR system when the exhaust gas temperature drops below the SCR minimum operating temperature. This typically occurs below 25% power, if the engine was not calibrated with thermal management strategies to maintain the exhaust gas temperature, which is always the case. Because vessels may choose to travel more slowly within ECAs due to economic or operational considerations, NO_X reduction benefits may be curtailed due to frequent operation at less than 25% power.

This study evaluated engine load profiles within the North American ECA for Category 3 ocean-going vessels (OGVs) and assessed the feasibility of extending SCR operation below 25% power. The findings include: 1) A large percentage of operation in the ECA is less than 25% power, 2) Existing Category 3 engine technology can maintain adequate exhaust gas temperature for SCR operation below 25% power, and 3) NO_X emissions within ECAs by Tier III Category 3 vessels can be significantly reduced by utilizing SCR at less than 25% power.

EXECUTIVE SUMMARY

As part of an ongoing effort to assess the NO_X ECAs and how they could be improved, this study investigated the following Tier III-related issues. First, whether the efficiency of SCR emission control technology on Category 3 vessels can be improved, both underway and while operating at low load, based on the current engine certification process. Second, an evaluation of ship operating profiles in the ECAs irrespective of engine category, specifically how much time is spent at low load operation and, using that information, how the certification test cycles could be revised to incorporate that operation, and if exhaust gas temperature can be maintained to extend SCR operation to 10% power or lower.

SwRI engaged in a substantial effort to develop a clear picture of the operating profile of real Category 3 vessels operating in the North American ECA. This was a key input for examining the potential for additional NO_X reductions. The resulting operating histogram is shown in Figure 1 below. These values indicate that 42% of all operation in the ECA occurs below the 25% load point, and 75% of all operation in the ECA occurs below the 50% load point.

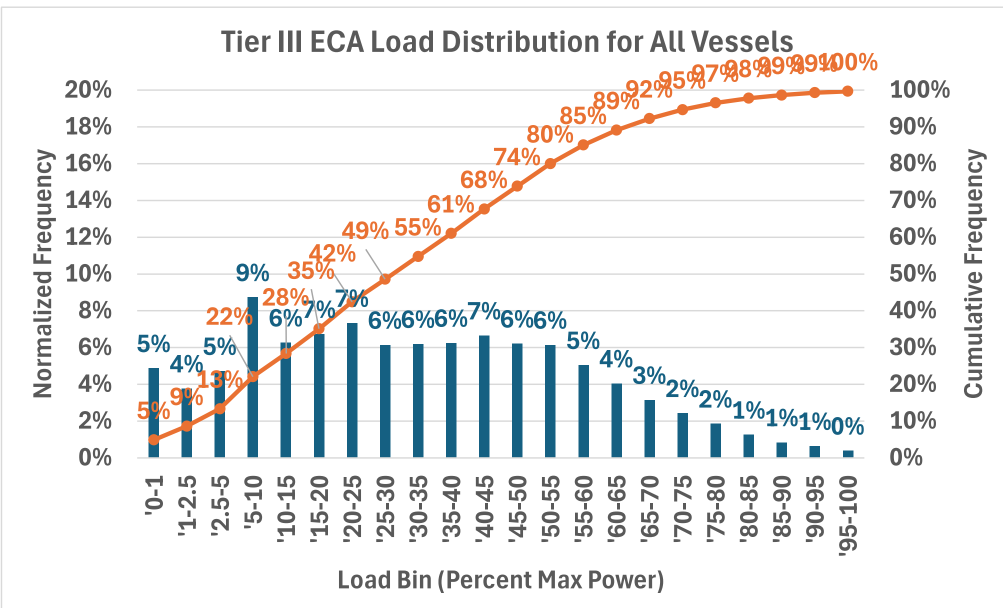


FIGURE 1. LOAD FREQUENCY DISTRIBUTION FOR TIER III VESSELS IN THE NORTH AMERICAN ECA

Because the current Tier III standard does not result in substantial NO_X emission control below the 25% load point, a significant operation in the ECA is happening at loads where there is not much NO_X control. Figure 2 shows the resulting NO_X mass histogram when the load histogram above is combined with NO_X emission rate projections for a typical Tier III vessel. It shows that 47% of the NO_X mass emitted in the ECA occurs at loads below 25%. It should be noted that this projection assumes that there is some NO_X control in the 20-25% load bin due to the need for some operating margin below 25% to ensure reliable compliance with Tier III requirements.

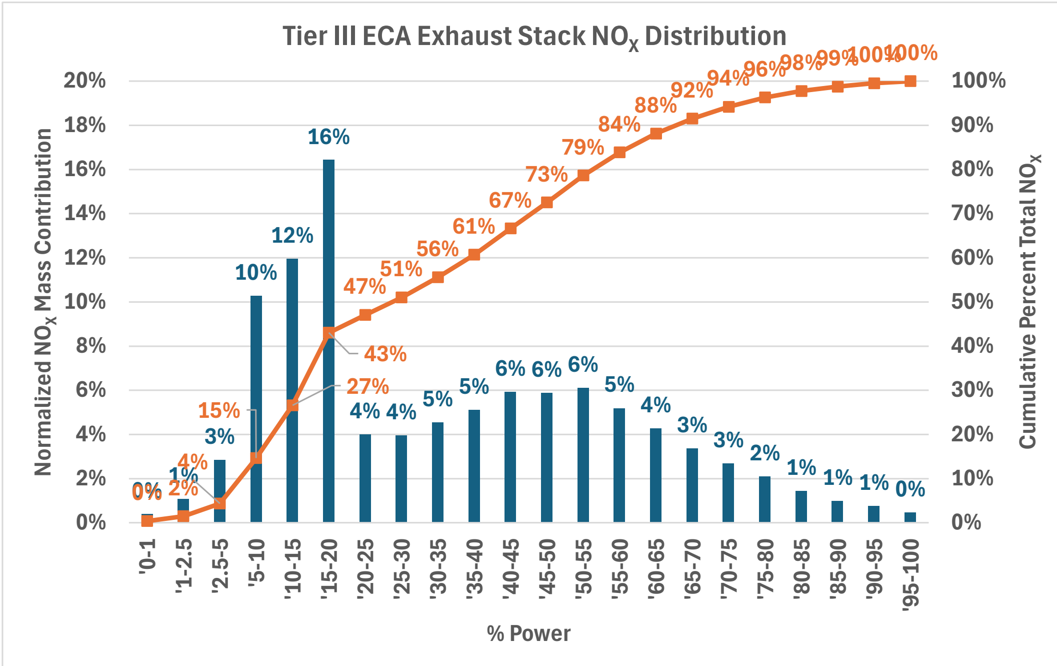


FIGURE 2. NO_X MASS DISTRIBUTION BY LOAD BIN FOR TIER III VESSELS IN THE NORTH AMERICAN ECA

SwRI examined a number of technology scenarios based around the application of high pressure (pre-turbine) vanadium SCR and various levels of thermal management. Both of these are technologies that are currently used on the main propulsion engines of many Tier III vessels today. The scenarios examined various combinations of extending NO_X control to lower loads and pushing for additional NO_X conversion in the current Tier III control range above 25% load. These scenarios, and their resulting NO_X reductions are shown in Table 1 below.

TABLE 1. SUMMARY OF NO_X IMPROVEMENT TECHNOLOGY SCENARIOS EXAMINED

Scenario	Details	Minimum Load	NOx Reduction	Standard Notes
1	Current Mode Reweights	25%	9%	Reweigh to ECA Duty Cycle
2	Max SCR, Best Temperatures	5%	53%	Add 10% Mode and 5% Mode Cap, Tighten Standard Limits
3	Max SCR, Intermediate Temperatures	10%	45%	Add 10% Mode, Tighten Standard Limits
4	Conservative SCR, Intermediate Temperatures	10%	23%	Add 10% Mode
5	Conservative SCR, Best Temperatures	5%	30%	Add 10% Mode and 5% Mode Cap

Scenario 1 is included for comparisons, but generally indicates the simple measure of re-weighting the test cycle modes to more accurately reflect ECA operation will achieve only modest NO_X reductions without additional changes. All of the remaining scenarios involve extending the range of NO_X control for the test cycle down to at least the 10% load point, which would likely require adding a new test cycle mode at 10% (though it may be possible to achieve the same result using carefully designed mode caps). As noted earlier, existing literature indicates that this is likely achievable using technologies already deployed on many Tier III OGVs. It should be noted the scenarios 2 and 5, which extend NO_X control down to the 5% load point will likely require some additional thermal management technology, such as intake air heating or variable valve actuation, to achieve. Technology scenarios marked as “Max SCR”, include reductions in the NO_X standard that would require improving high load SCR NO_X conversion efficiency to 88% from the current level of roughly 80%.

Based on the scenarios given in Table 1, it appears that NO_X reductions on the order of 50% may be available using a reasonable combination of technologies, many of which are already applied on Tier III vessels today. All of these technology scenarios are considered feasible for application on Category 3 2-stroke engines, though some, as noted, would require the fitting of additional technologies beyond what are currently seen today on Tier III vessels.

1.0 INTRODUCTION AND BACKROUND

The Environmental Protection Agency (EPA) is conducting research to assess the potential for further reductions in NO_X emissions from large Category 3 marine diesel engines (≥ 30 l/cyl displacement) used on ocean-going vessels (OGVs).

This study examines OGV operation in the North American NO_X Emission Control Area (ECA), designated through amendment to Annex VI of the International Convention for the Prevention of Pollution from Ships (MARPOL). Emissions from an engine installed on a ship with a keel laid date beginning January 1, 2016 may not exceed the Annex VI Regulation 13 Tier III NO_X limits while that engine is operating in an ECA. The Tier III NO_X emission limits are as follows, where n = rated engine speed (crankshaft revolutions per minute)[1]:

- 3.4 g/kWh when n is less than 130 rpm;
- $9 \cdot n^{(-0.2)}$ g/kWh when n is 130 or more but less than 2,000 rpm;
- 2.0 g/kWh when n is 2,000 rpm or more.

Marine diesel propulsion engines are certified to these NO_X limits using the standard ISO marine test cycles: E3 test cycle, for a fixed pitch propeller propulsion engine, or a propeller-law operated non-propulsion engine; or E2 test cycle for a propulsion engine that does not operate with a fixed pitch propeller. Examples of applications that use E2 include those used in diesel-electric installations or operated with a controllable-pitch propeller. The E2 test cycle has emission measurement mode points at a constant speed of 100% and load points at 25%, 50%, 75% and 100% of the maximum continuous power rating (MCR) specified on the engine nameplate. The E3 test cycle has emission measurement mode points at the variable speeds and loads of 100% and 100%, 91% and 75%, 80% and 50%, and 63% and 25% of the MCR. The NO_X emissions at these mode points are weighted using the test cycle modal weighting factors and then summed for comparison to the emission limit. The test cycles also include a mode cap that limits the modal emission values to 1.5 times the Annex VI Regulation 13 Tier III NO_X standard. The mode cap was added to the test cycles to ensure that the emission control system stays functional down to at least the 25% load mode point.

Over the last 10 years, increases in fuel prices have led ship operators to reduce vessel speed (slow steaming) to reduce ship fuel consumption and operating costs. In addition, some coastal areas have Vessel Speed Reduction (VSR) zones to reduce air emissions or address other environmental or safety concerns. For example, a VSR established in 2001 at the ports of Los Angeles and Long Beach in California to reduce emissions set a speed limit between 10 and 15 knots, depending on ship type, within 20 to 40 nautical miles from shore. Thus, widespread slow steaming means that ship operation may not be accurately portrayed by the marine diesel engine propulsion certification test cycles (E2, E3).

[1] See MARPOL Annex VI, Regulation 13.5.1.1.

A U.S. government paper submitted to a recent meeting of the IMO Sub-Committee on Pollution Prevention and Response (PPR) indicated that due to these two conditions, operating at slow speed and lack of Tier III control below 25% engine load, the MARPOL Annex VI ECA NO_X requirements are not yielding the expected emission reductions and related air quality benefits in the North American ECA.[2]

As part of an ongoing effort to assess the Annex VI NO_X ECA program and how it could be improved, this study investigated the following Tier III-related issues. First, whether the efficiency of SCR emission control technology on Category 3 vessels can be improved, both underway and while operating at low load, based on the current engine certification requirements. Second, an evaluation of ship operating profiles in the North American ECA irrespective of engine category to examine how much time is spent at low load operation and, using that information, how the certification test cycles could be revised to reflect actual operating profiles. Third, if exhaust gas temperature can be maintained, whether it is possible to extend SCR operation to 10% load or lower.

[2] See PPR 11/INF.4, *Assessment of the impacts of the MARPOL Annex VI emission control regulations in the United States portion of the North American Emission Control Area*

2.0 ASSESSMENT OF OCEAN-GOING VESSEL LOW-LOAD OPERATION

The purpose of this task was to objectively quantify the duty cycle of category 3, ocean going vessels operating within the North American ECA. The general approach was to use publicly and commercially available data sources to identify when ships were operating within the ECA and estimate the load at which they were operating. A very high-level diagram of this process is shown in Figure 3.

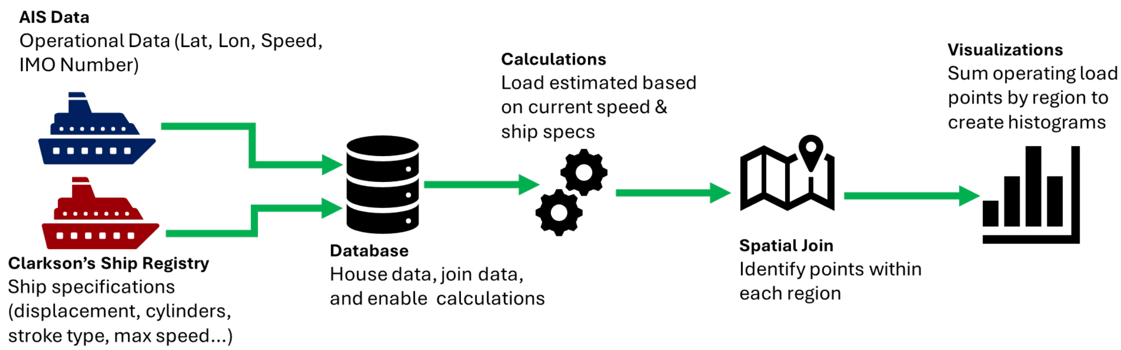


FIGURE 3. HIGH LEVEL DIAGRAM OF SHIP DATA ANALYSIS PROCESS

The estimated duty cycle for Tier III, Category 3 ships operating within the North American ECA is shown in Figure 4. It clearly shows that the actual duty cycle is very different from the duty cycle of the certification test cycle. Significant work was required to generate this ECA-based duty cycle. An equivalent amount of effort was made to ensure that this derived duty cycle is representative of ships operating throughout the entire ECA and to verify that the data used to create it were valid. This section describes the process used to validate and create these histograms.

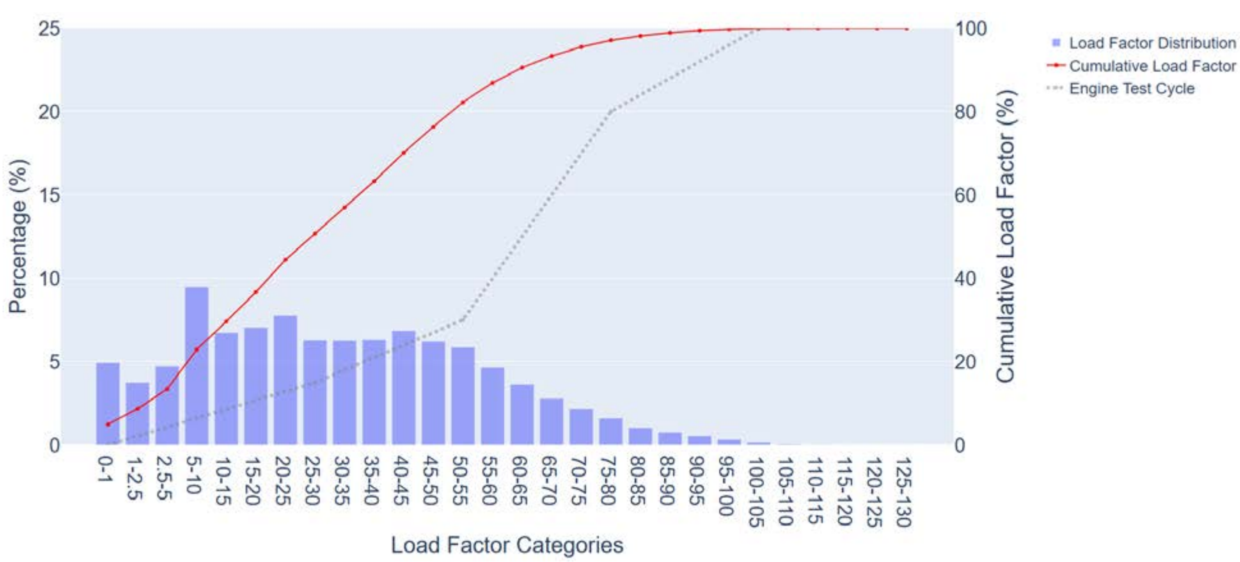


FIGURE 4. ESTIMATED DUTY CYCLE OF TIER III CATEGORY 3 SHIPS OPERATING WITHIN THE NORTH AMERICAN AND CARIBBEAN SEA ECAS (N=255)

2.1 Datasets

Four datasets were used to perform this analysis.

- Ship operation data collected from the Automatic Identification System (AIS).
- Ship specifications from Clarksons World Fleet Register, augmented by information from Sea-web™
- ECA boundaries
- C3RIA regions

The first dataset was 2022 AIS data that was acquired by EPA from the United States Coast Guard (USCG) through a historical AIS data request[3]. This dataset included ship operational data including latitude, longitude ship speed, ship dimensions, and ship's assigned International Marine Organization (IMO) number. Some of the ship dimensions in this dataset were not used because their accuracy was unknown (United States Coast Guard, 2025). AIS data can be publicly downloaded from the Marine Cadastre website (NOAA Office For Coastal Management, 2025), however the dataset used for this study also included satellite data that were provided by the USCG. This dataset had already been cleaned by the EPA prior to its receipt by SwRI.

The second dataset was from the Clarksons World Fleet Register. This dataset of ship specifications included items like IMO number, maximum operating speed, service speed, engine Tier, engine manufacturer, engine displacement, number of cylinders and ship type (Clarksons, 2024). These data were matched with the AIS data based on IMO number to tie a ship's speed and location to a set of specifications. The combined data were then used to estimate the ship engine load.

The third dataset included boundary definitions for the North American ECA and the United States Caribbean Sea ECA, Figure 5. These boundaries are defined by MARPOL Annex VI (International Maritime Orginization, 2024). Ship operation within these regions requires the use of lower sulfur ECA compliant fuels, and for ships whose keel was laid beginning 1/1/16, Tier III engines must be operated with NO_X emission control devices engaged. The ECA boundary definition files for this dataset were supplied by the EPA.

[3] https://www.navcen.uscg.gov/contact/ais-historical-request

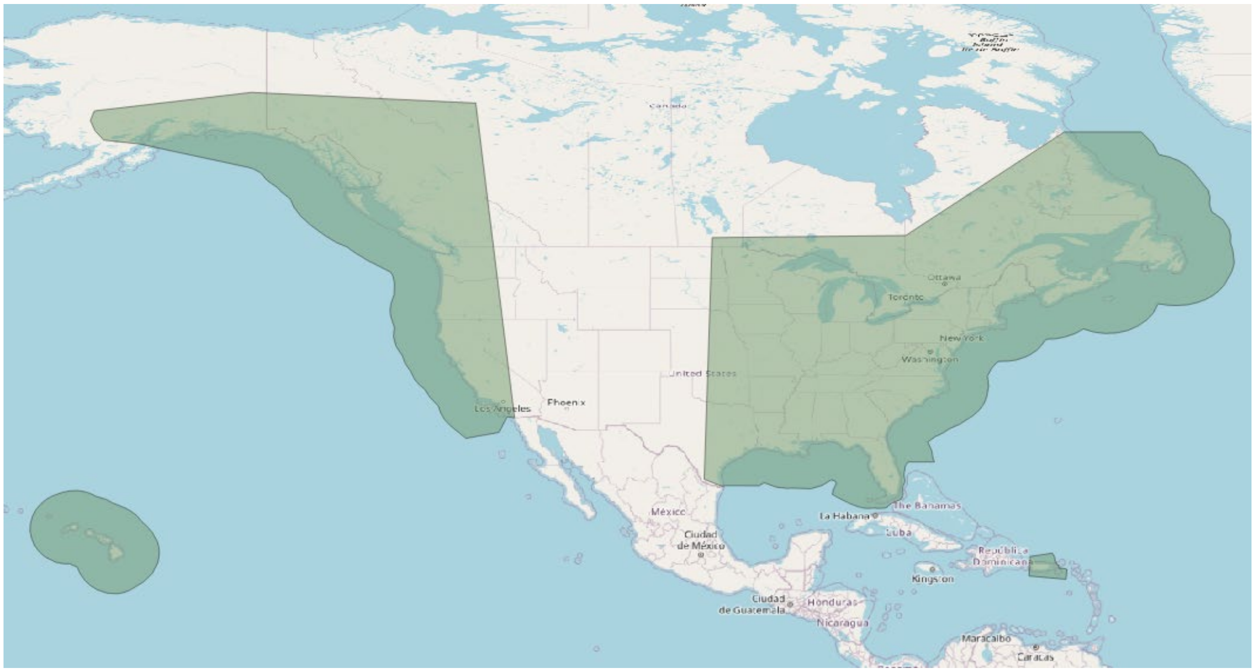


FIGURE 5. MAP OF NORTH AMERICAN & CARIBBEAN SEA EMISSION CONTROL AREAS AS RECEIVED FROM EPA

A fourth set of boundaries was also used for this study to define several regions. This dataset was supplied by the EPA and was known as the C3RIA dataset. This dataset was used to define regions within the North American ECA, as shown in Figure 6, to ensure there were no major differences in ship operation between different regions within the North American ECA. This set of boundaries did not include areas for the Caribbean Sea ECA or the Canadian portion of the North American ECA. Some work was performed to address these gaps that is discussed later in the report.

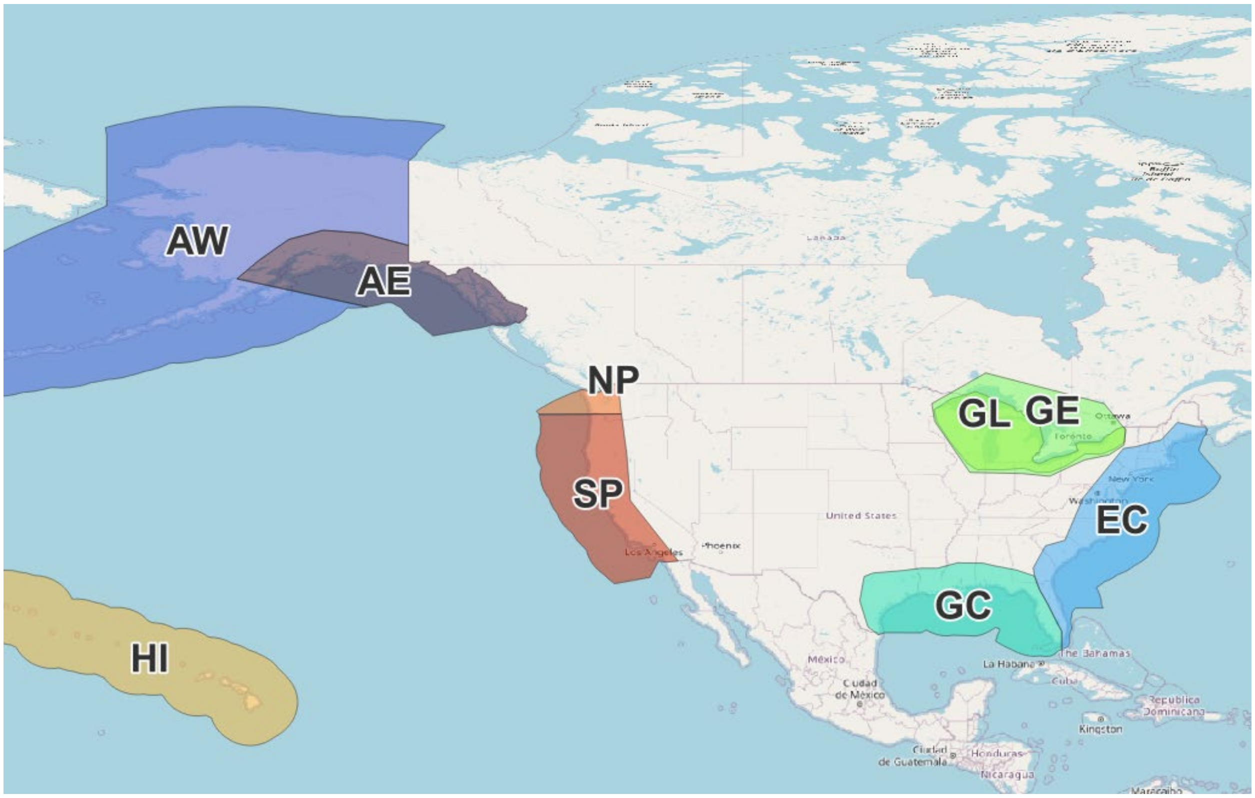


FIGURE 6. MAP OF THE BOUNDARIES DEFINED IN THE C3RIA DATASET AS SUPPLIED BY EPA

2.2 Tools Used for Data Analysis

The analysis required for this project required several different capabilities. The first requirement was the ability to interact with a very large dataset that exceeded physical random access memory (RAM). The AIS data included almost 1.5 billion records. The second requirement was the ability to join the Clarksons data and the AIS data based on the IMO number field. These first two requirements drove the need for a combined database. The final major requirement was to be able to understand where each datapoint was geographically located. This requirement drove the need for several geospatial tools.

The use of open-source tools was a deliberate decision for this project. The use of these tools will enable the analysis to be reproduced more easily than if proprietary tools had been used. The major tools used for this project are shown in Figure 7 and include Python, QGIS, Postgresql, PostGIS, and Geopandas.

FIGURE 7. OPEN-SOURCE TOOLS USED FOR ANALYSIS & VISUALIZATION (GEOPANDAS DEVELOPERS, 2024) (OPENSTREETMAP FOUNDATION, 2024) (PLOTLY, 2024) (POSTGRESQL GLOBAL DEVELOPMENT, 2024) (PYTHON SOFTWARE FOUNDATION, 2025) (QGIS DEVELOPMENT TEAM, 2024)

Visualization was another major aspect of work. There were three open-source projects that were leveraged for this project. The first was the OpenStreetMap project. Their maps were used as the basemap for all maps in this report. The second project was the Plotly plotting library. The final project was QGIS. This software was used for creating most of the maps in this report.

TABLE 2. TABLE OF SOFTWARE VERSIONS USED FOR ANALYSIS AND VISUALIZATION

Software	Version
Python	3.12
PostgreSQL	16.6
PostGIS	3.4.4
QGIS	3.34
GeoPandas	0.14.2
Plotly	5.24.1

2.3 Region Definition

Several regions were defined for this study to explore whether actual ship duty cycles differ between regions. It was expected that the ship type composition would differ between regions due to the differences in industries served by various ports, but it was unknown if the duty cycle was different.

The regions were defined using the North American and U.S. Caribbean Sea ECA boundaries along with the C3RIA boundaries that were supplied by the EPA. The resulting boundaries that were derived are shown in Figure 8.

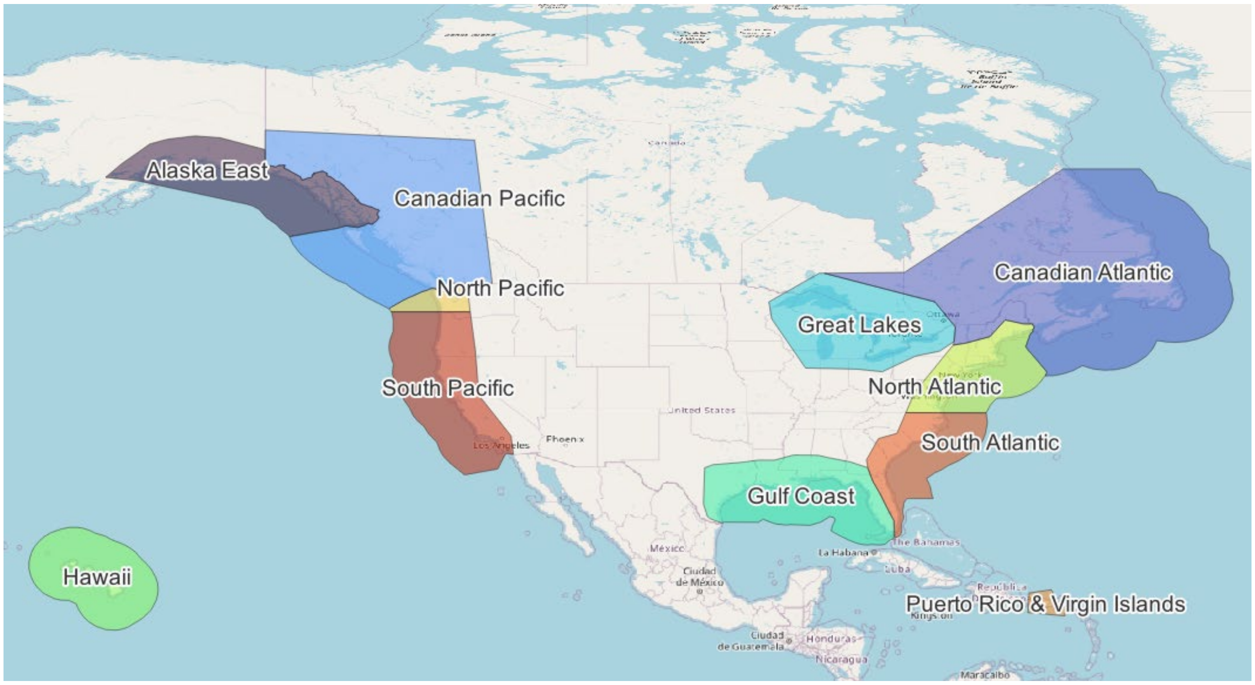


FIGURE 8. BOUNDARIES USED FOR COMPARING DUTY CYCLES BETWEEN ECA REGIONS

The regions in Figure 8 were derived by first trimming the C3RIA regions by the North American ECA. This operation is shown in the lower right corner of Figure 9. Most regions from the C3RIA dataset were not greatly impacted. However, the Alaska West (AW) region was mostly eliminated. The decision was made to remove that region entirely because the remaining portions were mostly over land. The Hawaii region was also reduced in size relative to the C3RIA definition because the ECA around Hawaii was much smaller. This can be seen by comparing the right two panes in Figure 9.

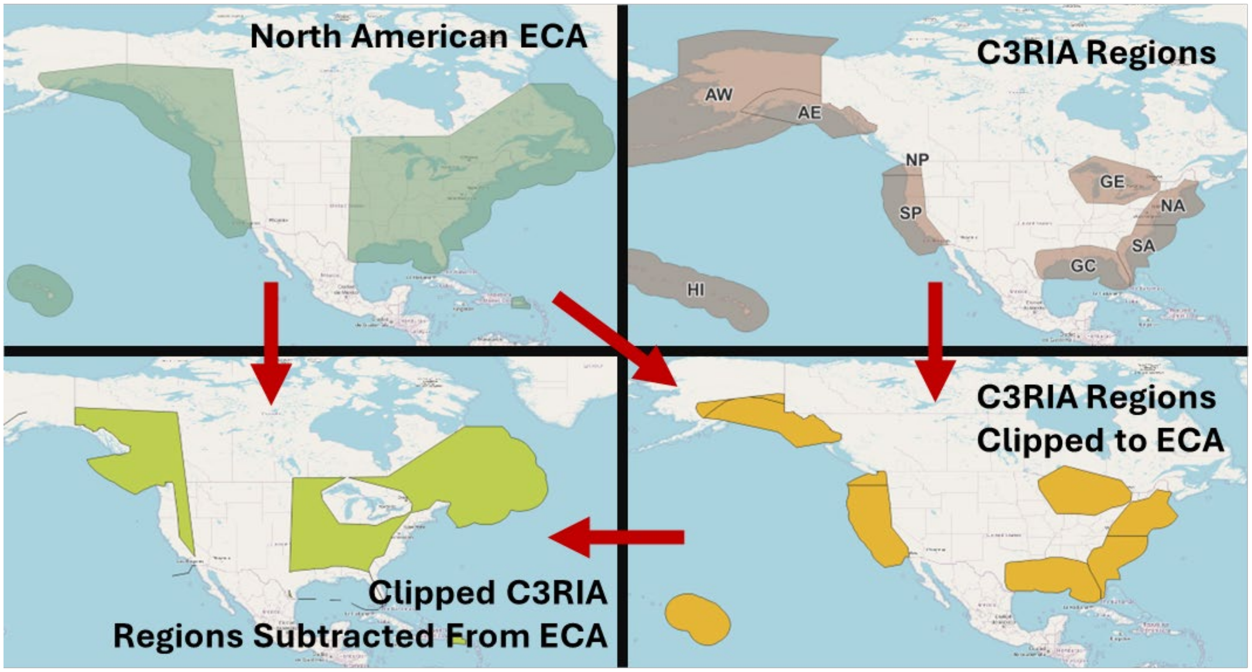


FIGURE 9. FIGURE SHOWING INPUTS AND INTERMEDIATE STEPS TO DERIVE REGIONS WITH ARROWS INDICATING DATA USED FOR EACH STEP

A few other regions were added including the Canadian Pacific, Canadian Atlantic, and Puerto Rico & Virgin Islands regions . These regions were created to ensure the entire ECA was represented by individual regions. The goal of the study was to characterize ship duty cycle within the complete ECA. These extra regions were derived by subtracting the clipped C3RIA regions from the ECA. The areas calculated from this operation are shown in the bottom left corner of Figure 9. The Canadian regions were further trimmed by hand to reflect where ship traffic would be expected. The Puerto Rico & Virgin Islands region was simply the U.S. Caribbean Sea ECA. There was no reference to the U.S. Caribbean Sea ECA in the C3RIA dataset.

2.4 Description of the Ship Included in the Study

The Clarksons World Fleet Register was used as the source of the ship specifications for the 12,303 Category 3 ships that operated in the ECA in 2022 (see Section XX below for how those ships were identified using AIS data). These specifications were used to estimate engine load and identify ship categories, for the purpose of the duty cycle analysis. The parameters that were used from the Clarksons data were IMO number, engine tier level, cylinder bore, cylinder stroke, maximum speed, and service speed. Each of these is described below.

The breakdown of ships by Annex VI Regulation 13 NO_X emissions Tier is shown in Figure 10. Tier III ships were the smaller population within this group. These ships were of the most interest for this study because they would be equipped with engines having Tier III emission control systems. It was speculated that inclusion of aftertreatment systems could cause different operational characteristics from the other emission tier levels. The Tier I and Tier II ships were separately analyzed to verify that the duty cycles for Tier III ships were similar to Tier I and Tier II ships.

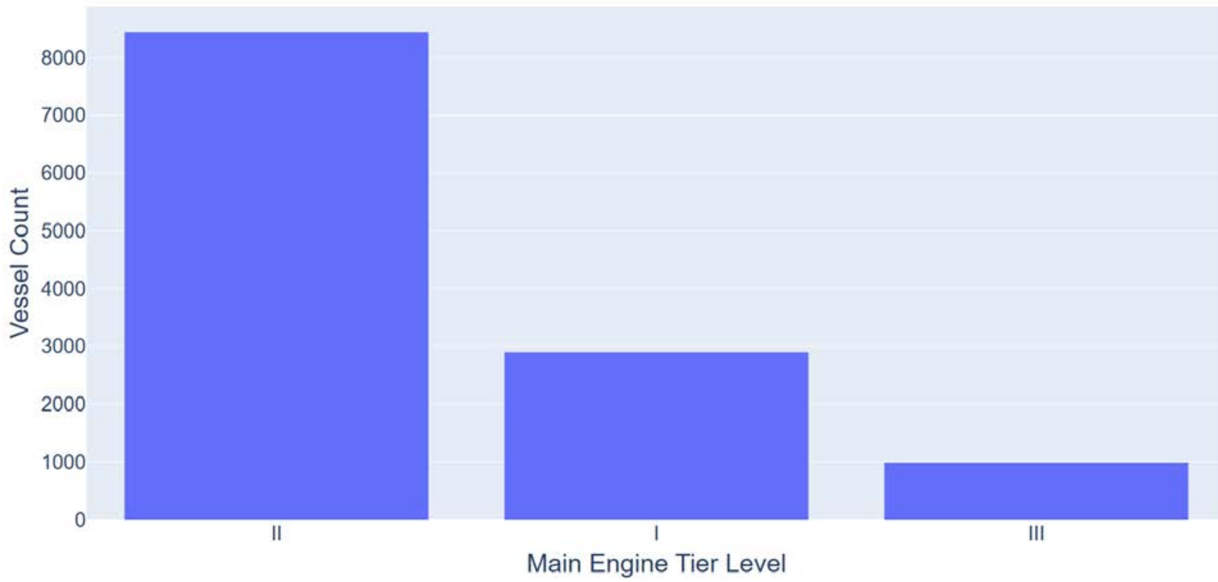


FIGURE 10. HISTOGRAM OF IMO NO_X TIER FROM CLARKSONS WORLD FLEET REGISTER FOR SHIPS INCLUDED IN THE STUDY (N=12,303)

Figure 11 shows the breakdown of ship types for the 12,303 ships studied, obtained from the Clarksons database. There were 48 different types of ships identified in the dataset, but 11 ship types made up the bulk of the fleet with over 100 ships per classification. These ship types are denoted by yellow highlighting in Figure 11. It is important to note the log scale on the y-axis in Figure 11.

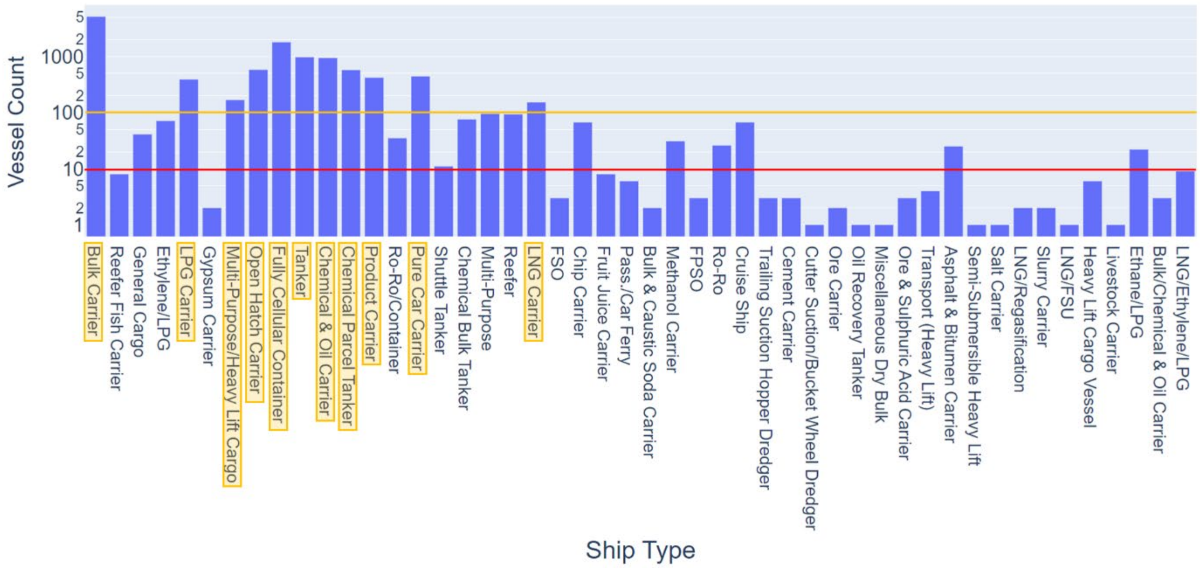


FIGURE 11. HISTOGRAM OF CATEGORY 3 SHIPTYPES FROM CLARKSONS WORLD FLEET REGISTER (N=12,303)

A histogram of main propulsion engine types for the 12,303 ships studied is shown in Figure 12. Two-stroke diesel engines were the most common propulsion engines for Category 3 ships included in the analysis. There were a few diesel-electric ships and a single hybrid ship. These were used in cruise ships and were not considered for this study because load estimation would be very difficult.

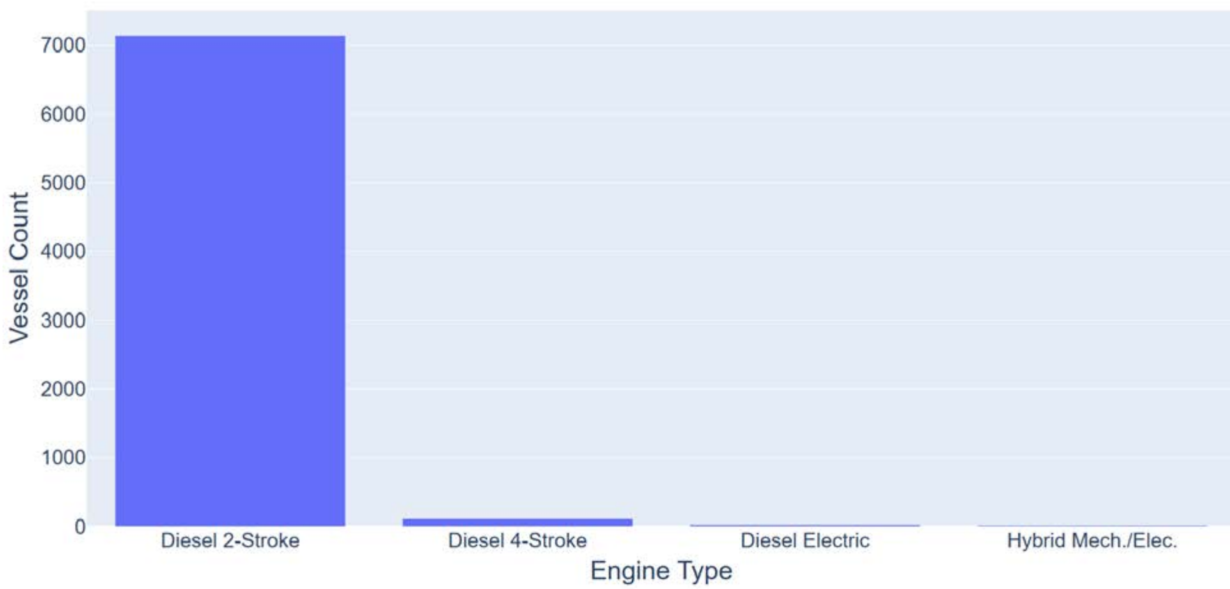


FIGURE 12. HISTOGRAM OF MAIN PROPULSION ENGINE TYPES FROM CLARKSONS WORLD FLEET REGISTER FOR CATEGORY 3 SHIPS (N=7,264)

A histogram of the bore diameter of the 12,303 main propulsion engines included in the analysis is shown in Figure 13. Most of these engines have bore diameters of 500 mm or 600 mm. The majority of engines have bore diameters that align with 100 mm increments (i.e. exactly 500, 600, 700, 800, and 900 mm).

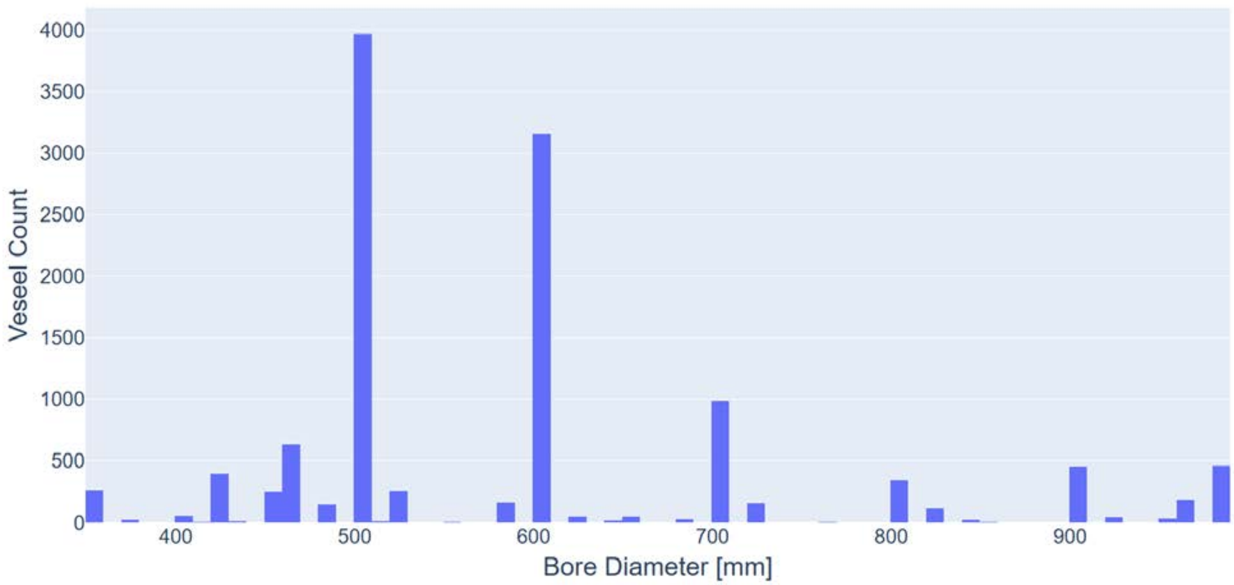


FIGURE 13. HISTOGRAM OF BORE DIAMETER FOR MAIN PROPULSION ENGINES INCLUDED IN THE STUDY FROM CLARKSONS WORLD FLEET REGISTER (N=12,303)

A histogram of main propulsion engine stroke for these same engines is shown in Figure 14. Most of these engines have a stroke length between 2,000 mm and 2,500 mm.

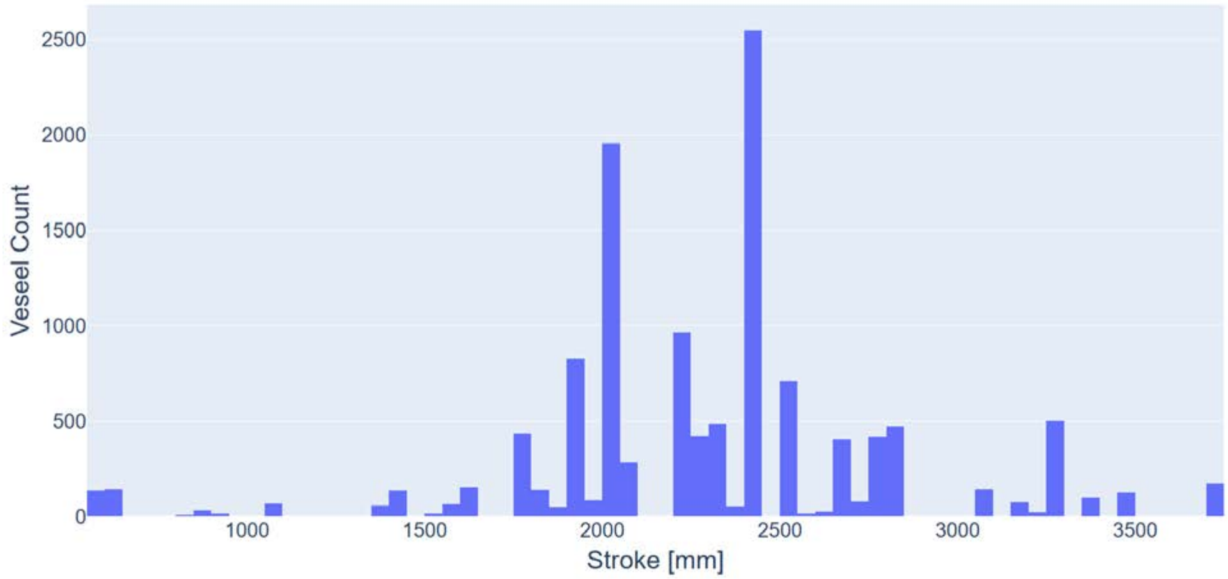


FIGURE 14. HISTOGRAM OF STROKE FOR MAIN PROPULSION ENGINES INCLUDED IN THE STUDY FROM CLARKSONS WORLD FLEET REGISTER (N=12,303)

This combination of bore and stroke results in the histogram of cylinder displacement shown in Figure 15. There are two peaks of cylinder displacement for the 12,303 main propulsion engines included in the data set, centered on 475 liter/cylinder and 675 liter/cylinder.

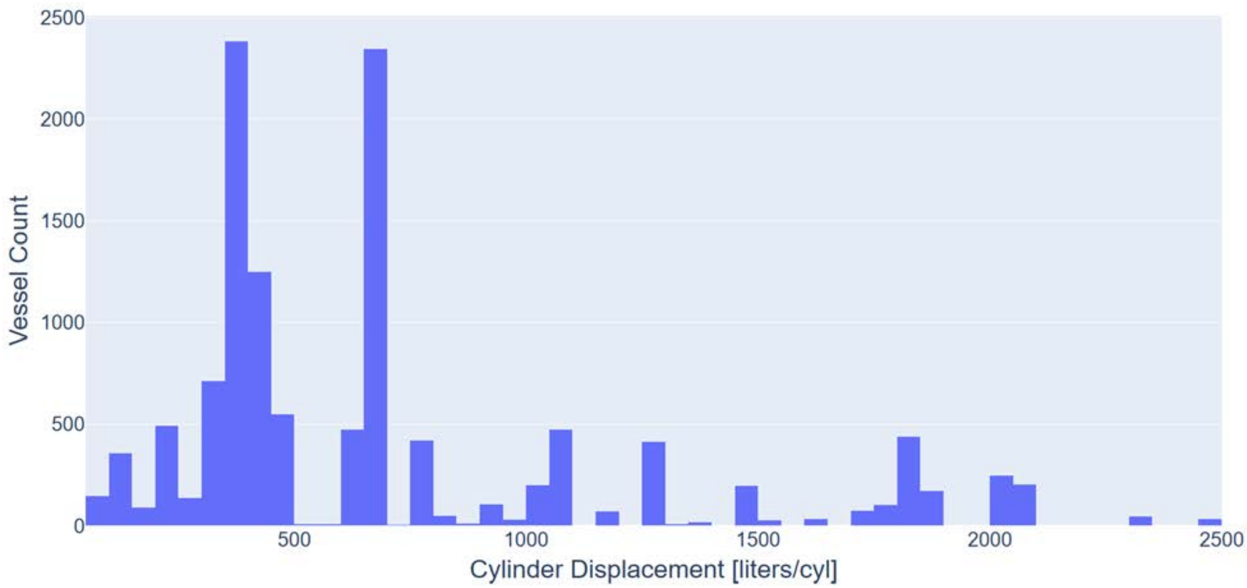


FIGURE 15. HISTOGRAM OF INDIVIDUAL CYLINDER DISPLACEMENT FOR MAIN PROPULSION ENGINES INCLUDED IN THE STUDY FROM CLARKSONS WORLD FLEET REGISTER (N=12,303)

A histogram of cylinder count per engine for the engines included in the study is shown in Figure 16. The most common cylinder count is 6 cylinders. The highest and lowest cylinder counts were 16 and 4 cylinders, but these were very uncommon.

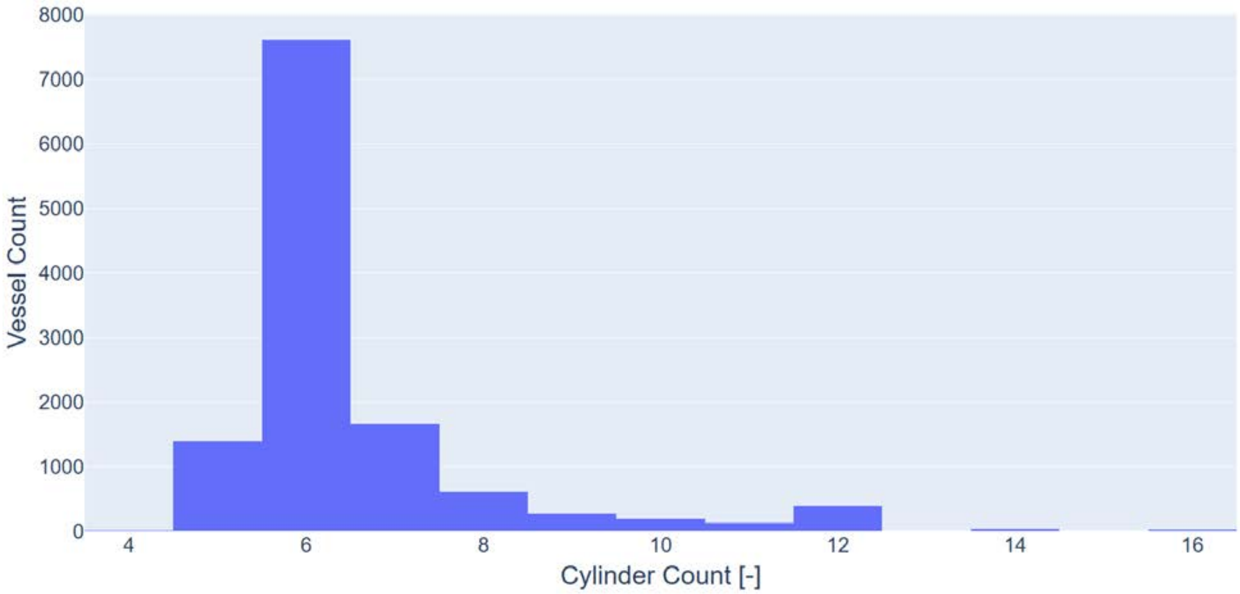


FIGURE 16. HISTOGRAM OF ENGINE CYLINDER COUNT FOR MAIN PROPULSION ENGINES INCLUDED IN THE STUDY FROM CLARKSONS WORLD FLEET REGISTER (N=12,303)

The resulting engine displacements for the engines included in the study are shown in Figure 17. The histogram shows peaks at 2,000 liters, 4,000 liters, 6,000, and 7,500 liters. The vast majority of the engines in this population are 8,000 liters or below.

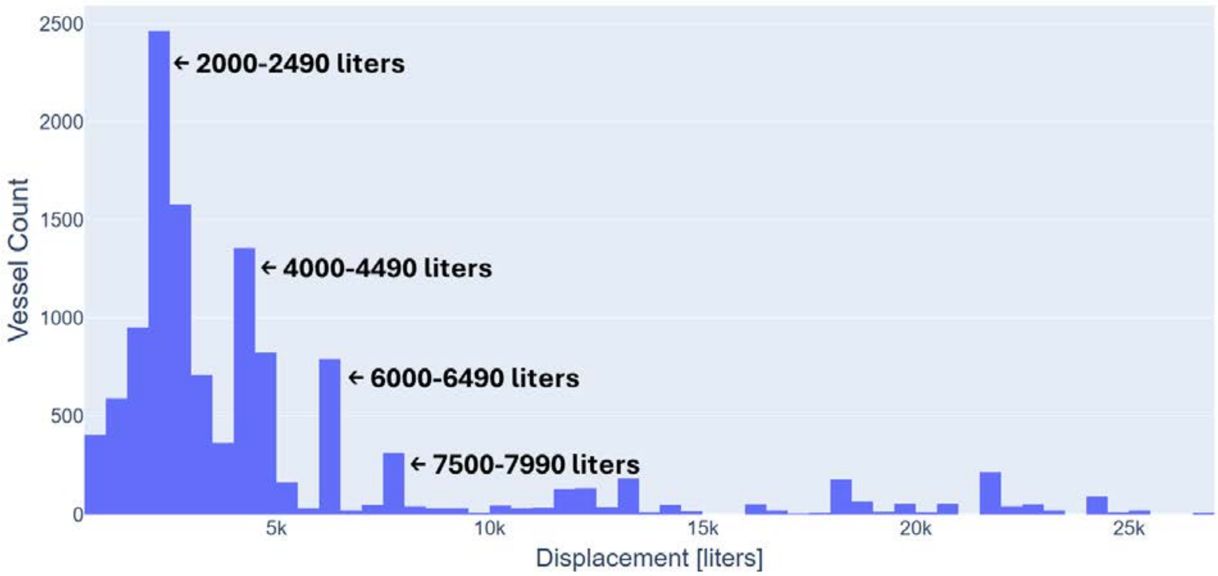


FIGURE 17. HISTOGRAM OF ENGINE DISPLACEMENT FOR MAIN PROPULSION ENGINES INCLUDED IN THE STUDY FROM CLARKSONS WORLD FLEET REGISTER (N=12,303)

A histogram of main engine power for the 12,303 ships included in the study is shown in Figure 18. There is a peak centered at 13,500 hp, and most of the main engines are rated for 25,000 hp or less.

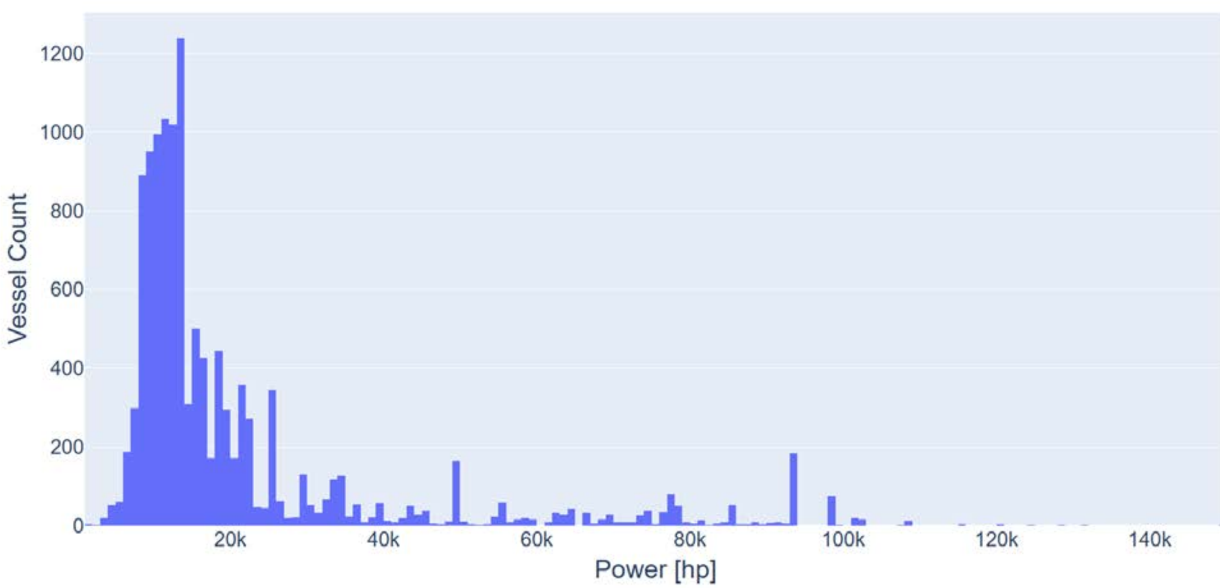


FIGURE 18. HISTOGRAM OF MAIN ENGINE POWER FOR MAIN PROPULSION ENGINES INCLUDED IN THE STUDY FROM CLARKSONS WORLD FLEET REGISTER (N=12,303)

A histogram of maximum ship speed for 630 of the ships included in the study is shown in Figure 19. It is important to note that the vessel counts in this chart are much lower than prior histograms because only 630 ships in the Clarksons database had data for maximum ship speed. Most of these ships were rated for a maximum speed between 14 and 17 knots. There was also another grouping between 20 and 22 knots. The maximum speed was 30 knots. There were four ships with a maximum speed rating about 29 knots. Three of these were fully cellular container ships with keel laid dates in 2007 and 2008. The fourth was a cruise ship.

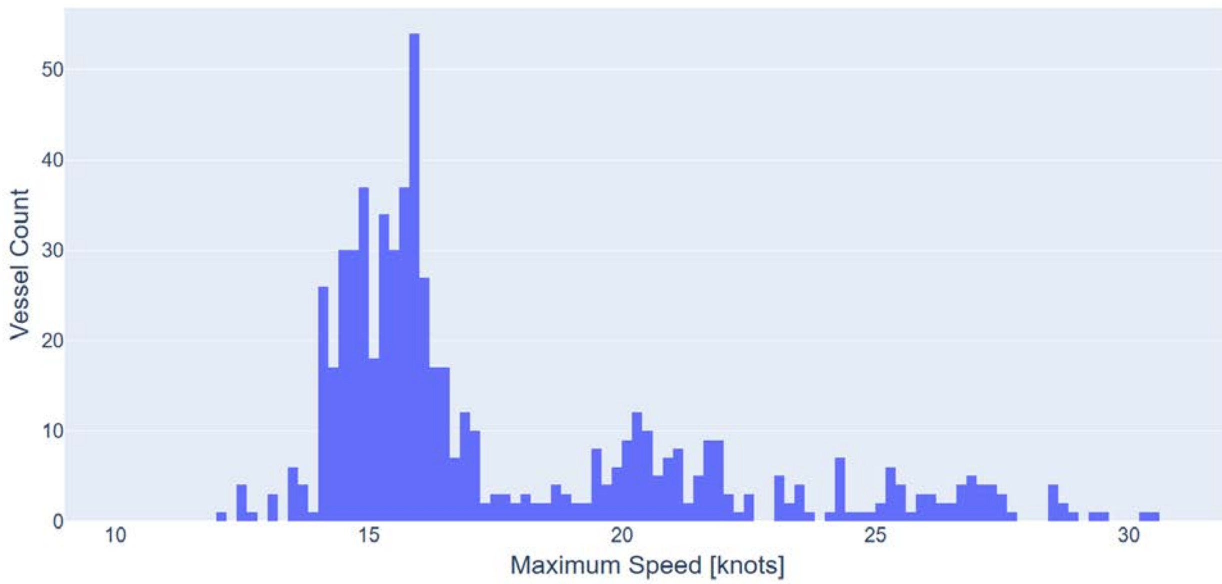


FIGURE 19. HISTOGRAM OF MAXIMUM SHIP SPEED FROM CLARKSONS WORLD FLEET REGISTER (N=630)

A histogram of service speeds for 8,502 ships included in the study is shown in Figure 20. Most ships had a service speed between 13 knots and 16 knots. However, there were smaller peaks at 20, 22, and 24 knots. Again, the vessel counts in this chart are less than the number of ship included in the study because only 8,502 ships had a specified service speed specified in Clarksons database. Several of these ships were also in the group with a specified maximum speed (N=630).

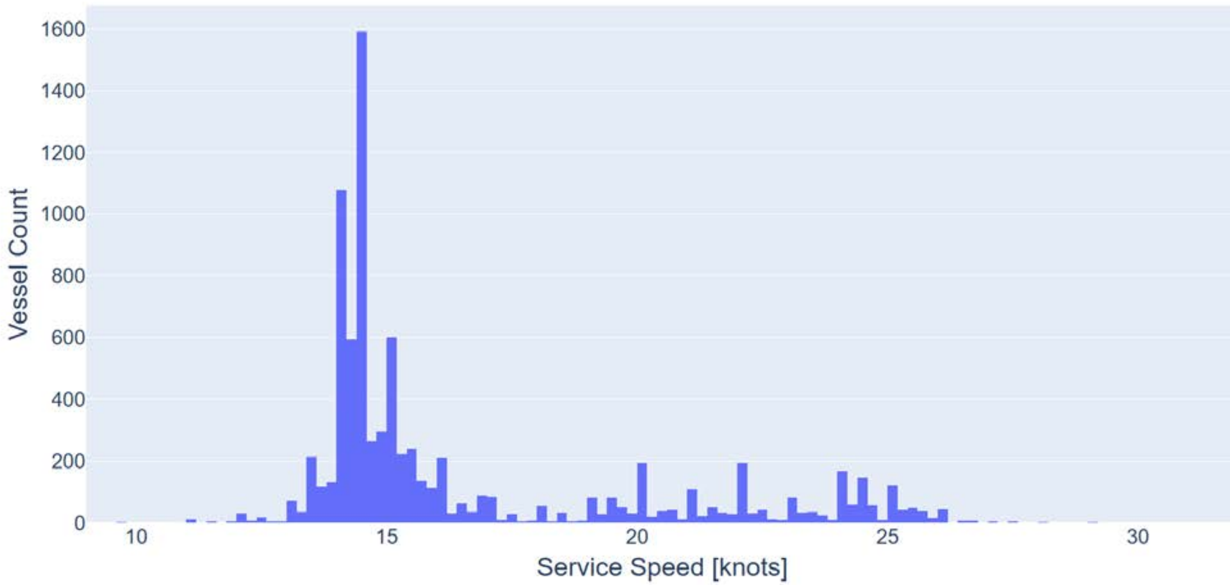


FIGURE 20. HISTOGRAM OF SHIP SERVICE SPEED FROM CLARKSONS WORLD FLEET REGISTER (N=8,502).

The composition of the Clarksons World Fleet Register data consists of ships powered with two-stroke, 6-cylinder engines. The most common ship types that these engine powered were bulk carriers, fully cellular container ships, tankers, and chemical & oil carriers. It is important to note that not all fields were populated within this dataset which will impact the data analysis in later sections.

2.5 AIS Data

The 2022 AIS dataset used for this study consisted of about 1.5 billion ship position records (data points), both inside and outside the ECA. Because of the large size of this dataset, much of the characterization and validation of this dataset was done programmatically.

As received from the USCG, the 2022 AIS dataset was aggregated into five-minute intervals and was split into separate files for data collected by terrestrial AIS (TAIS) and satellite AIS (SAIS) receivers. Prior to transmitting the AIS data to SwRI, EPA performed the following cleaning steps. First the SAIS and TAIS files were combined into a single dataset. Entries with duplicate Maritime Mobile Service Identities (MMSI), IMO, and PERIOD (timestamp) fields were removed. In cases where there were duplicate records with both SAIS and TAIS records, the TAIS record was retained and the SIAS record was removed. Next EPA filtered the dataset and retained entries where the MMSI indicated that the transmitter was one of the following: a ship, an auxiliary craft associated with a parent ship, or a group of ships. Finally, it should be noted that a database malfunction at the USCG resulted in sparse data retrieval for the months of May and June 2022.

This 2022 AIS dataset included fields for latitude, longitude, GPS reported speed, and IMO number. There were several other fields included in this dataset, but the four listed fields were the only ones used for this study. The steps taken to characterize and filter the dataset for the purposes of this study are described below.

The first major step in filtering the dataset was to remove all AIS data points that did not have an IMO number. These points were removed because they could not be associated with a set of ship specifications from the Clarksons dataset. While an attempt was made to remove points with invalid IMO numbers based on their checksums[4], this was not completed because several of the invalid IMO numbers matched ships in the Clarksons dataset.

The next step was to create a heatmap of the AIS data from all of the 23,245 ships with an IMO number, regardless of engine Tier or vessel class. Two heatmaps are shown in Figure 21, the top heatmap is <u>all</u> AIS data from ships with an IMO number, and the bottom heatmap is AIS data only from 983 Tier III ships as identified in the previous section. These two heatmaps were created with a 0.1° hexagonal grid. The position data points within each hexagon were counted in order to create these maps. It should be noted that the color scale is not constant between these two maps because the total point count is very different. However, roughly similar traffic patterns can be observed between the two maps suggesting that Tier II and Tier III operate in similar ways.

[4] A checksum is a mathematical equation that can be applied to the IMO number to verify that the contents of the received IMO number are valid.

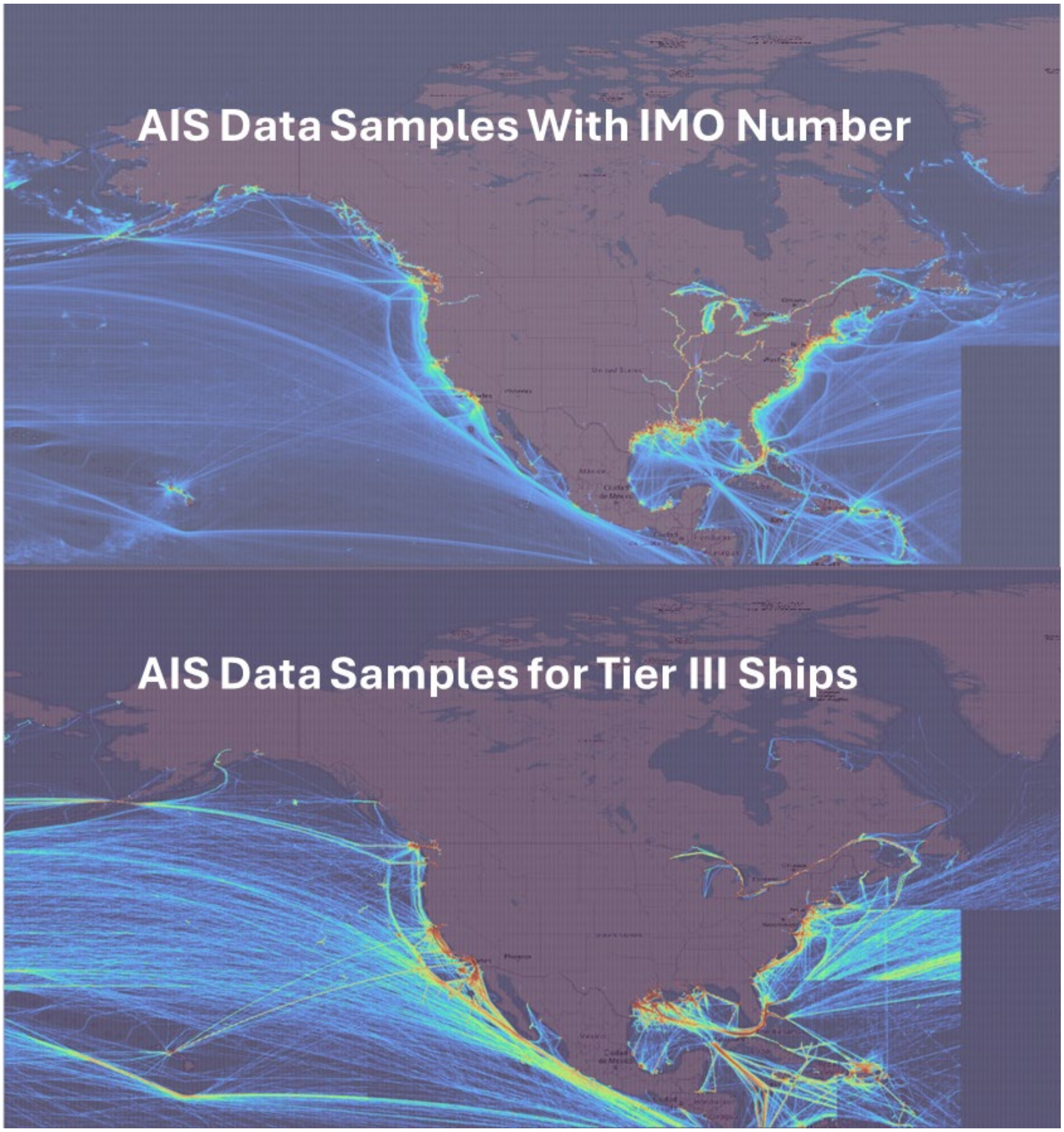


FIGURE 21. HEAT MAPS BASED ON AIS DATA WITH IMO NUMBER COMPARING TRAFFIC PATTERNS FOR ALL SHIPS WIYH AIS DATA VS TIER III SHIPS

There are some subtle differences that can be seen between the two heat maps that may reflect differences in the Tier III fleet composition, which includes only those with Category 3 propulsion engines. One of these differences can be seen between Newfoundland and Greenland in Figure 21: the heat map for the complete dataset shows lots of small tear drop shapes in that area while the Tier III map does not contain these shapes. These features were not present in the Tier III map because they were generated by vessels that were not Category 3, Tier III vessels.

The next step was to consider the context for the part of the dataset that had an IMO number and were able to be associated with a ship in the Clarksons dataset, regardless of emissions Tier.

The first step was to identify how many of these ship position records fell within the North American ECA. It was found that about 2/3 or 110 million of these data points were contained in the ECA. The remaining third fell outside the ECA. The preponderance of points inside the ECA may occur because ships at anchor or in port may continue broadcasting their position even when they are not moving or are drifting. The points at anchor or in port were included for the location data point counts but omitted for estimation of load. These counts are depicted by engine tier in Figure 22.

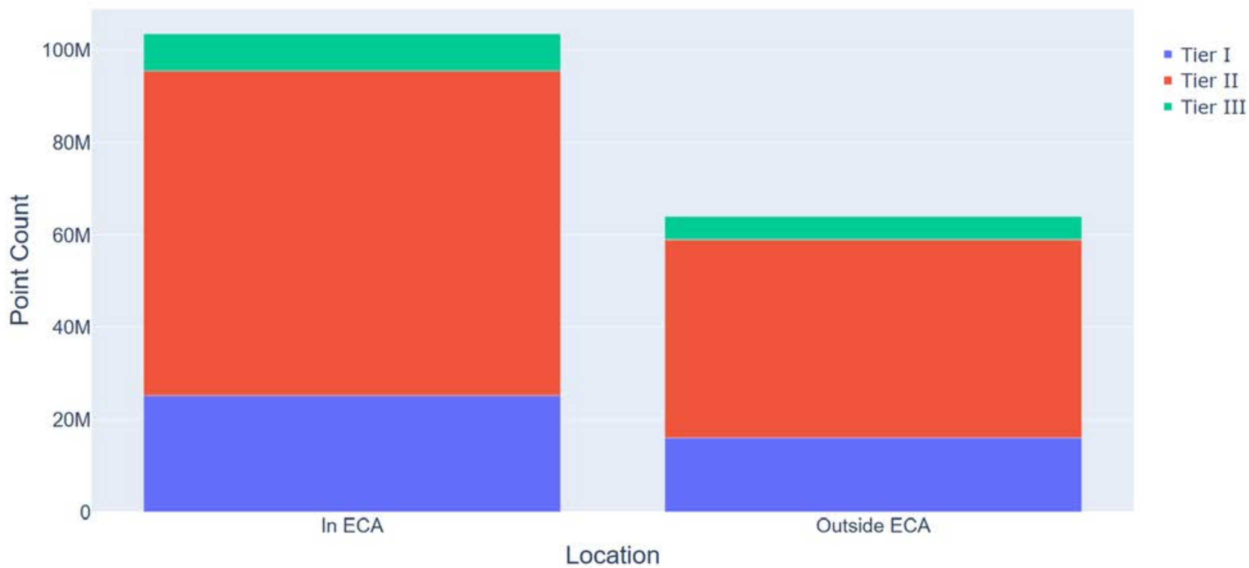


FIGURE 22. BAR CHART OF SHIP LOCATION COUNTS INSIDE AND OUTSIDE OF ECA FOR AIS DATA WITH AN IMO NUMBER JOINED TO THE CLARKSONS DATASET (N=167,512,041)

The next division that was investigated for the AIS data was region. This was done to understand which regions have the most ship traffic. Understanding this distribution helps to identify any biases that may exist within the dataset caused by regional differences. Point counts by region inside the ECAs are shown in Figure 23. It was found that the Gulf Coast region dominated the point counts by region. Alaska East had very few points.

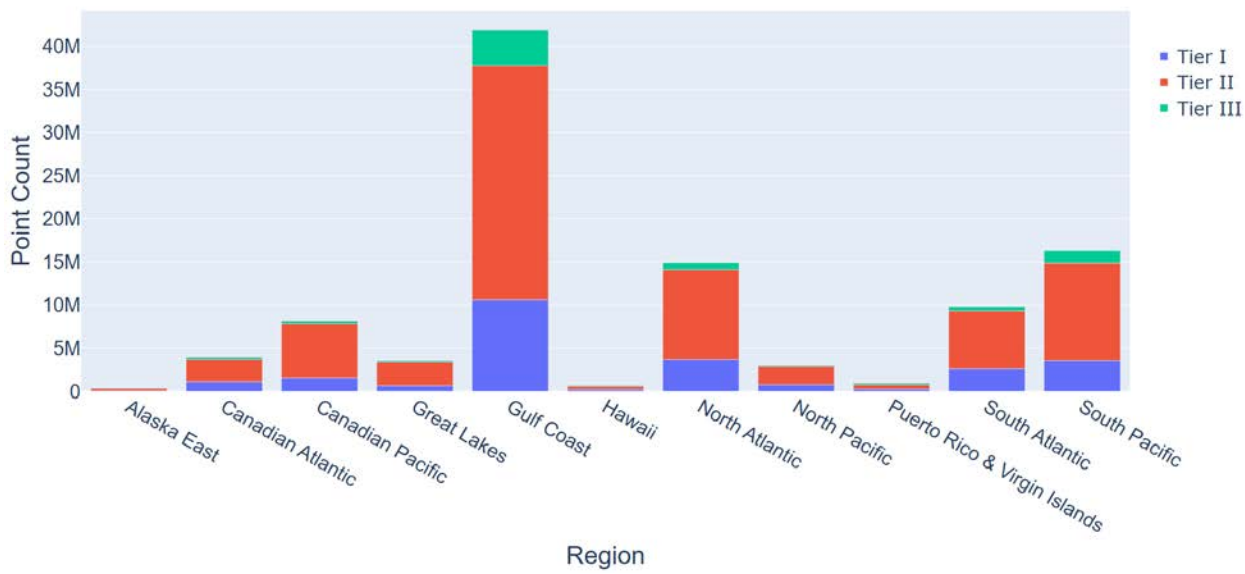


FIGURE 23. BAR CHART OF POINT COUNTS BY REGION FOR AIS DATA WITH AN IMO NUMBER JOINED TO THE CLARKSONS DATASET (N=103,552,955)

Another way to look at the data is by IMO number. There were 10,745 unique IMO numbers for ships that entered the ECA that could be linked to a ship in the Clarksons dataset. The data presented in Figure 24 shows the number of ship location counts for unique IMO numbers found in each region within the dataset. It is important to note that the sum of these bars (24,161) is much larger than the number of ships that were ever present in the ECAs (10,745) because many ships visited several regions. Over half of the ships in the dataset were observed in the Gulf Coast region. Understanding these counts is important when looking at regional differences. Results for regions with a small number of unique IMO numbers may be skewed by a small number of ships that may have atypical operational characteristics relative to the rest of the fleet.

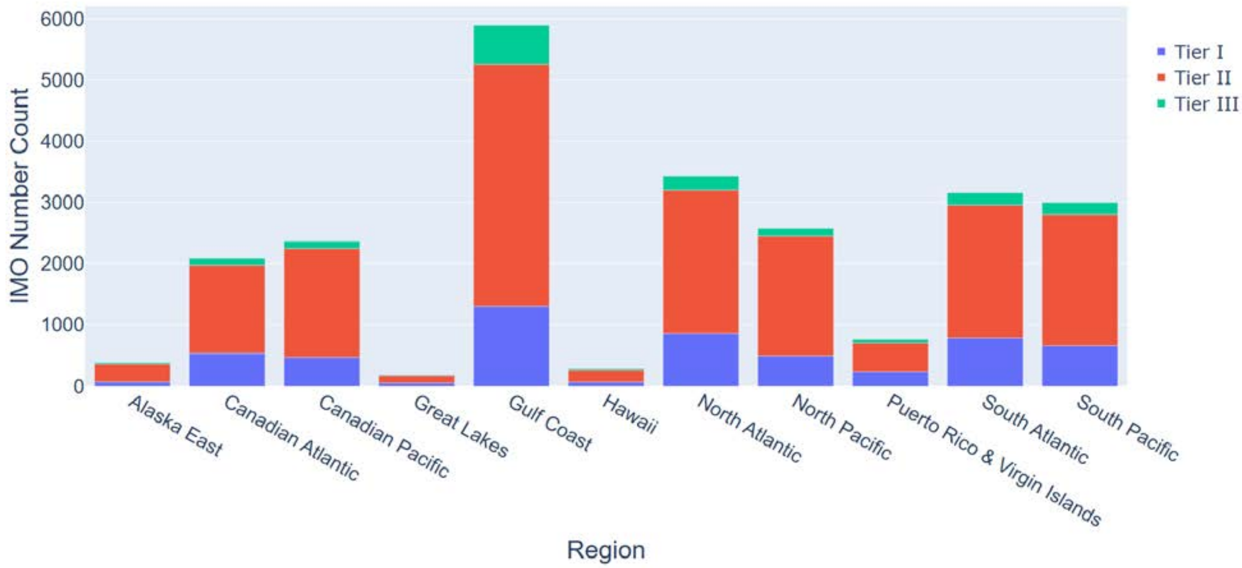


FIGURE 24. BAR CHART OF UNIQUE VESSEL COUNT BY REGION FOR AIS DATA BASED ON IMO NUMBER JOINED TO THE CLARKSONS DATASET

2.6 Duty Cycle Estimate & Results

The next step in the analysis was to estimate the ship propulsion engine load from the AIS data and summarize it. This involved using ship design speed and current speed for each point. This data was then grouped by region, ship type, and engine tier level to create a fleet level duty cycle.

2.6.1 Load Estimation

The load (L) for each ship propulsion engine as a fraction of the maximum continuous power rating was estimated using the propeller law shown below. This method was chosen because it only requires the use of current vessel speed (V) and its maximum speed (V_{max}). A value of 1.0 was chosen for M, the decimal fraction of the maximum continuous engine power rating at maximum vessel speed, in the equation below to reflect that 100% of MCR corresponds with maximum speed in normal sea conditions (MAN Energy Solutions, 2023). This approach is reasonable because…

$$L = M \cdot \left(\frac{V}{V_{\max}}\right)^3$$

It was mentioned in the description of the Clarksons dataset above that only a fraction of ships had maximum or service speed identified. This omission required a special process to produce the best estimate of engine load possible. If maximum speed was available, this value was used for the estimation of load with the propeller law. If only the service speed was available, it was divided by 0.94 to estimate the maximum speed for use with the propeller law. If no speed was specified, the load was set to a null value so that it would not influence the duty cycle development. This process is illustrated in Figure 25. This approach was only applied to points with a speed greater than 2 knots. It was assumed that ships traveling slower than 2 knots were anchored, at berth, or not operating under their own power.

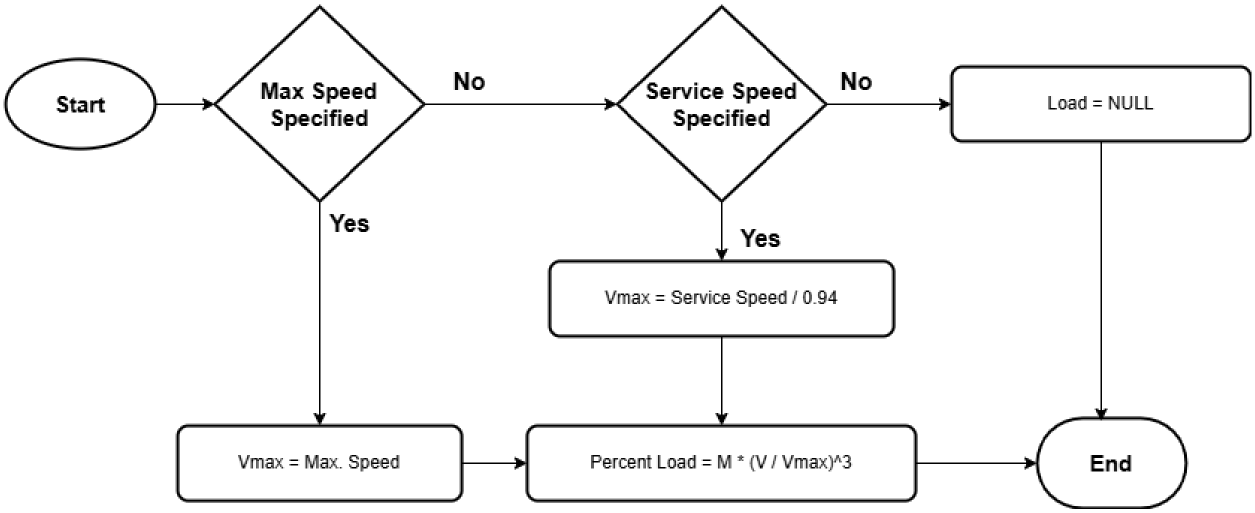
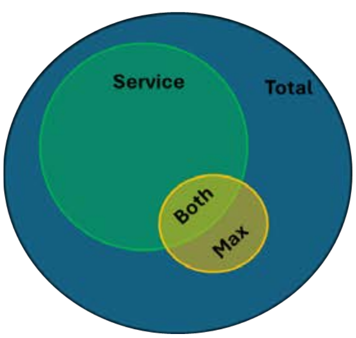


FIGURE 25. FLOW CHART DESCRIBING THE PROCESS USED TO CALCULATE PERCENT LOAD BASED UPON THE AVAILABILITY OF MAXIMUM SPEED OR SERVICE SPEED IN CLARKSONS SHIP SPECIFICATIONS

The omission of maximum and service speed led to the omission of a significant amount of data. The majority of Tier I ships had valid speed data. However, this filtering rendered 66% of the Tier III ship data and about 34% of Tier II data unusable. An attempt was made to predict these speeds for each ship, but we were unable to develop a method to confidently verify the ship design speed predictions. A Venn diagram and table outlining how many ships had maximum or service speeds specified are shown in Figure 26.



	Tier I	Tier II	Tier III
Total	2,894	8,432	977
Valid Max Speed	164	432	34
Valid Service Speed	2,726	5,453	323
Valid Service or Max Speed	2,733 (94.4%)	5,548 (65.8%)	329 (33.7%)
No Valid Speed	266 (9.2%)	3401 (40.3%)	653 (66.8%)

FIGURE 26. VENN DIAGRAM AND TABLE OF SHIPS WITH MAXIMUM SPEED, SERVICE SPEED, BOTH SPEEDS, OR NO SPEED SPECIFIED

The final validity check on the data for each ship was the engine load. Many ships had predicted maximum engine loads in excess of 125%. It was assumed that ships that exhibited loads in excess of 125% anywhere in the dataset had an incorrect specification or estimate of maximum vessel speed. These ships were omitted to ensure the data used for duty cycle estimation were as reliable as possible. It would take a speed significantly greater than the rated speed to cause the estimated load to be in excess of 125%.

Combining all the filters for identifying the points of interest results in a very small percentage of the total available points being used: 1,960 Tier I ships, 4,040 Tier II ships, and 255 Tier III ships. This filtering process is illustrated in Figure 27. This chart visually describes how the data points used for this study were defined from the dataset as received. The largest reduction in useable points was due to the lack of an IMO number in the AIS dataset. This eliminated about two-thirds of the data. The next largest filter was the lack of an IMO number match with the Clarksons data. The lack of a match in most cases was due to Clarksons not having the information for a specific ship. This could be because data for non-category 3 vessels was not obtained from Clarksons. Similarly, the effect of filtering on the reduction in ship counts based on unique IMO numbers is shown in Figure 28. Taken together, these two figures show that load histograms generated by this study for Category 3 Tier III vessels in the ECA were based on 2,310,734 AIS speed data points from 255 ships.

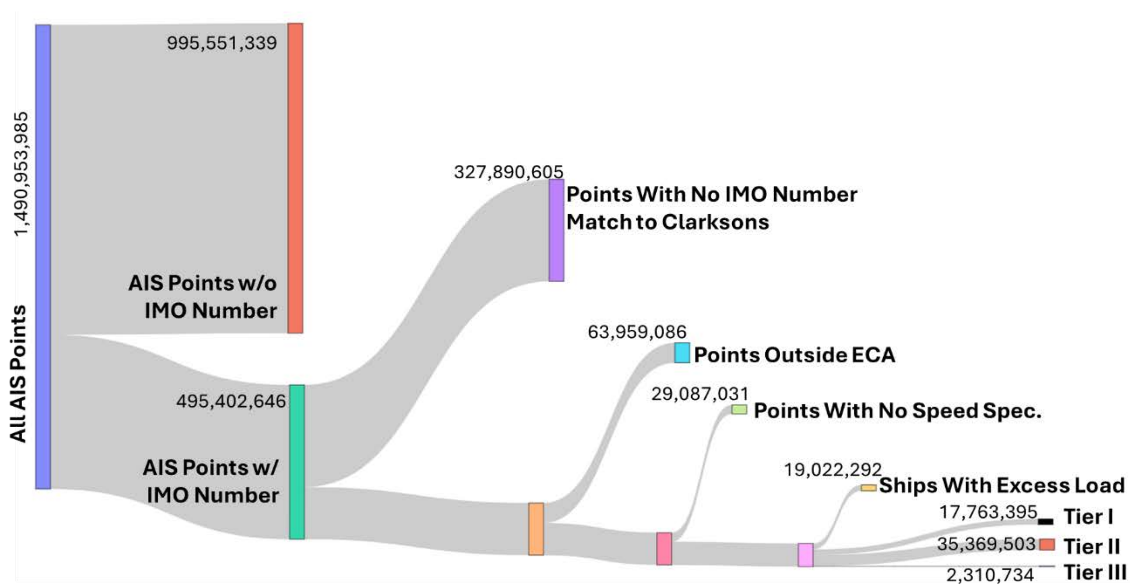


FIGURE 27. SANKEY DIAGRAM OF AIS POINT FILTERING TO IDENTIFY USEFUL POINTS FOR DUTY CYCLE DEVELOPMENT

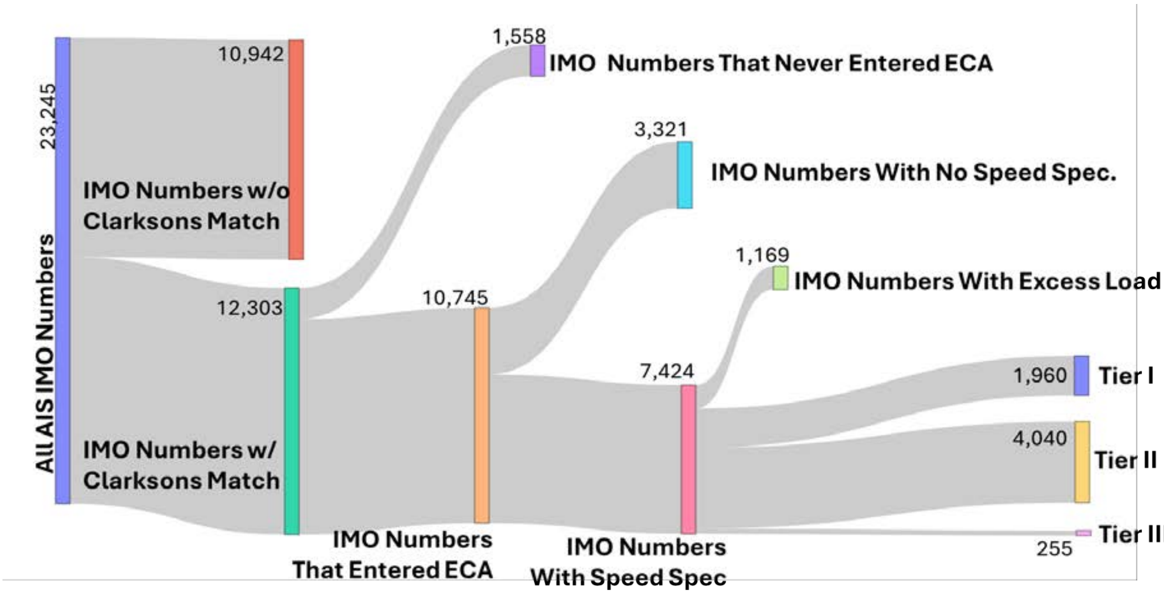


FIGURE 28. SANKEY DIAGRAM DEPICTING IMPACT OF FILTERING ON IMO NUMBERS USED FOR DUTY CYCLE DEVELOPMENT

For the purpose of this study, the ship types included in Clarsksons were combined into a smaller number of categories as shown in Table 3. There were many different types of ships used to classify the vessels in the Clarksons data. The data presented in Table 3 and Table 4 shows the mapping used for the study. The counts presented in Table 3 are the counts for each classification in the Clarksons data. Table 4 presents the counts for the ships used to develop the duty cycles in the next subsection. The study counts are lower for two reasons. The first is that some of these ships never entered the ECA. The second is that some of these ships had no speed specification. Comparison of these two ship counts is important to ensure that one type of ship is not being eliminated from the dataset and introducing a bias.

TABLE 3. TABLE DESCRIBING THE MAPPING OF SHIP CLASSIFICATIONS IN CLARKSONS TO SHIP GROUPING USED FOR STUDY AND THEIR DISTRIBUTIONS WITHIN THE CLARKSONS DATA

Study Classification	Clarksons Classification	Tier III	Tier II	Tier I
Tankers	Crude Tankers, Product Tankers, Chemical Tankers, Spec. Tankers	167	983	785
Gas Carriers	LPG, LNG	85	195	72
Bulk Carriers	Bulkers	19	2,319	681
General Containers	General Cargo	0	67	7
Containers	Containerships	28	753	590
Other	All other ship types	8	532	163
Total		**307**	**4,849**	**2,298**

TABLE 4. TABLE DESCRIBING THE MAPPING OF SHIP CLASSIFICATIONS IN CLARKSONS TO SHIP GROUPING USED FOR STUDY AND THEIR DISTRIBUTIONS WITHIN THE SHIP POPULATION USED FOR DUTY CYCLE DEVELOPMENT

Study Classification	Clarksons Classification	Tier III	Tier II	Tier I
Tankers	Crude Tankers, Product Tankers, Chemical Tankers, Spec. Tankers	111	692	555
Gas Carriers	LPG, LNG	50	107	44
Bulk Carriers	Bulkers	14	1,763	465
General Containers	General Cargo	0	15	7
Containers	Containerships	27	663	526
Other	All other ship types	5	379	135
Total		**207**	**3,619**	**1,732**

The final aspect of the data is how much time each ship spent within the ECA during the study period. The calculated time is based upon the number of points that exist within the ECA boundary for each IMO number. The total time per ship can be for one or several trips in 2022, depending on how many times the ship entered the ECA. The histogram presented in Figure 29 shows the distribution of how much time ships spent inside the ECA. It should be noted that there were 451 ships that entered the ECA but were within the ECA for less than an hour. This may also be the case with ships that operated in the ECA between 1 and 10 hours, or they may have been in innocent passage (no stop at a U.S. port). These ships are believed to have erroneously entered the ECA and probably made no operational changes while operating within the ECA (i.e., they probably operated in cruise mode for the brief amount of time they transgressed the boundary). While they were not omitted from the duty cycle development, their inclusion should not have an impact on the duty cycle results because they represent less than 0.01% of the 4.6 million hours of data.

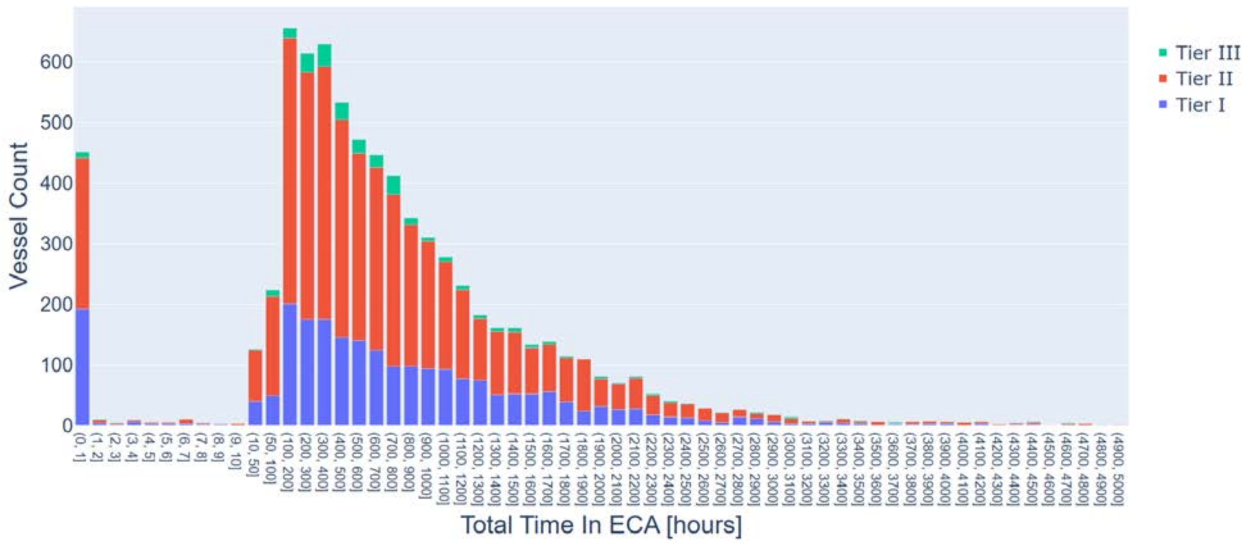


FIGURE 29. HISTOGRAM OF TIME SPENT IN ECA IN HOURS (N=7,357)

2.6.2 Estimated Duty Cycles

The duty cycle for ships at each emission tier level were developed by creating engine load histograms from the dataset generated using the process outlined in the previous section. The Sankey Diagrams in Figure 27 and Figure 28 depict how the dataset of interest was identified. The equation in Section 2.6.1 explains how the engine load was estimated.

These histograms were created at the fleet level within the North American ECA for each emission tier level. They were divided by tier to ensure that differences in engine technology did not cause differences in operational characteristics. The duty cycle for the Tier III ships is presented in Figure 30. This figure is identical to Figure 4. It is shown here for ease of comparison. The results for Tier I and Tier II ships are presented in Figure 32 and Figure 31 respectively.

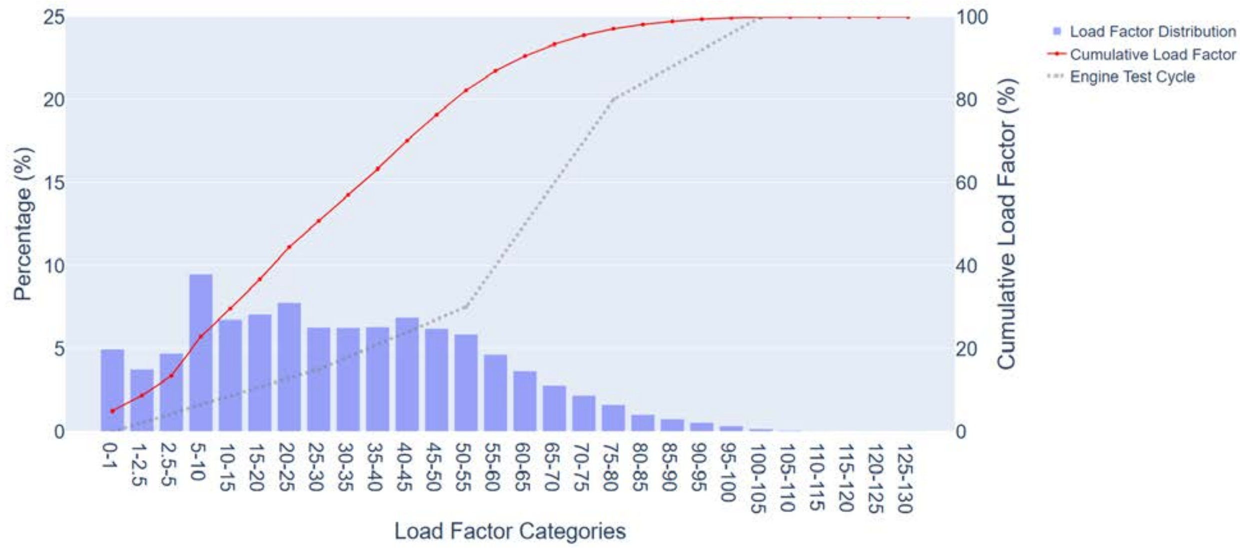


FIGURE 30. ESTIMATED DUTY CYCLE OF TIER III CATEGORY 3 SHIPS OPERATING WITHIN THE NORTH AMERICAN ECA (N=255)

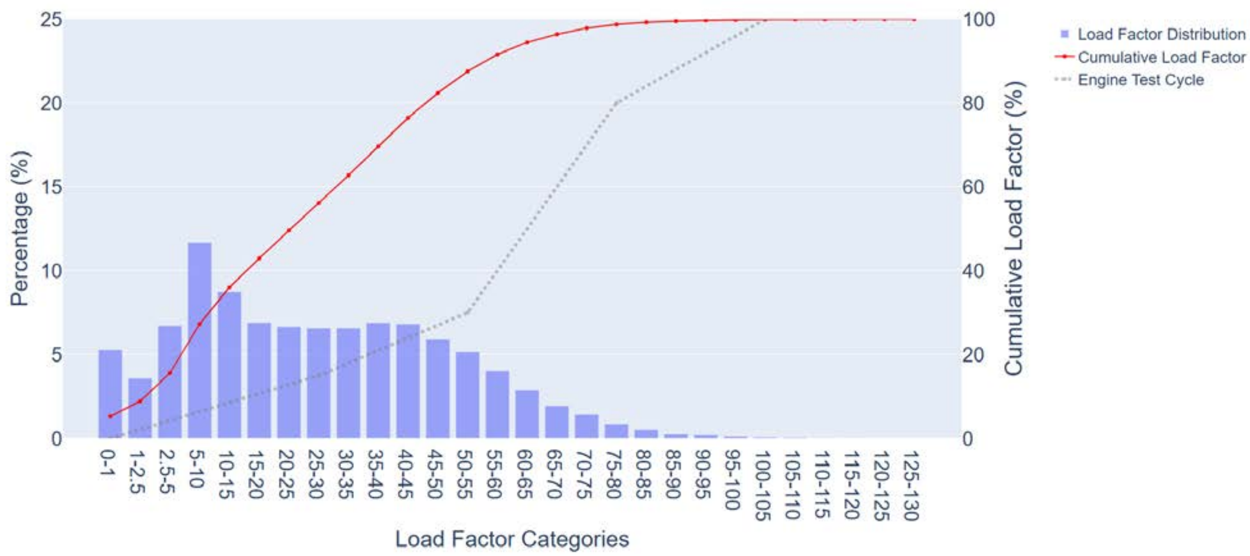


FIGURE 31. ESTIMATED DUTY CYCLE FOR TIER II CATEGORY 3 SHIPS IDENTIFIED IN NORTH AMERICAN ECA (N=4,040)

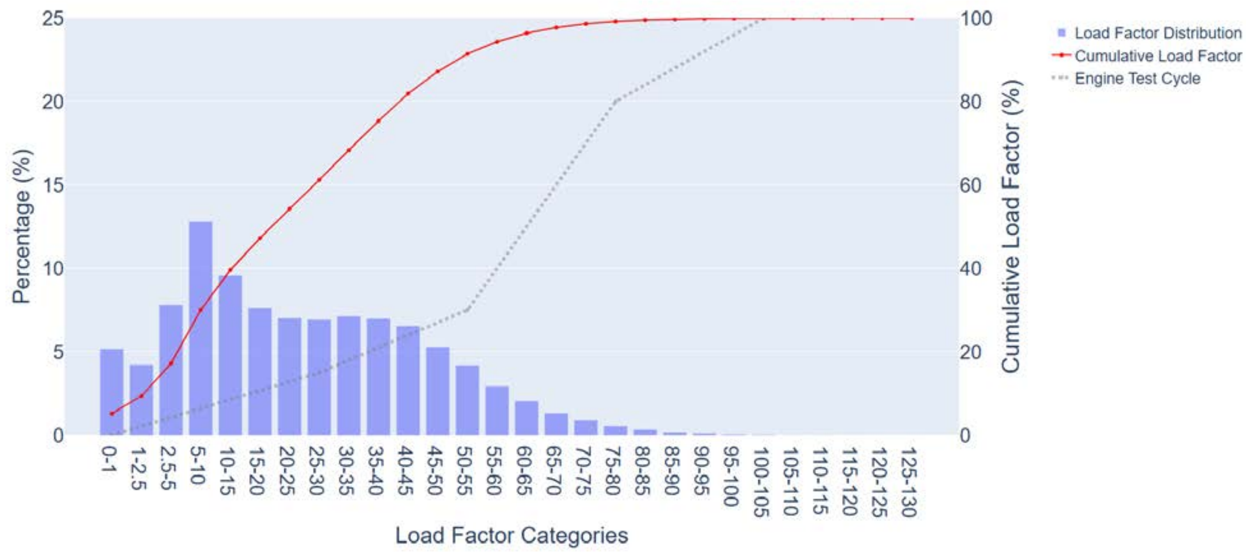


FIGURE 32. ESTIMATED DUTY CYCLE FOR TIER I CATEGORY 3 SHIPS IDENTIFIED IN NORTH AMERICAN ECA (N=1,960)

The estimated duty cycles for all three emission tiers appear to be very similar. Figure 33 presents the duty cycles for each tier relative to the certification test cycle. They estimated duty cycles all have a much lower load than the certification test cycle. The Tier III duty cycle is the most heavily loaded of the three duty cycles, but it is still dramatically different from the certification test cycle.

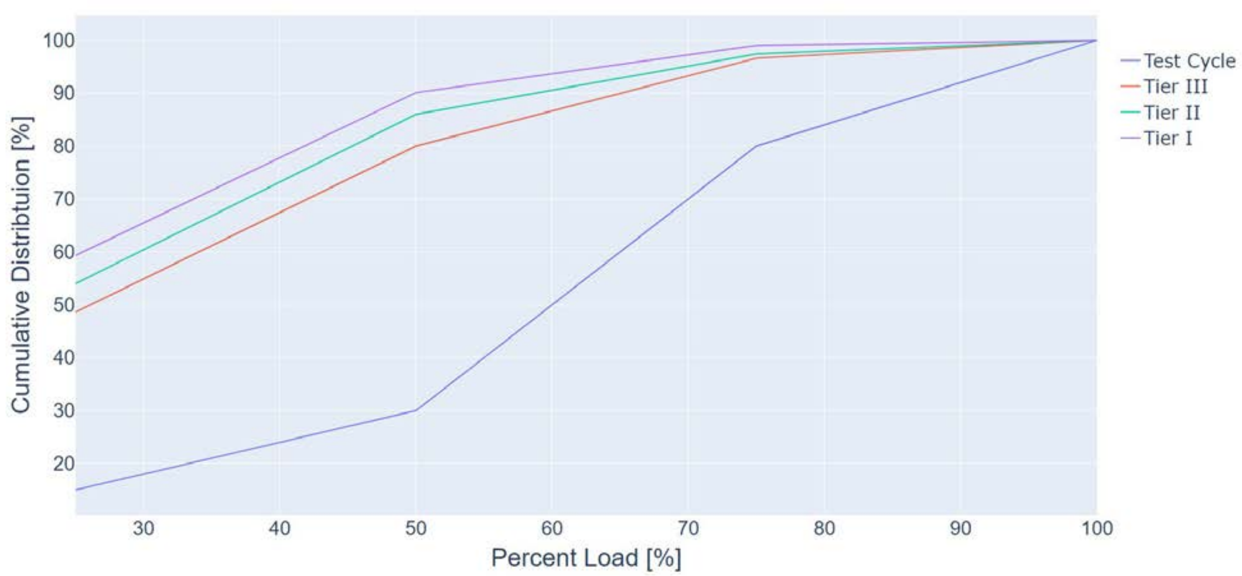


FIGURE 33. LINE CHART COMPARING CUMULATIVE DISTRIBUTION OF DUTY CYCLE FOR TIER I, TIER II, TIER III, AND CERTIFICATION TEST CYCLE

The duty cycles for each emission tier were broken out by major ship classification to ensure that the aggregate duty cycle is representative of the five major ship types shown in Table 4 or the legend for each duty cycle plot. The estimated duty cycles for each of these ship types are shown in Figure 34 to Figure 36. These duty cycles are very similar. The largest deviation is that the container ships have a duty cycle that is consistently lighter than the other ship types. The Tier III chart in Figure 34 does not have an entry for general container ships because there were none.

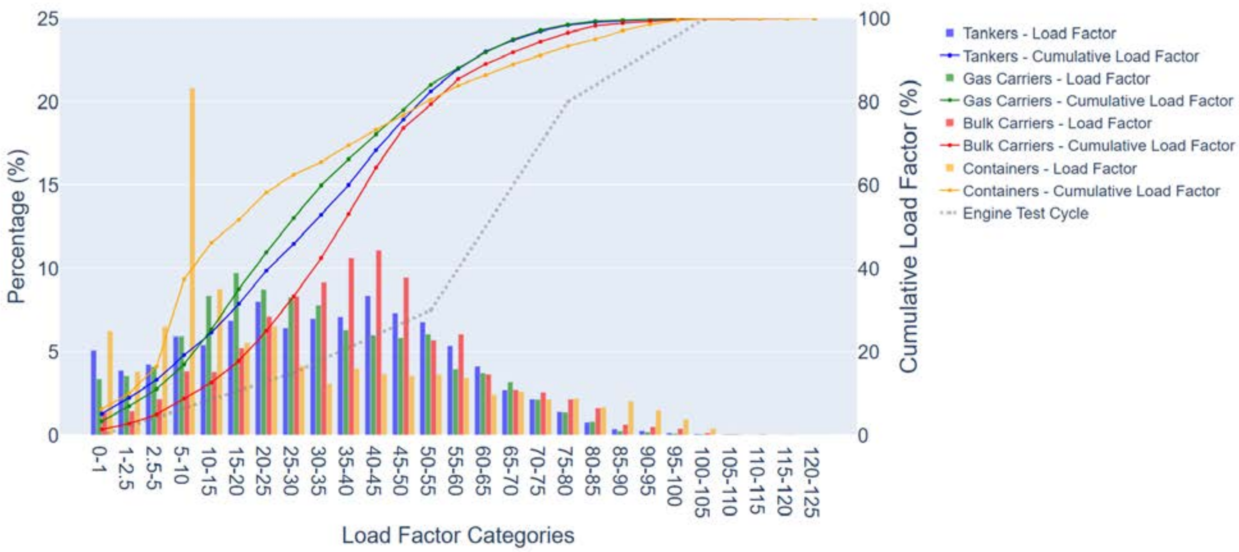


FIGURE 34. ESTIMATED DUTY CYCLE BY SHIP TYPE FOR TIER III SHIPS IDENTIFIED WITHIN THE NORTH AMERICAN ECA (N=255)

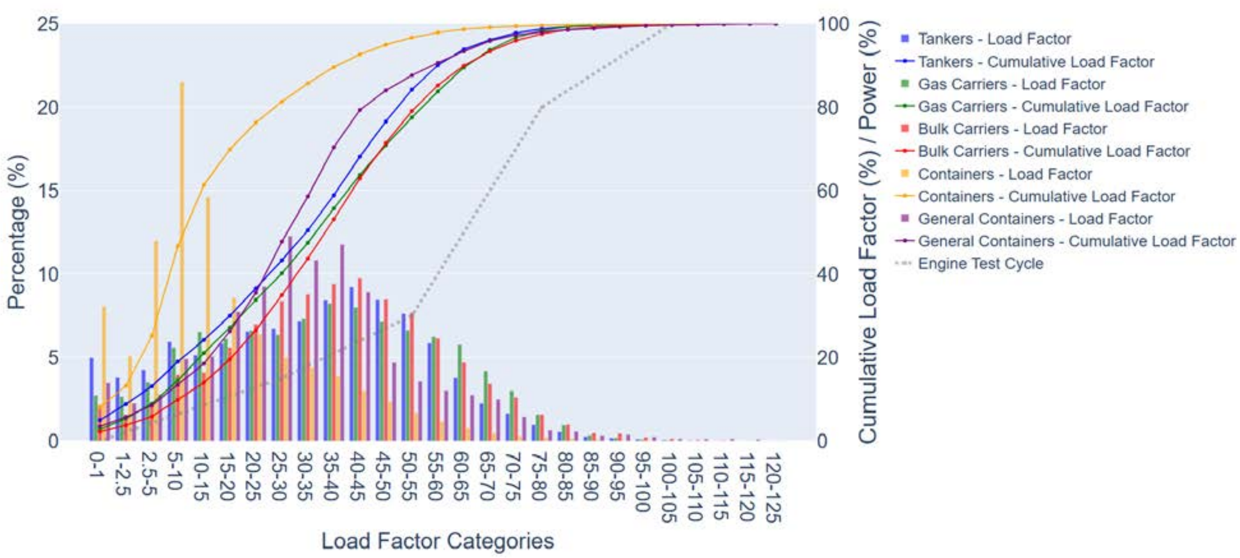


FIGURE 35. ESTIMATED DUTY CYCLE BY SHIP TYPE FOR TIER II SHIPS IDENTIFIED WITHIN THE NORTH AMERICAN ECA (N=4,040)

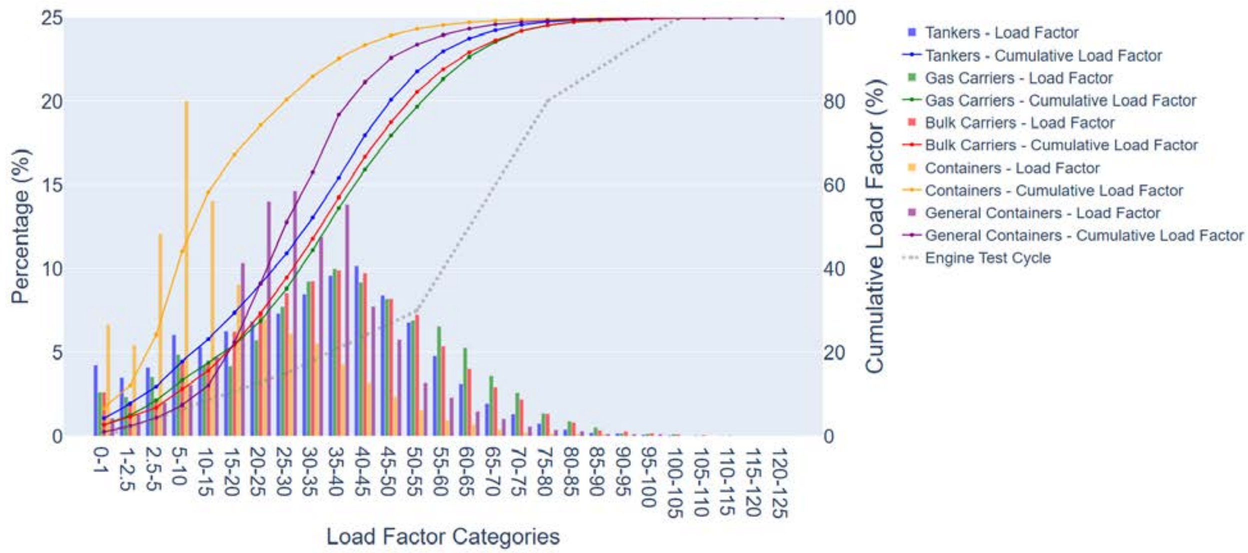


FIGURE 36. ESTIMATED DUTY CYCLE BY SHIP TYPE FOR TIER I SHIPS IDENTIFIED WITHIN THE NORTH AMERICAN ECA (N=1,960)

The duty cycle was also investigated from a regional perspective. These duty cycles are presented in Figure 37 to Figure 39. In general, these duty cycles do not deviate from the aggregate duty cycles, but there are a few exceptions that are discussed in the following paragraphs.

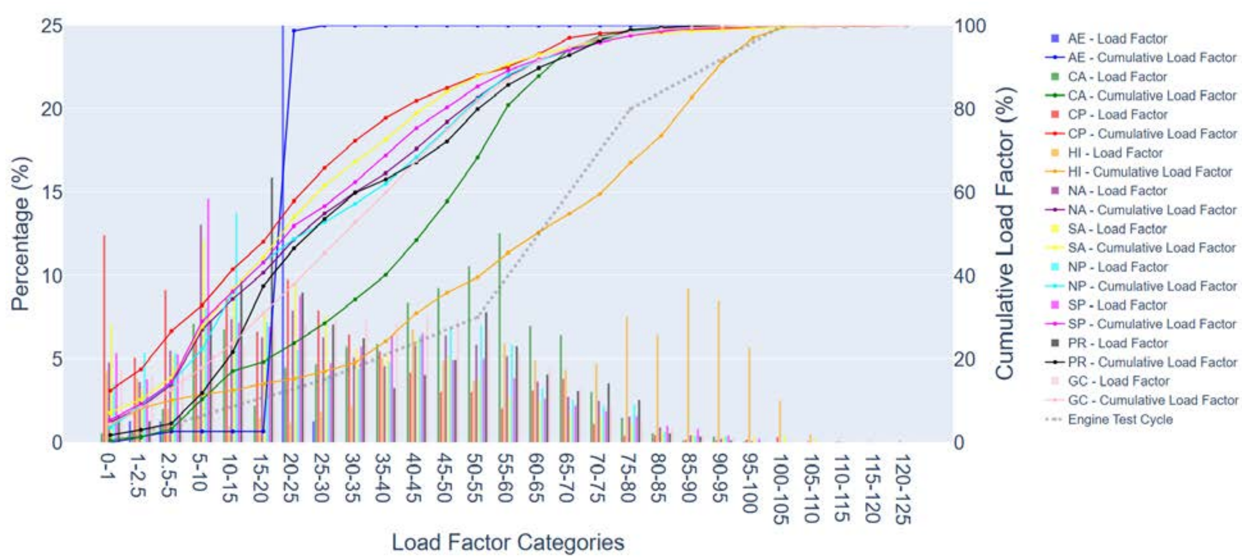


FIGURE 37. ESTIMATED DUTY CYCLE BY REGION FOR ALL TIER III SHIPS IDENTIFIED WITHIN THE NORTH AMERICAN ECA (N=255)

The Tier III duty cycles show the largest deviations between regions. This may be due to the small population size. A small population size can allow a single ship to greatly skew the average if it is being used in an atypical manner. The Hawaii and Alaska East regions had the lowest number of samples. These counts can be found in Figure 23. The Tier III duty cycle was impacted most because it had the least amount points. The Tier II and Tier I duty cycles did not appear to be impact because they had more data. Dividing them by region had less of an impact.

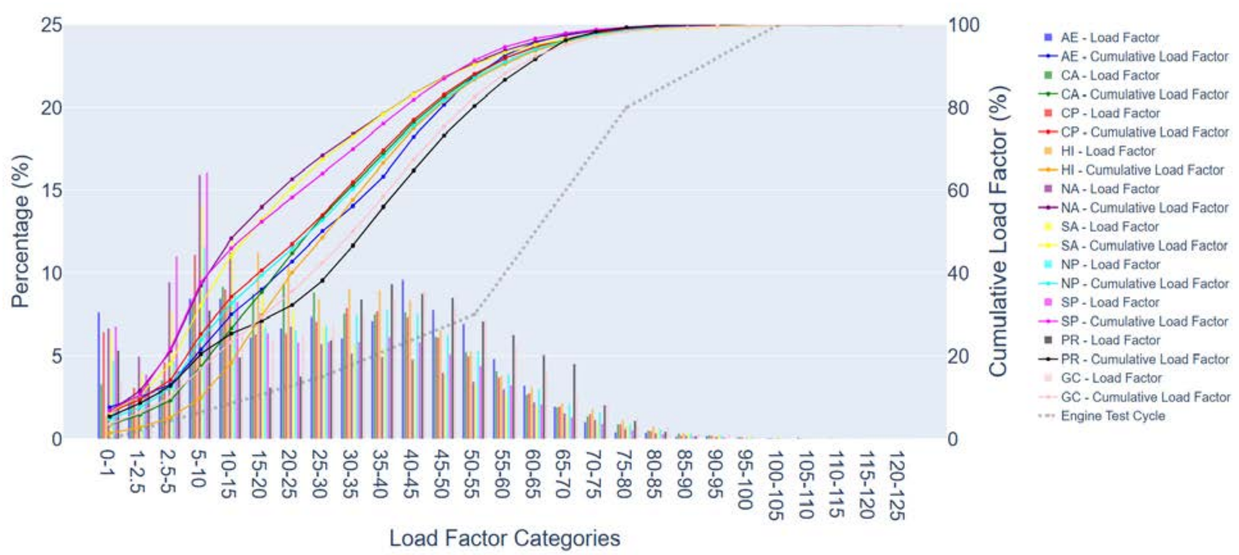


FIGURE 38. ESTIMATED DUTY CYCLE BY REGION FOR ALL TIER II SHIPS IDENTIFIED WITHIN THE NORTH AMERICAN ECA (N=4,040)

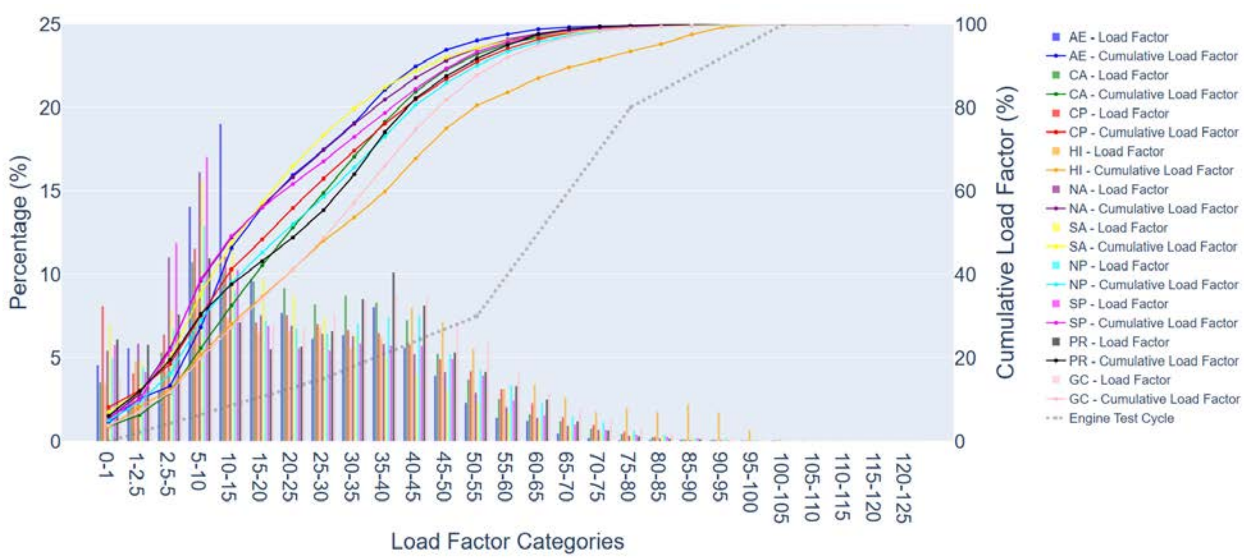


FIGURE 39. ESTIMATED DUTY CYCLE BY REGION FOR ALL TIER I SHIPS IDENTIFIED WITHIN THE NORTH AMERICAN ECA (N=1,960)

As shown in the above graphs, the operating duty cycles of ships within this dataset are very different from the certification test duty cycles. This means that the certification test cycles are not representative of actual ship operation. The impact of these deviations should be investigated to ensure that regulations are achieving the goals for which they were developed.

3.0 CATEGORY 3 MARINE DIESEL ENGINE TECHNOLOGY ASSESSMENT

3.1 Tier III Ships in the U.S. ECA

Details on ships entering the U.S. ECA in 2022 were obtained from the US Coast Guard AIS data. Each IMO number in the AIS database was cross referenced to the Clarksons vessel database to identify the ship. The Clarksons database contained the main engine manufacturer and some performance details such as the power and IMO emission level registration. Nine hundred and eighty-six (986) ships were identified as Category 3, Tier III, vessels. Over ninety-seven (97) percent of these Tier III ships were powered by 2-stroke engines.

There are two main pathways to achieve Tier III NO_X levels for 2-stroke marine diesel engines: exhaust aftertreatment in the form of selective catalytic reduction (SCR) or application of exhaust gas recirculation (EGR). The SCR system can be placed prior to the turbine (high-pressure, HP-SCR) or after the turbine (low-pressure, LP-SCR).

The main advantage of HP-SCR is that this location has higher exhaust temperatures needed for efficient SCR operation. The main disadvantage is the space constraint of having to place the SCR between the exhaust receiver and the turbine inlet. LP-SCR offers greater flexibility in the placement of the SCR but has the disadvantage of low exhaust temperatures after the exhaust has been expanded through the turbine, impacting the SCR efficiency and range of operation.

HP-SCR, where the SCR is placed before the turbine (high pressure side of turbine), seems prevalent when looking at options on the MAN Computerized Engine Application System (CAES) website (MAN Energy Solutions, n.d.), but this really depends on the size of the engine, with some of the largest engines having the LP-SCR as the only option.

Figure 40 illustrates the Tier III ship break down by fuel type. While there are a significant number of dual fuel ships, over seventy (70) percent of Tier III ships were strictly diesel fuel powered. It should be noted that dual fuel engines using early cycle, low-pressure, gas injection, with a small amount of diesel pilot can meet Tier III NO_X emissions without SCR but generally utilize EGR to reduce the tendency to knock. The WinGD X-DF and the MAN ME-GA engines are examples of engines using low-pressure gas injection. The MAN ME-GA product line was discontinued in late 2024 (Snyder, 2024) due to upcoming IMO methane emission regulations. Late cycle, high-pressure, injection of gas with a small diesel pilot will generally need EGR or SCR to meet Tier III NO_X levels. A comparison of the emissions of each approach relative to diesel only is shown in Figure 41 where the DF/gas LP represents the low-pressure gas injection approach, and the GD/gas HP represents the "gas diesel" high-pressure gas injection approach. As illustrated, the DF/gas LP approach significantly reduces engine out NO_X emissions and can achieve Tier III NO_X levels without SCR. The GD/gas HP approach would require similar NO_X countermeasures as the diesel (SCR or EGR) to meet the Tier III NO_X limits.

The Tier III ships operating in the US ECA were predominately powered by MAN engines as illustrated in Figure 42. MAN reported reaching 1,000 Tier III engine orders in 2021 of which 25-percent were EGR solutions and 75-percent were SCR (Søholt, 2021). The split between EGR and SCR solutions continues to evolve (in 2022 MAN reported EGR solutions were 36-percent of the market (Blenkey, 2022), but SCR continues to be the primary path. As an example of the variety of options to meet Tier III, the MAN 60ME engine was selected. Figure 43 illustrates the various engine configurations that are Tier III compliant (MAN Energy Solutions, n.d.)**Error! Bookmark not defined.**. This includes a DI (diesel) configuration as well as dual fuel configurations: GI (gas injection LNG), LGIM (liquid gas injection methanol), LGIP (liquid gas injection propane), and GIE (gas injection ethane). The DI, GI, and LGIM versions are offered with either HP-SCR or various versions of EGR. The LGIP and GIE versions are offered with HP-SCR only.

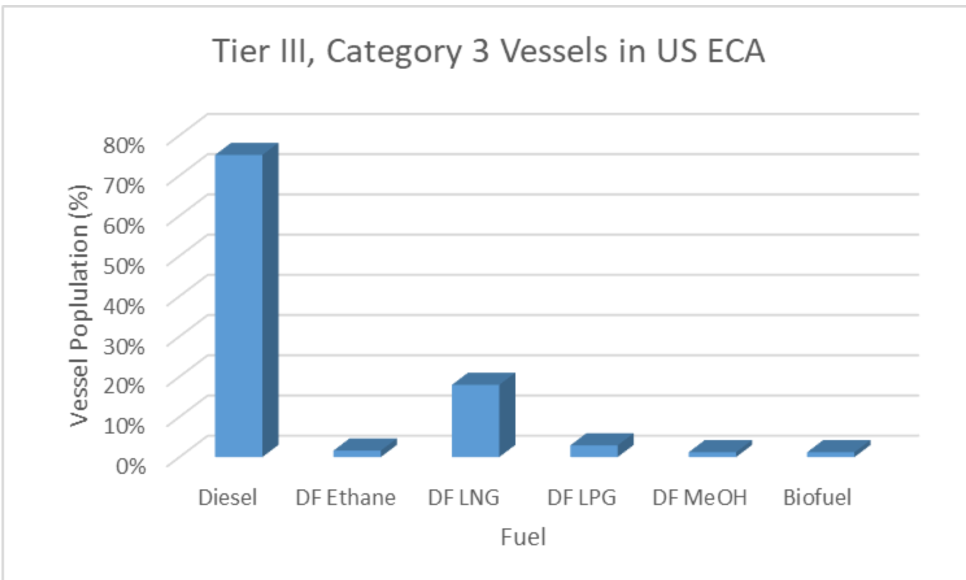


FIGURE 40. TIER III, CATEGORY 3, VESSELS IN THE US ECA

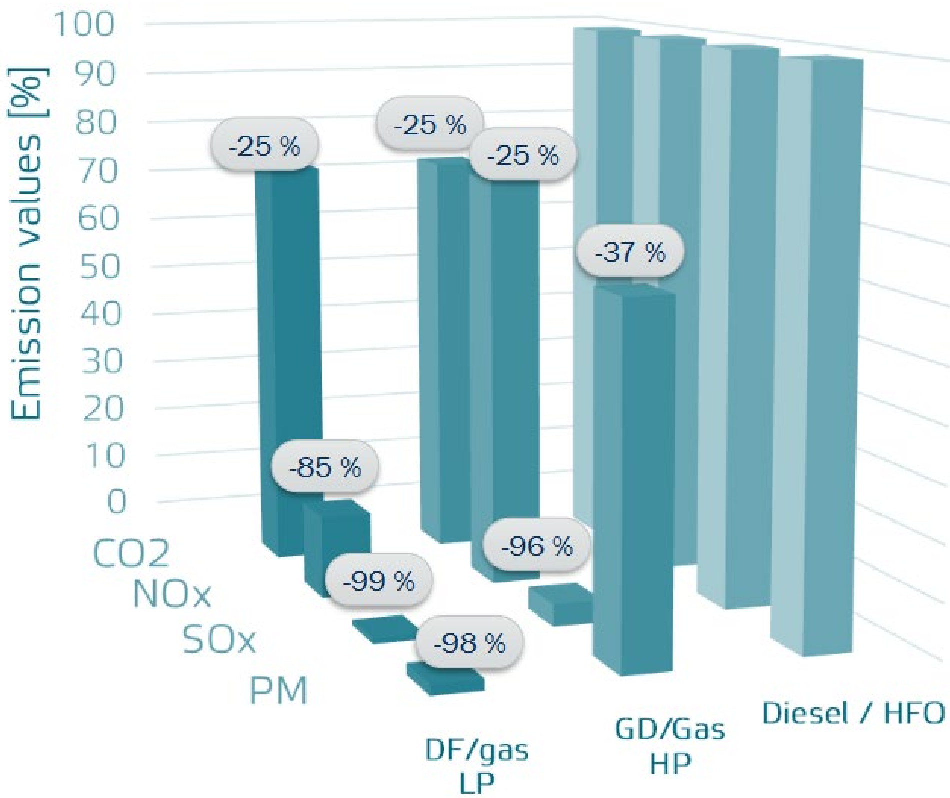


FIGURE 41. COMPARISON OF EMISSIONS FROM TWO DUAL FUEL TECHNOLOGY APPROACHS RELATIVE TO DIESEL (WERNER, 2019)

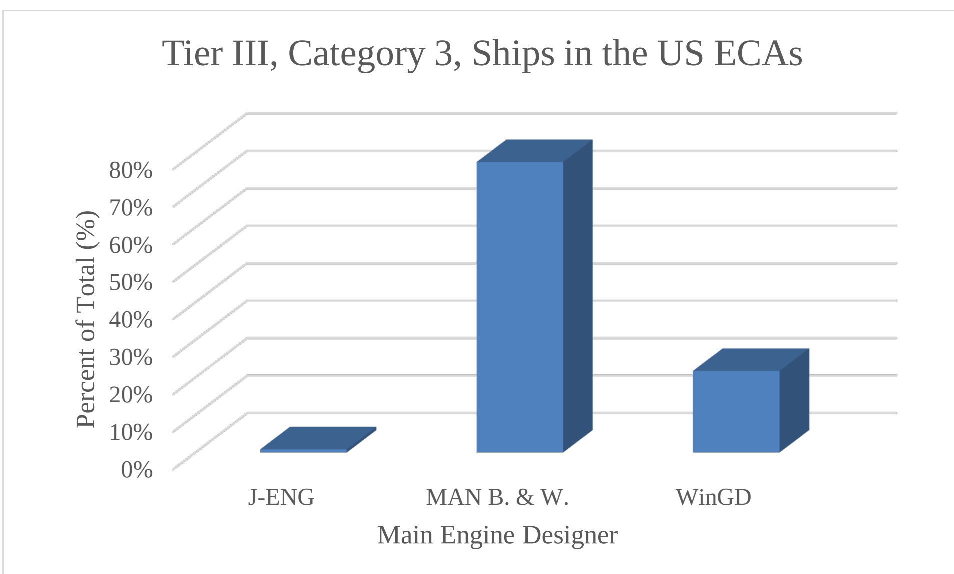


FIGURE 42. MAIN DIESEL ENGINE DESIGNER

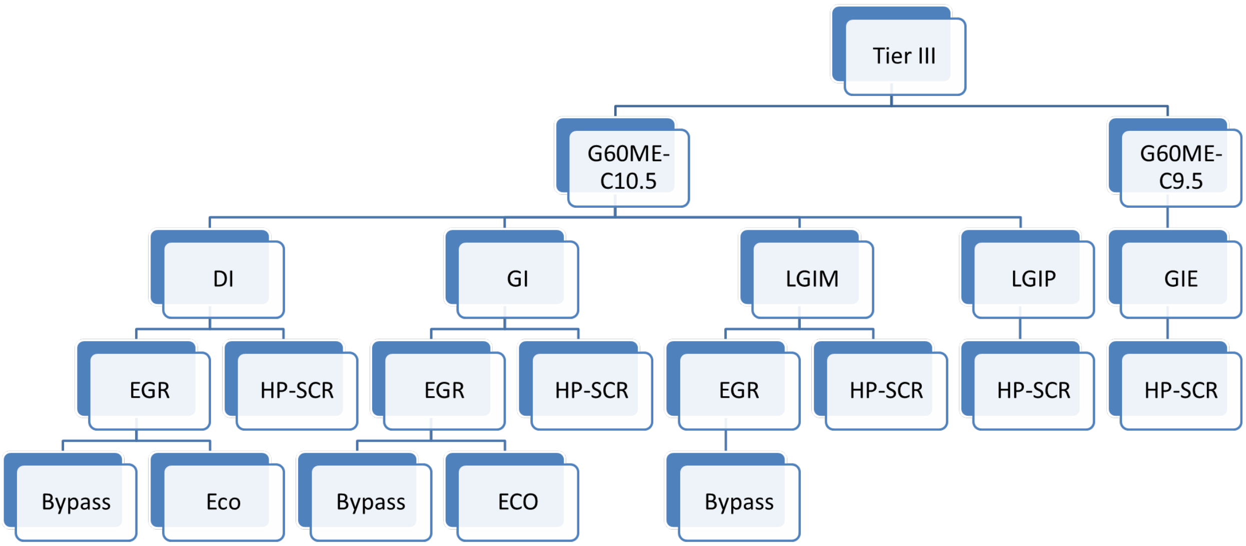


FIGURE 43. TIER III OPTIONS FOR MAN 60ME ENGINE

3.2 Two-stroke Engine Overview

There are two main challenges for using SCR with 2-stroke engines and fuel that contains sulfur. First, the SCR NO_X conversion efficiency is dependent on temperature, with lower temperatures having lower conversion efficiency. Second, the sulfur in the fuel results in combustion products that can form ammonia bisulfate (ABS) in the exhaust by reacting with water and ammonia (reductant required for SCR operation). The kinetics for ammonia bisulfate formation are dependent on the concentration of the reactants, pressure, and temperature. More details on ABS formation kinetics and rates is given in Section 4.4 of this report. High exhaust temperatures are required to prevent the formation of ABS. Diesel engines in general, and 2-stroke engines in particular, tend to have low exhaust temperatures at light loads due to very lean operation (high air-fuel ratios). Additional discussion of ABS formation can be found in Section 5.0.

Typical 2-stroke engine components and operation are depicted in Figure 44. Key features include:

1) Intake ports built into the liner fix the intake opening and closing events.
2) Single exhaust valve per cylinder hydraulically driven on modern engines.
3) Exhaust gases, collected in manifold, drive a turbocharger.
4) Turbocharger compressor supplies air through the charge air cooler and water mist collector to the scavenge air receiver and then the cylinders.
5) Positive differential pressure ($P_{in} – P_{ex}$) required to force air into the cylinder.
6) At low load, exhaust energy is not sufficient to drive the turbocharger and create boost, requiring the addition of an auxiliary blower to supply the scavenge air as illustrated in Figure 45 and Figure 46.

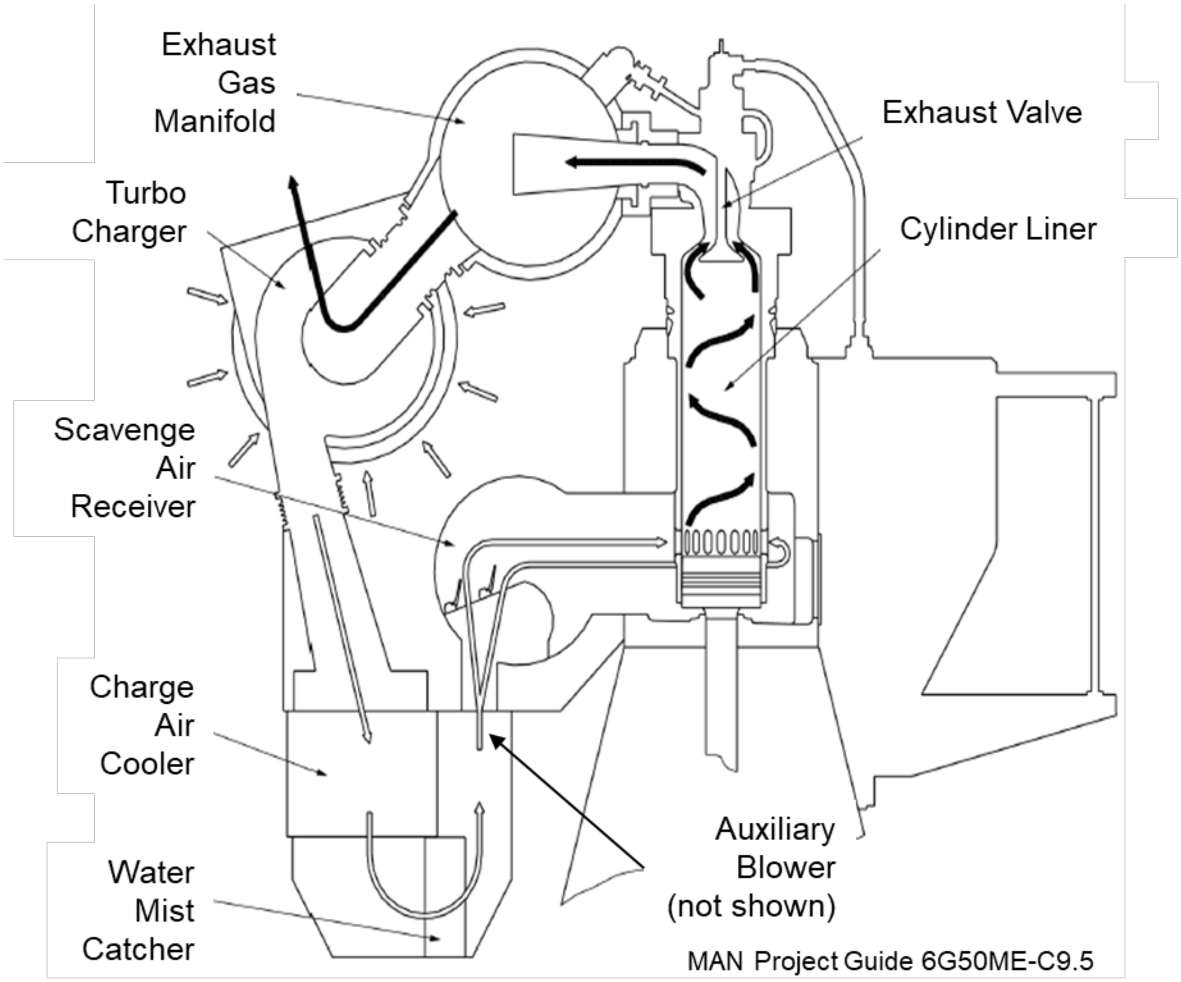


FIGURE 44. TWO-STROKE ENGINE LAYOUT AND OPERATION (MAN ENERGY SOLUTIONS, 2021)

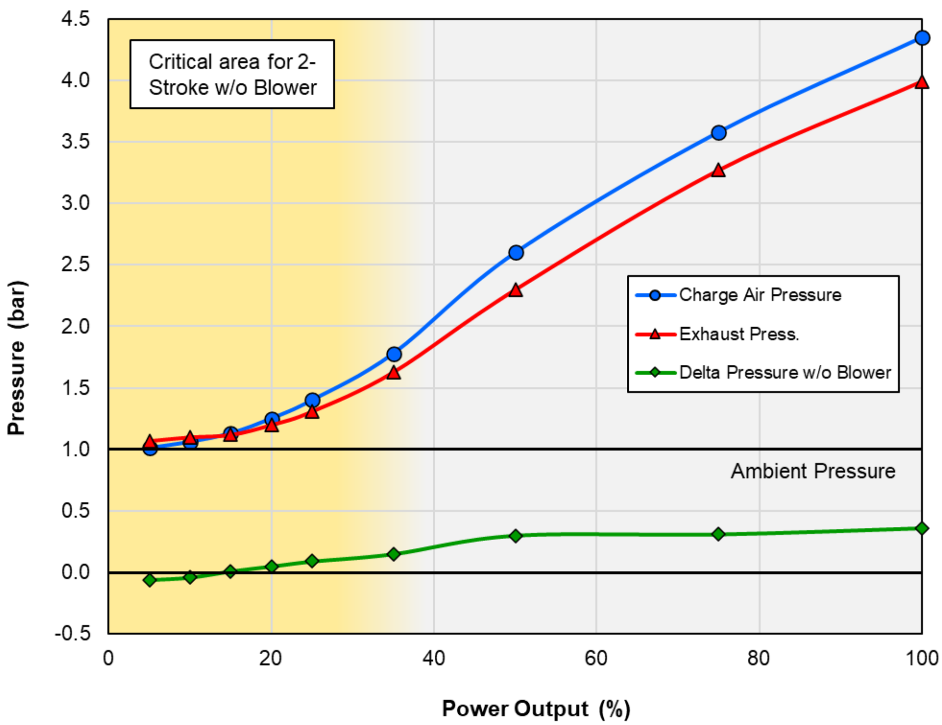


FIGURE 45. INTAKE AND EXHAUST PRESSURE CHARACTERISTICS OF TYPICAL 2-STOKE ENGINE (ILLUSTRATION ONLY)

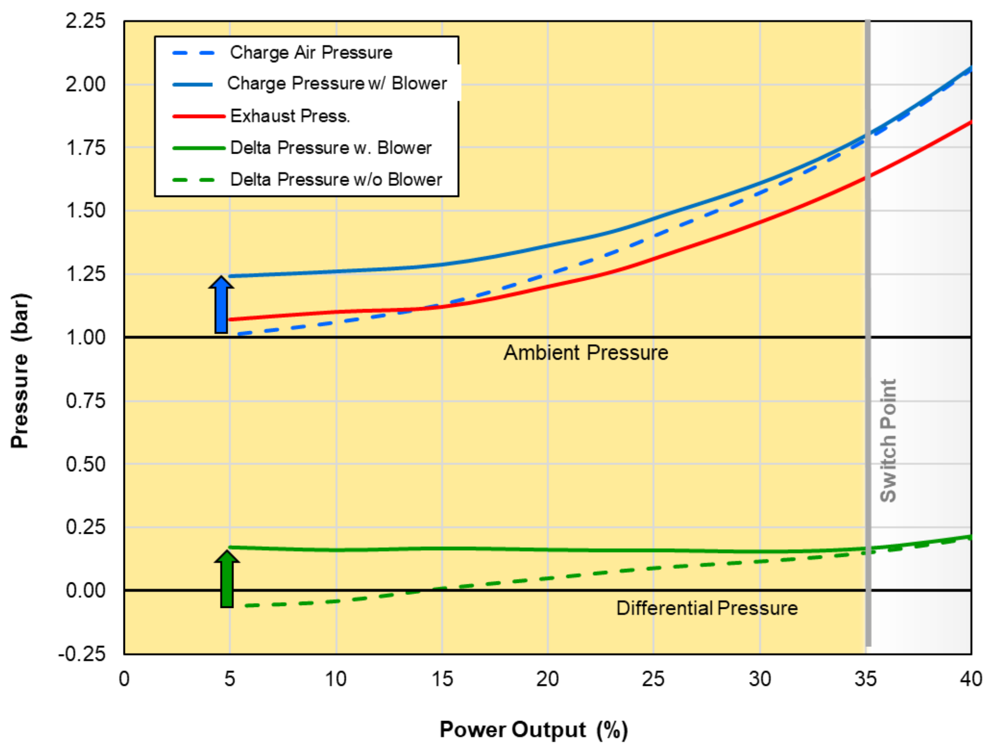


FIGURE 46. INTAKE AIR PRESSURE IMPROVEMENT WITH AUXILIARY BLOWER (ILLUSTRATION ONLY)

Typical exhaust temperatures for a range of Tier II WINGD engines are shown in Figure 47 (WinGD, n.d.). The legend refers to the bore diameter, ranging from 52-92 cm. As shown, turbine inlet temperatures for 50-percent load and above are well above 300°C for avoidance of ABS formation and for high SCR conversion efficiency. These temperatures are favorable for HP-SCR. In contrast, the turbine outlet temperatures are much lower and present a challenge for LP-SCR operation and avoidance of ABS formation. More detail regarding ABS formation rates and temperatures is given in Section 4.4 of this report.

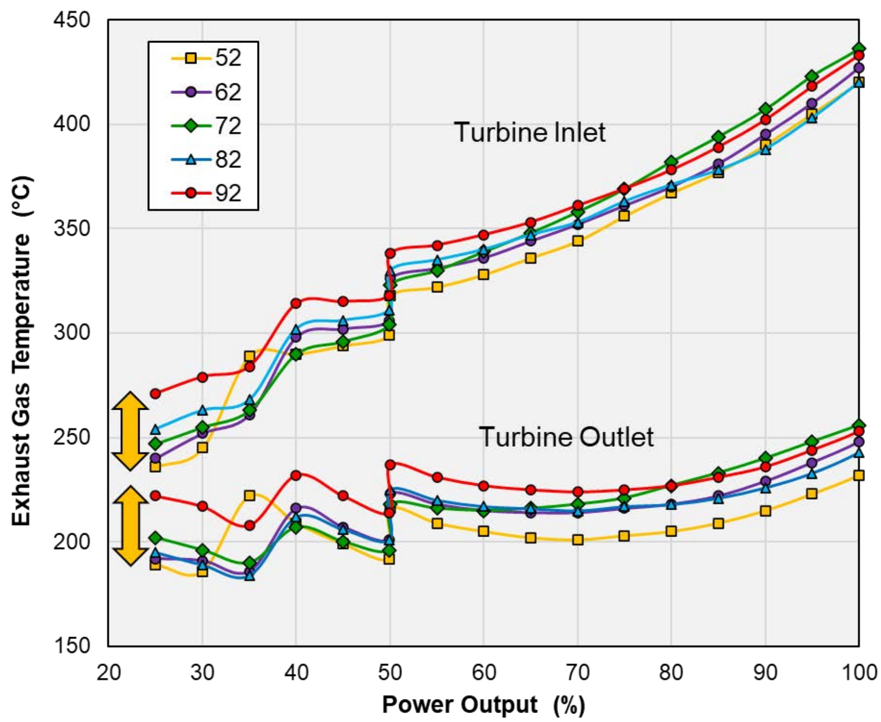


FIGURE 47. EXHAUST TEMPERATURE RANGE FOR TIER II ENGINES

3.3 Current Tier III Thermal Management Technologies

To meet the Tier III NO_X emission target with SCR, engine manufacturers have modified the engine performance to raise exhaust temperatures to enable HP-SCR operation at loads between 25- and 50-percent, depending on the engine model. A comparison between Tier II and Tier III exhaust temperatures is shown in Figure 48 for the WinGD 8X52 engine (WinGD, n.d.). As shown, Tier III operation has higher exhaust temperatures below 55-percent load. This is accomplished with a turbocharger bypass valve (wastegate) which reduces flow to the turbine and, correspondingly, the compressor flow and the scavenging air to the cylinder, thus lowering air-fuel ratio (AFR) and raising exhaust temperatures. The turbocharger bypass flow is shown in Figure 49 while the estimated AFR (illustration only) is shown in Figure 50.

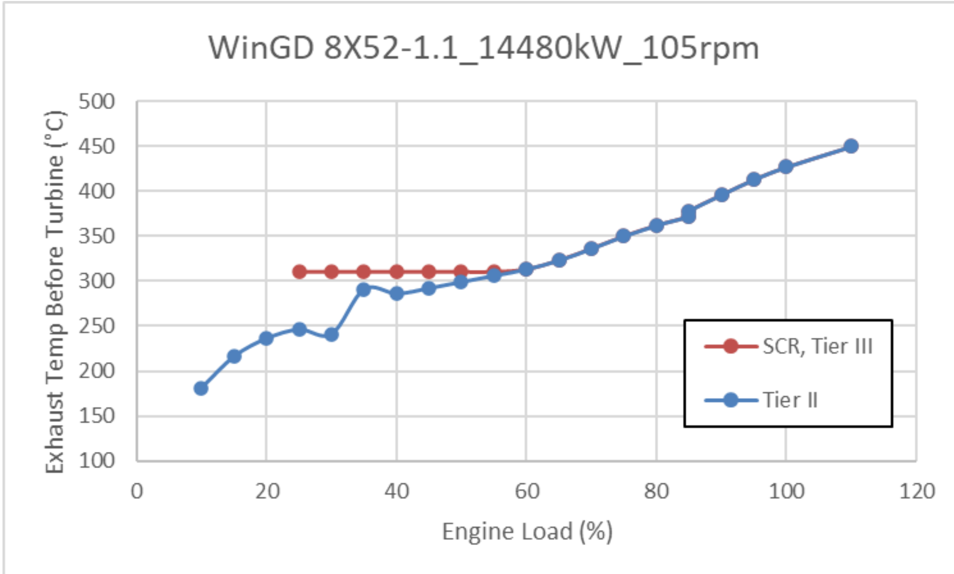


FIGURE 48. COMPARISON OF TIER II AND TIER III BEFORE TURBINE EXHAUST TEMPERATURE FOR WINGD 8X52 ENGINE

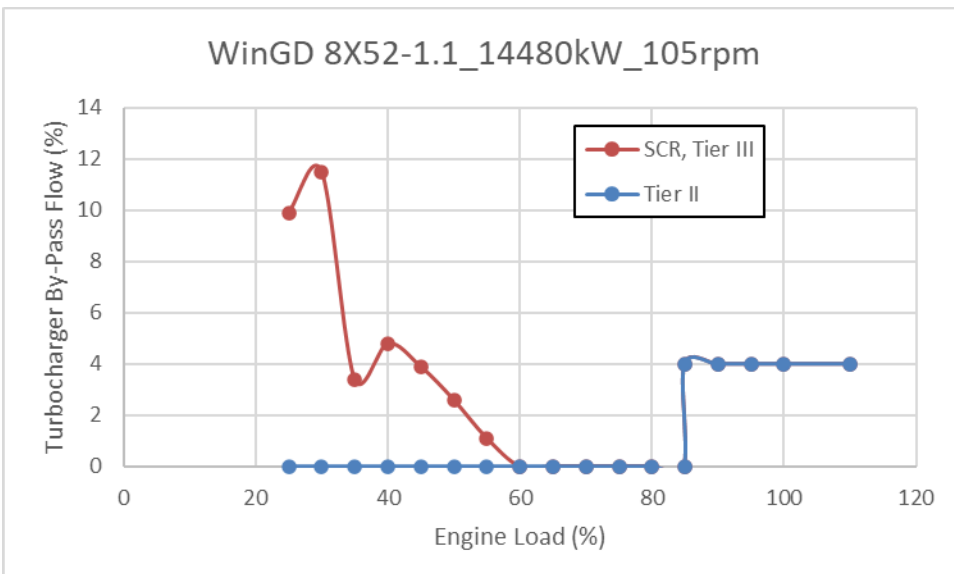


FIGURE 49. TURBINE BYPASS FLOW FOR WINGD 8X52 ENGINE

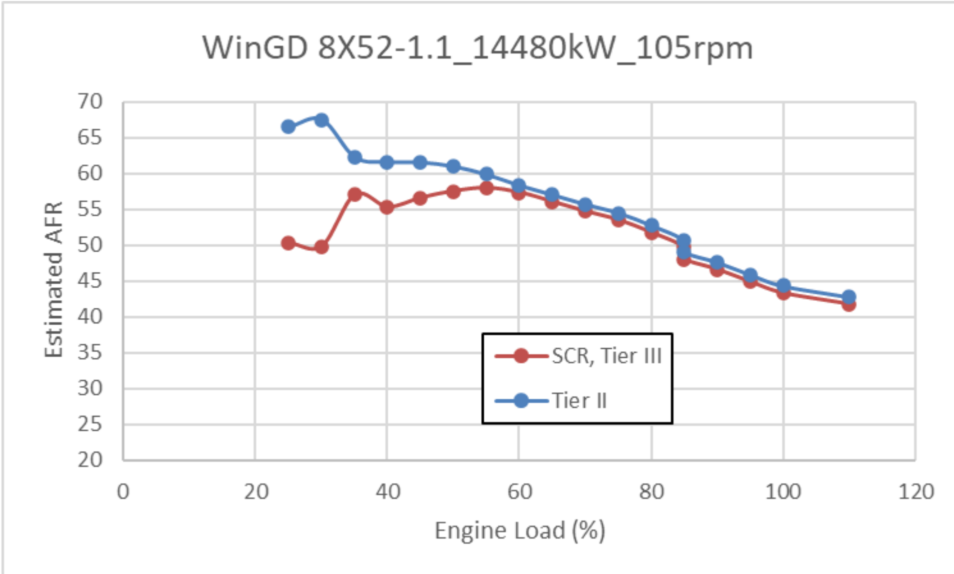


FIGURE 50. ESTIMATED AFR FOR TIER II AND TIER III OPERATION

In another example, Hitachi Zosen Corporation published results for a MAN engine using HP-SCR down to 8-percent load (Fujibayashi, T., et. al., 2013). At light loads, cylinder by-pass, which routed air from the scavenging air receiver (compressor outlet) to the turbine inlet, was used to reduce the air flow to the cylinder (bypassing the cylinder while still maintaining the scavenging differential pressure). This resulted in lowering the AFR, which raised exhaust temperatures. This approach was demonstrated to be successful in maintaining approximately 75-percent NO_X conversion down to 8-percent load as shown in Figure 51.

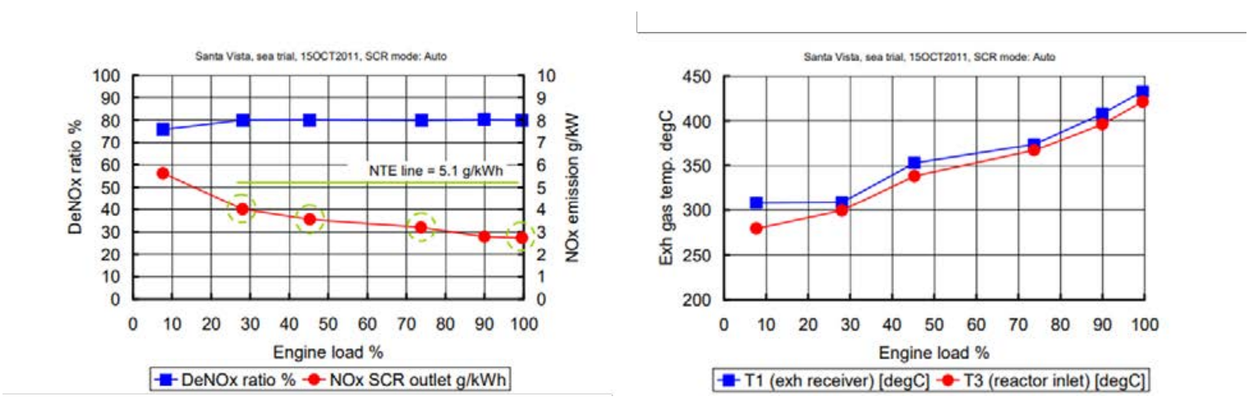


FIGURE 51. SCR PERFORMANCE AND SCR INLET TEMPERATURES (FUJIBAYASHI, T., ET. AL., 2013)

In addition to controlling AFR via either turbocharger or cylinder bypass, modern two-stroke engines also employ common rail injections systems and hydraulically actuated exhaust valves (Kyrtatos, A., et.al., 2016) (Kindt, 2016) providing flexibility over injection timing and exhaust valve opening and closing. Either feature can be tuned to provide an increase in exhaust energy (and temperature), albeit, with a small increase in fuel consumption.

3.4 Applicability of Heavy-Duty Engine Thermal Management Technologies

Emission regulations for the heavy-duty truck industry have been progressively more stringent since the first regulation in 1974. As such, HD truck emission abatement technology is understandably more advanced than in the present-day marine industry. HD thermal management technologies to optimize and improve SCR light off and performance at light loads are sophisticated and can included (United States Environmental Protection Agency, 2022):

- Intake throttle
- Heated aftertreatment system
- Exhaust flow bypass systems
- Late combustion phasing
- Variable valve actuation
- Cylinder deactivation
- Pre-turbine aftertreatment location
- Aftertreatment insulation
- Use of 0.0015% ULSD fuel

The applicability of each of these technologies to the two-stroke marine diesel engines was considered. Engineering assessment was made based on 8 criteria: feasibility, NO_X reduction potential, space consumption (need for space to install the technology), benefit at high loads, reliability, Cost – CAPEX, cost OPEX, ability to retrofit, and scalability for different size engines. Table 5 provides a brief description of each evaluation criteria.

TABLE 5. THERMAL MANAGEMENT TECHNOLOGY EVALUATION CRITERIA

Feasibility	The degree of challenge to implement the technology and applicability.
NO_X Reduction Potential	How far does the technology extend the SCR operating range?
Operating Range	Is this technology beneficial at higher loads as well as low loads?
Integration	How easy is it to integrate and how much space does it take?
Reliability	How does it impact overall reliability? How reliable is the technology?
CAPEX	Relative to today's Tier III systems, does it increase the CAPEX expenditure? Additional components?
OPEX	Is there a BSFC penalty, fuel cost increase, or cost of additional fluids?
Retrofit	Can the system be easily retrofitted on Tier II engines?
Scalability	Can this be applied to small and large marine diesel engines?

3.7.1. Intake Throttling/AFR Reduction

Intake throttling is used in the HD engine industry to reduce the air flow to the cylinder and thereby lower AFR and increase exhaust temperatures. Due to the nature of the scavenging process for two-stroke engines, that requires a higher intake than exhaust pressure, intake throttling, used by itself, is not applicable to two-stroke engines. One might envision using a suction blower on the exhaust side of the engine to lower the exhaust pressure (sub-atmospheric) in combination with throttling to control the air flow and scavenging. Some implementations of EGR in two-stroke engines use a blower on the exhaust side to drive EGR so using a similar setup at low load to reduce the exhaust pressure might be feasible. As noted previously, current industry practice includes some form of AFR reduction at loads below 60-percent in the form of either cylinder or turbine bypass and this type of setup might be used along with throttling and an exhaust suction blower to facilitate optimization.

- Feasibility Low for intake throttling, only with exhaust suction blower, High for other methods of AFR reduction
- NO_X Reduction Potential: SCR down to ~10-15 % MCR
- Space Consumption: Suction blower installation
- Benefit at High Loads Only at low loads, no benefit at high loads
- Reliability: Negative influence due to blower maintenance
- Cost - CAPEX Installation cost of blower
- CO2 - OPEX Slightly higher, 3 to 5 % BSFC
- Retrofit Possible
- Scalability Could be scaled to large engines

3.7.2 Aftertreatment System Heating

In the HD engine industry, heat addition is usually accomplished by adding fuel to the exhaust which then oxidizes over a DOC. Tier III marine diesel engines don't currently use a DOC in conjunction with the SCR system because the fuel sulfur limit of the ECA is too high (1000 ppm). Implementing that approach would require additional capital expenditure for the DOC, an additional space requirement, and ULSD fuel. HP-SCR systems are already space constrained so there may not be room to accommodate a DOC. The space constraints of an HP-SCR system would also make a burner challenging. The impact of the additional energy on the turbocharger needs to be considered. Adding energy to the exhaust pre-turbine has the potential of increasing the turbine and compressor work which would generate more boost increasing air flow and tending to increase AFR and reduce exhaust temperature. A turbine or compressor bypass valve may be required to maintain balance and get the desired effect. The energy requirement would represent a BSFC penalty and additional CO_2 emissions.

A burner would be more practical for a LP-SCR system where space is less constrained. Burners are already in use today in marine applications. One example is the use of a burner to heat a portion of the exhaust sufficiently to evaporate the dosed urea prior to recombination with the main exhaust stream (MAN Energy Solutions, 2021).

An electric heater prior to the SCR or an electrically heated catalyst (EHC) might be a possibility, but durability would be a concern when scaled up to large marine engine sizes. In the HD engine industry, 48V e-heaters are primarily used for cold start to facilitate catalyst light off and emission reduction, as well as low load thermal management. Electric grid heating in the exhaust port, much like a grid heater in the intake port for help during cold start on 4-stroke HD diesel engines, might be adaptable for use in marine applications. The space requirement would be small, but the energy and heat transfer needed over a short distance would make this approach challenging.

- Feasibility — Low for HP-SCR systems
- NO_X Reduction Potential: — SCR down to ~5-10 % MCR combined with other thermal management
- Space Consumption: — Minimal space for heater or burner prior to HP-SCR, space available for LP-SCR
- Benefit at High Loads — Low load only, no benefit at high loads
- Reliability: — Negative influence, burner or heater durability
- Cost - CAPEX — Installation cost
- CO2 - OPEX — Slightly higher, 3 to 5 % BSFC
- Retrofit — Challenging depending on the approach
- Scalability — Could be scaled to large engines

3.7.3 Exhaust Flow Bypass System

Exhaust flow bypass is used for HD engines to retain energy in the exhaust instead of expanding it through the turbine assisting with catalyst light off and temperature at low loads. Since this discussion is considering SCR installations that are HP-SCR, installed prior to the turbine, exhaust flow bypass will be of little help to raise SCR inlet temperatures, improving SCR efficiency and reducing NO_X emissions at low load.

- Feasibility — Low for HP-SCR systems

3.7.4 Late Combustion Phasing

Late combustion phasing (retarding the start of fuel injection, SOI) can be helpful in reducing the cycle efficiency shifting energy from piston work (expansion) to the exhaust. The increase in exhaust temperature is expected to be small relative to that required so this approach would be insufficient on its own but may be used in conjunction with other technologies. Another incremental approach toward making the combustion cycle less efficient would be to reduce the compression ratio. Technology for variable compression ratio (VCR) currently exists and might be used in conjunction with late combustion phasing.

- Feasibility — High with existing common rail injection systems

- NO_X Reduction Potential: SCR down to ~ 10 % MCR
- Space Consumption: No, engine internal
- Benefit at High Loads Load low, no benefit at high loads
- Reliability: Slightly lower
- Cost - CAPEX Similar to current Tier III engines
- CO2 - OPEX Slightly higher, 4 to 6 % BSFC
- Retrofit Likely, injector space claim should be similar
- Scalability Could be scaled to large engines

3.7.5 Variable Valve Actuation

Variable valve actuation (VVA) is used in the HD engine industry for active thermal management, altering lift and/or timing of intake and/or exhaust valves. For two-stroke engines, because the intake ports are fixed in the cylinder liner, VVA can only be applied to the exhaust valve. Modern two-stroke engines, as noted above, have hydraulically actuated exhaust valves providing timing flexibility for both opening and closing of the valves. Late closing of the exhaust valve reduces the effective compression ratio and peak cylinder pressure (useful at high loads) while early opening of the exhaust valve transfers energy from work to the exhaust (useful at low loads).

Early exhaust valve opening (EEVO) has multiple effects. Since EEVO reduces the expansion ratio (ratio of the cylinder volume at EEVO to the cylinder volume at TDC), the cycle efficiency would be decreased which would increase the BSFC, requiring more fuel at a given power level. If air flow were to remain constant, the additional fuel would reduce AFR and contribute to higher exhaust temperatures. However, the additional exhaust energy available to the turbocharger can potentially increase boost and lead to higher airflow, increasing AFR. The turbocharger response is highly dependent on where the operating conditions fall within the turbocharger operating range (turbine and compressor maps). At the lightest loads, the turbocharger is generally not functioning as an additional blower is required. At moderately light loads, the turbocharger response could be significant and turbine bypass (wastegate) or cylinder bypass could be required to optimize tradeoffs affecting BSFC, scavenging, AFR, and exhaust temperatures.

Timing of the exhaust valve closing, in essence, controls the time available for the piston to force gases from the previous cycle out of the cylinder during the compression event. Later EVC reduces the compression work and in-cylinder residual exhaust gas. Earlier EVC traps more residual exhaust gas and increases the compression work. Trapping more residual exhaust gas leads to higher in-cylinder temperatures and ultimately, higher exhaust temperatures. Optimization of the exhaust valve events is critical to low load exhaust temperatures.

- Feasibility High with existing hydraulically controlled exhaust valve

- NO_X Reduction Potential: SCR down to 10 to 15 % MCR, could extend to 5% combined with other thermal management
- Space Consumption: No, engine internal
- Benefit at High Loads No limit, most benefit at low load
- Reliability: Slightly lower
- Cost - CAPEX Low, only software, similar to current Tier III engines
- CO2 - OPEX Slightly higher, 2 to 4 % BSFC
- Retrofit Likely, exhaust valve space needed is similar
- Scalability Could be scaled to large engines

3.7.6 Cylinder Deactivation

The use of cylinder deactivation in modern two-stroke marine diesel engines is possible. Common rail injection systems enable fuel shutoff on a cylinder basis and the VVA system can deactivate the exhaust valve. Deactivating one or more cylinders would increase the load of the firing cylinders and the overall exhaust temperature due to the reduction in AFR. Increasing the load of some cylinders may also lead to an efficiency improvement, reducing CO_2 emissions.

Cylinder deactivation on a 16-cylinder, EMD, locomotive uniflow two-stroke engine at a low speed, low power, condition (referred to as Notch 1) was demonstrated recently by Fritz and Riley (Fritz & Riley, 2024). The results are shown in

Figure 52. The figure illustrates multiple strategies that were applied to achieve the desired exhaust temperatures. The engine was equipped with two Roots blowers. The initial step taken was to deactivate one of the blowers while operating on 16 cylinders, reducing the AFR from ~160 to ~90:1. Deactivating 8- and 12-cylinders lead to higher exhaust temperatures. Additionally, while operating on 4-cylinders, the injection timing was retarded by 8 degrees, from 4°BTDC to 4°ATDC which yielded a slight increase in exhaust temperature. Additional backpressure was added to reduce scavenging, which led to modest exhaust temperature increase. Finally, the AFR was further reduced, while operating on 4- and 8-cylinders, by bleeding off air after the compressor (similar to the cylinder bypass concept) resulting in a reduction in AFR and corresponding increase in exhaust gas temperature. For slow speed inline marine diesel engines, torsional vibrations would need to be evaluated to check if cylinder deactivation would be feasible.

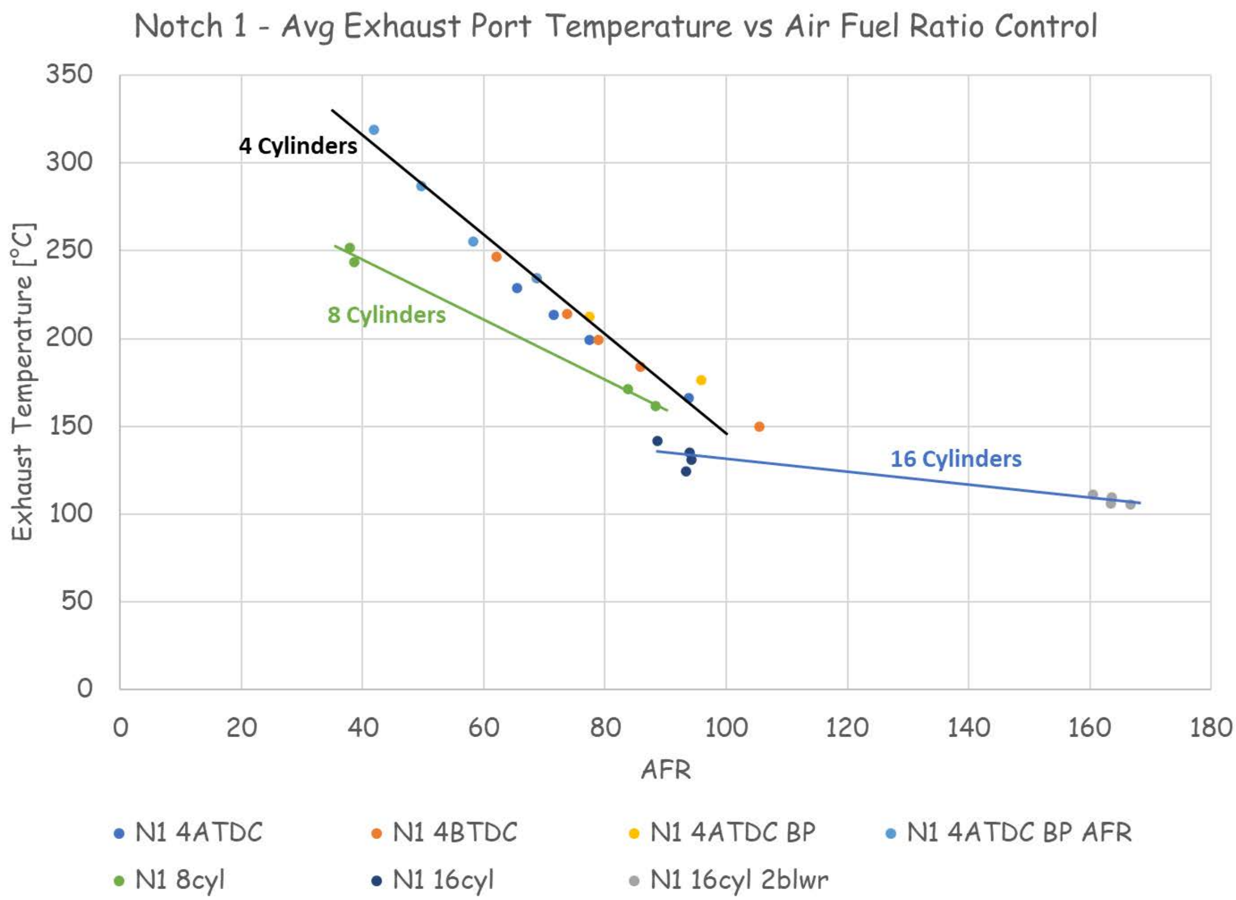


FIGURE 52. EFFECT OF CYLINDER DEACTIVATION ON EXHAUST TEMPERATURES OF A 16-CYLINDER, EMD LOCOMOTIVE TWO-STROKE ENGINE AT LIGHT LOAD (FRITZ & RILEY, 2024)

- Feasibility — Possible, but torsional vibrations need to be checked
- NO_X Reduction Potential: — SCR down to 10 to 15 % MCR, could extend to 5% combined with other thermal management
- Space Consumption: — No, only software
- Benefit at High Loads — Low load, no benefit at high load
- Reliability: — Negative influence possible
- Cost - CAPEX — Low, only software
- CO2 – OPEX — Slightly lower, 1 to 2 % BSFC improvement
- Retrofit — Likely, injector, exhaust valve systems and controller
- Scalability — Could be scaled to large engines

3.7.8 Pre-turbine Aftertreatment Location

Close coupling of the aftertreatment system in heavy and light-duty applications improves catalyst light off and thermal management. However, there are no significant instances of pre-turbine aftertreatment systems in production for on-highway engines. However, in two-stroke C3 marine applications, manufacturers are placing the SCR system pre-turbine, when possible, to achieve the highest exhaust temperatures at the SCR inlet. This approach appears to be most prevalent for smaller engines. In 2019, MAN reported the trends for LP-SCR, HP-SCR, and EGR by engine size (Struckmeier, D., et al., 2019), which can be found in Figure 53. A typical HP-SCR layout locates the SCR separate from the engine but in the engine room as illustrated in Figure 54. Recommendations from the initial development and demonstration of HP-SCR included keeping the piping as short as possible and insulating the pipes (Fujibayashi, T., et. al., 2013). Designs have progressed and WinGD now offers an integrated SCR system (Kyrtatos, A., et.al., 2016) (Spahni, M., et. al., 2023), called iSCR for about 39-percent (WinGD, n.d.) of diesel engine models where the SCR is placed on the engine as shown in Figure 55. The exhaust flow path for the WinGD iSCR system is shown for Tier II and Tier III modes in Figure 56. (Spahni, M., et. al., 2023). The flow path is controlled by shutoff valves in the two flow paths. In Tier II mode, the exhaust is routed directly from the exhaust manifold to the manifold outlet. In Tier III mode, the exhaust is routed from the exhaust manifold (top) to the bottom through passages at the ends of the manifold. This exhaust then flows through two SCR catalysts back to the center of the unit and then upwards to the exhaust manifold and manifold outlet. Urea dosing takes place in the exhaust port runner with air assisted injectors.

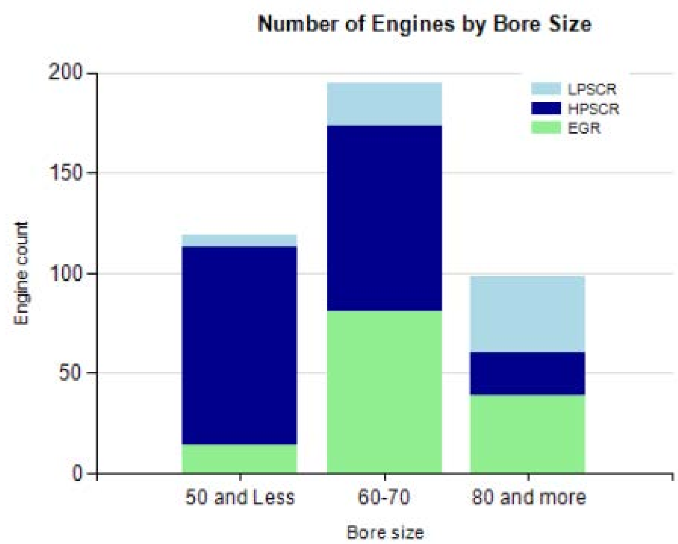


FIGURE 53. CHOICE OF IMO NO_X TIER III STRATEGY FOR MAN TWO-STROKE ENGINES IN RELATION TO ENGINE SIZE (STRUCKMEIER, D., ET AL., 2019)

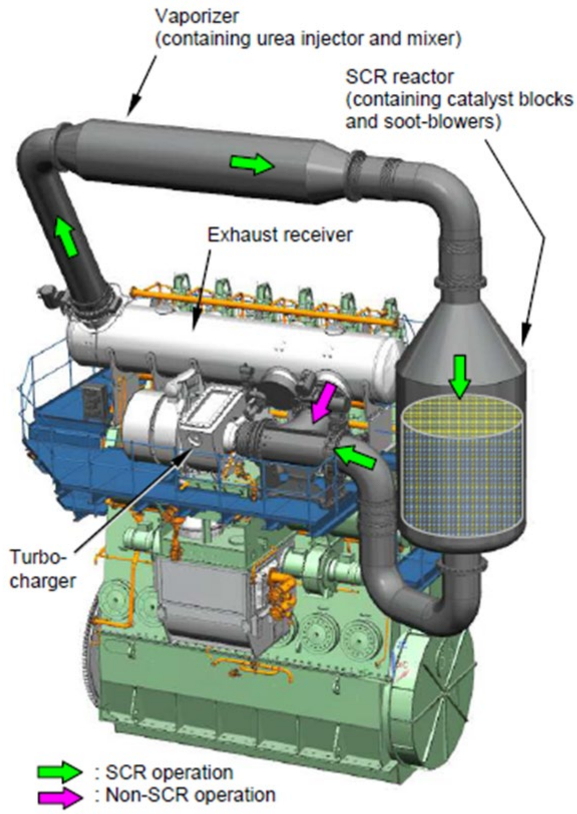


FIGURE 54. EXAMPLE OF HIGH-PRESSURE SCR SYSTEM LAYOUT (FUJIBAYASHI, T., ET. AL., 2013)

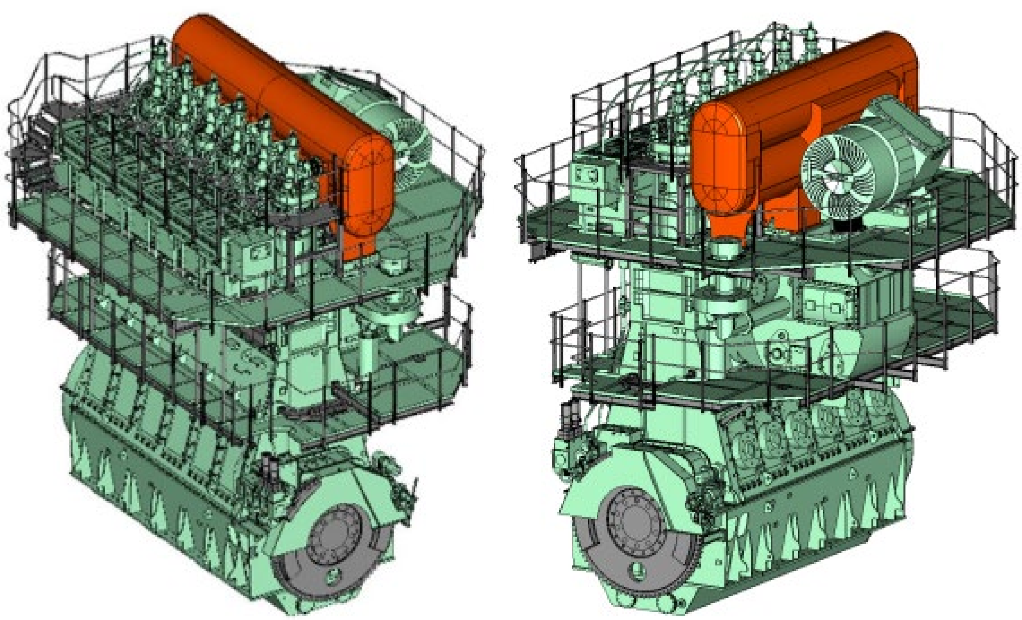
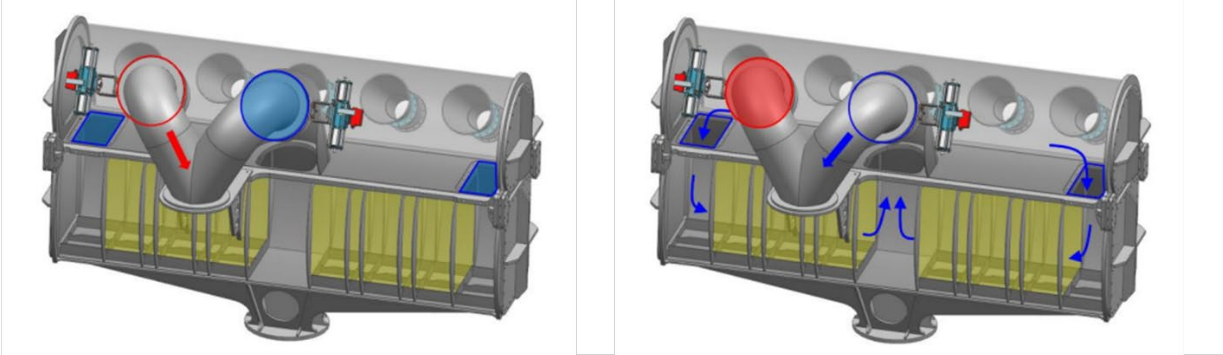


FIGURE 55. EXAMPLE OF WINGD ISCR, INTEGRATED HPSCR SYSTEM (KYRTATOS, A., ET.AL., 2016)



iSCR in Tier II mode – Tier II valve (red) open, Tier III valves (blue) closed

iSCR in Tier III mode – Tier II valve (red) closed, Tier III valves (blue) open

FIGURE 56. EXHAUST FLOW PATH OF WINGD ISCR IN TIER II (LEFT) AND TIER III (RIGHT) MODES (SPAHNI, M., ET. AL., 2023)

• Feasibility	High, already implemented on many engines
• NO_X Reduction Potential:	Needs other technologies to extend SCR operating range, SCR down to ~ 5-10% with thermal management
• Space Consumption:	Constrained
• Benefit at High Loads	Beneficial for mid-load operation
• Reliability:	No influence
• Cost - CAPEX	Same as current
• CO2 – OPEX	Dependent on other technologies implemented
• Retrofit	Possible, Exhaust system and piping to turbocharger
• Scalability	SCR volume requirement may limit application

3.7.9 Aftertreatment Insulation

For stationary engines, insultation of exhaust piping is typically done as a safety measure to reduce the outer surface temperature. If the piping is not insulated, some sort of shielding would be used. For HP-SCR systems, additional insultation or shielding would likely be of no benefit.

• Feasibility	Low

3.7.10 ULSD

The current sulfur limit for diesel fuel in the ECA is 0.1 percent by weight. The sulfur in the fuel ends up primarily as SO_2 in the exhaust, although a small portion (typically around 5%) is oxidized to SO_3 (and H_2SO_4 or a few other oxidized SO_2 byproducts). SO_3 can potentially react with the SCR reductant, NH_3, to form ammonium bisulfate (ABS) at low temperatures. ABS deposition results in catalyst fouling and corrosion over time, and therefore SCR can only be utilized above the ABS condensation temperature, which for current systems at loads above 20% is generally around 300°C. Therefore, for marine SCR systems, avoidance of ABS condensation is one of the primary factors limiting low temperature conversion in the system. Since the formation of ABS is dependent on the concentration of SO_3 as well as temperature, reducing the SO_3 concentration can enable the SCR to operate at a lower temperature, in essence, extending the low load operating zone of the SCR. A more complete analysis of ABS formation is discussed below in Section 4. Switching to ULSD, with a sulfur limit of 15 ppm, would extend the operating range of the SCR system, reducing the ABS formation temperature at light loads to as low as 225°C (which is below the effective temperature of the vanadium-SCR catalysts anyway). This would, in turn, require much less thermal management to operate the SCR at even very light loads in the range of 5%. Ideally, this could be a direct replacement of the current ECA fuel with ULSD, however there would be a cost impact to fuel price. It does not seem practical to carry an additional fuel type for only low load operations in the ECA, so it is likely that this change would have to apply to all fuel used in the ECA. Ultimately this may be cost-benefit analysis weighing the additional cost of ULSD compared to 0.1-percent fuel versus the addition and complexity of a third fuel system.

- Feasibility High, could replace current ECA fuel
- NOx reduction Potential: SCR down to 5% MCR with less thermal management need than other methods
- Space consumption: May require an additional fuel tank
- Benefit at High Loads No limitation
- Reliability: No influence
- Cost - CAPEX Same as current
- CO2 – OPEX Fuel price increase, but some fuel consumption decrease, net increase in cost expected
- Retrofit Yes
- Scalability Could be scaled to large engines

3.5 Other Thermal Management or SCR Technologies

3.8.1 Intake Air Heating

Raising the starting point temperature of the process will raise the temperatures throughout the process. Addition of heat to the intake air would raise the temperature leading to a higher exhaust gas temperature. Typically, the charge air cooler after the compressor is used to remove heat added during the compression of the air. However, at light load, there is no compressor work being done on the fluid, so the temperature rise across the compressor is small. If the charge air cooler water temperature was increased, it would add heat to the inlet air raising the temperature. This could be done with modification of the cooling water circuit to add a heat exchanger for using process steam, if available on board, or installing an electric heater. Approximately, a 1-to-1 temperature increase would be expected. Higher in-cylinder temperatures would lead to some combustion effects such as a short ignition delay and higher NO_X, which could be mitigated by optimizing the fuel injection timing.

- Feasibility High

- NO_X Reduction Potential: SCR down to ~ 5-10 % MCR combined with other thermal management
- Space Consumption: Small
- Operational Range Low load only, no benefit at high load
- Reliability: No influence
- Cost - CAPEX Small
- CO2 - OPEX Slightly higher
- Retrofit Likely
- Scalability Could be scaled to large engines

3.8.2 Direct Use of Ammonia

Urea could be replaced directly by ammonia, or a separate burner or heater could be used to vaporize the urea and decompose it into ammonia. Ammonia is often used on board for refrigeration systems so safety protocols may already exist. A burner for urea decomposition is also in use today on some ships, generally for LP-SCR systems (MAN Energy Solutions, 2021). This approach would allow SCR introduction of ammonia down to ~150°C. However, low temperature SCR performance is not currently limited by low temperature DEF injection limits, especially on HP-SCR systems. Instead, low temperature SCR performance is limited primarily by the risk of ABS formation, which direct NH_3 introduction does nothing to resolve. In addition, the current vanadium-based SCR technology that is preferred on OGVs does not perform well at temperatures below 250°C, so the lower temperature introduction of NH_3 would not yield much performance benefit. The estimated potential for using ammonia is summarized below.

- Feasibility Moderate, safety concerns
- NO_X Reduction Potential: Limited improvement without other thermal management technologies.
- Space Consumption: Additional tank system
- Benefit at High Loads No benefit at high load
- Reliability: No influence
- Cost - CAPEX Installation cost
- CO2 – OPEX Probably slightly higher
- Retrofit Likely
- Scalability Could be scaled to large engines

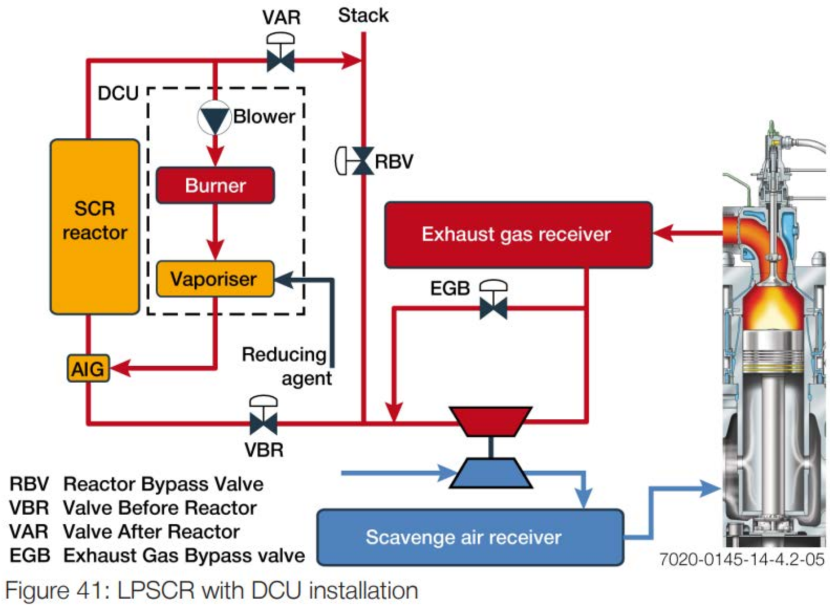


FIGURE 57. ILLUSTRATION OF UREA DECOMPOSITION UNIT (DCU) AND BURNER FOR LP-SCR SYSTEMS (MAN ENERGY SOLUTIONS, 2021)

3.8.3 Cylinder and/or Turbine Bypass

Cylinder bypass and turbine bypass have been mentioned (see sections 3.3 and 3.7.1) previously as examples of AFR management technologies currently in use at loads above ~20-percent to raise exhaust temperatures. These technologies should be considered for extension of exhaust thermal management to operation below 20-percent load, particularly since it would entail mostly additional calibration work with minimal engine modifications. Also, there may be synergy with other technologies (i.e. cylinder deactivation or VVA) that may not be currently in-use or used to the fullest potential.

3.6 Summary

The technologies described above are summarized in the Table 6, below. To visually illustrate the discussion, engineering judgement was used to assign a numeric score to each category for each technology. The numeric score was based on a scale of 1 (unfavorable) to 5 (favorable). Conditional formatting (i.e., color coding) was used to help visualize the rankings. Some comments about the rating process are appropriate.

As discussed above, exhaust bypass and exhaust system insulation are not expected to be feasible technical solutions for increasing exhaust temperatures. Intake throttling might be possible with a suction blower on the exhaust side but is ranked as slightly unfavorable. Heated aftertreatment systems have an unfavorable rating due to questions about durability. The other technologies have favorable ratings with technologies that are currently existing in some form having the highest rating.

In terms of NOx reduction potential, intake throttling and heated aftertreatment were given the same score as for feasibility due to challenges in implementation may hamper the NOx reduction potential. The direct use of ammonia has the least NOx reduction potential as discussed above. The other technologies received a slightly favorable rating.

None of the technologies considered are beneficial at improving SCR performance at high loads so all were rated neutral in this regard.

Some of the technologies discussed are already in use so there is no space or integration issue. These technologies, which included late combustion phasing, VVA, cylinder deactivation, and cylinder or turbine bypass, received a score of 5. Pre-turbine aftertreatment is already in use on some engines, but not on others, so while the in-use cases would score a 5, the other cases would score a 1, resulting in an aggregate of 3. The impact of direct use of ammonia or ULSD on space and integration really depends on the implementation. These could be drop-in technologies, replacing systems already in use or add-on technologies. Without knowledge of the implementation details, these technologies were rated as 3, neutral. Intake throttle, heated aftertreatment, and air heating intake would require additional hardware and corresponding space claim. These technologies were rated as slightly unfavorable, as 2.

Heated aftertreatment systems have the biggest question mark regarding reliability. Whether it is an electrically heated catalyst, a burner prior to the SCR, or an electric grid heater prior to the SCR, the high temperature required imposes a reliability risk. This technology was scored as 1. Other technologies not currently in use, cylinder deactivation and air heating intake were scored as slightly unfavorable for reliability, 2. Late combustion phasing, VVA, cylinder or turbine bypass, and direct use of ammonia are technologies currently in use in some fashion and would have little impact on reliability. Pre-turbine aftertreatment, if compared to applications currently using LP-SCR, would likely improve reliability due to the decreased risk of ABS formation. So this technology was rated as slightly favorable. The use of ULSD would reduce the risk of ABS formation and was rated as 4.

No technology is going to reduce CAPEX so, at best, any technology is neutral and rated a 3. If a given technology is going to include additional expenditure, then the rating will be lower than 3 but would have to be a major expense to receive a rating of 1.

Operating cost, OPEX, is really a judgment on the BSFC penalty or the cost of additional fluids (DEF, ammonia, ULSD). Only cylinder deactivation has the potential for improving BSFC, so it received a favorable rating. Pre-turbine aftertreatment is a “passive” technology that is currently in-use today so the impact on operating cost would be neutral. Other technologies would incur some BSFC penalty or require additional or more expensive fluids so received a below neutral rating. Late combustion phasing and VVA, while effective in increasing exhaust temperatures, may have a less favorable trade-off with BSFC than other technologies so received the lowest rating.

Except for heated aftertreatment systems, all technologies are viewed as retrofittable to Tier II engines. This is reflected by a rating of 4 for these technologies. The use of ULSD was rated 5 since it could be a direct replacement for today's current ECA fuel. Heated aftertreatment systems, depending on the implementation method as discussed above, may pose more of a challenge, which is reflected in its rating.

In general, apart from the pre-turbine SCR, all technologies are viewed as scalable to larger engines. Pre-turbine SCR may be limited for the largest engines due to the volume of the SCR.

TABLE 6. THERMAL MANAGEMENT TECHNOLOGY EVALUATION

Technology	Feasibility	NOx Reduction Potential	Benefit at High Loads	Integration/ Space Consumption	Reliability	Capital Cost	Operating Cost	Suitability for Retrofit	Scalability
Intake Throttle/AFR Reduction	2	2	3	2	2	2	2	4	4
Heated Aftertreatment System	2	2	3	2	1	2	2	2	4
Exhaust Flow Bypass Systems	1	n/a	n/a	n/a	n/a	n/a	n/a	n/a	n/a
Late Combustion Phasing	5	4	3	5	3	3	1	4	5
Variable Valve Actuation	5	4	3	5	3	3	1	4	5
Cylinder Deactivation	4	4	3	5	2	3	4	4	5
Pre-Turbine Aftertreatment	5	4	3	3	4	3	3	4	3
Aftertreatment Insulation	1	n/a	n/a	n/a	n/a	n/a	n/a	n/a	n/a
ULSD	5	4	3	3	4	3	2	5	5
Intake Air Heating	4	4	3	2	2	2	2	4	5
Direct Use of Ammonia	4	1	3	3	3	2	2	4	5
Cylinder or Turbine Bypass	5	3	3	5	3	3	3	4	5

4.0 MARINE DIESEL ENGINE NO_X AFTERTREATMENT AND TECHNOLOGY PACKAGE ASSESSMENT

The previous report section discussed a number of potential technologies that could be applied to Category 3 2-stroke marine primary propulsion engines as a means to achieve further NO_X reductions. To support an analysis of the potential for further NO_X reductions in the ECA, it was necessary to develop a projected technology package that could serve as a modeling case. Ultimately, a set of technology packages based around high pressure (pre-turbine) SCR and thermal management technologies were developed. These projections included different levels of aggressiveness in the application of both SCR and thermal management. This section of the report details the technology choices that were made in the development of these packages, including the background, assumptions, and supporting data used to develop the final projections.

4.1 Exhaust Temperature Profile for Marine 2-Stroke Engines

SCR efficiency is dependent, in part, on the exhaust temperature at the inlet of the SCR system. With the objective of extending the SCR system performance below 25-percent load, an exhaust temperature profile was required in this region. The literature was reviewed to determine data available for estimating the exhaust temperature profile versus load. Unfortunately, most of the published data was at 25-pecent load and higher. Fujibayashi, et al. (Fujibayashi, T., et. al., 2013) published data of SCR system performance and inlet temperatures down to 8-percent load. WinGD published engine performance data which includes exhaust temperature before the turbine in their general technical data (GTD) software (WinGD, n.d.). The WinGD data is at 5- and 10-percent load steps down to 25-percent load. WinGD GTD also provides the nominal exhaust temperature decrease from 25-percent load for 20-, 15-, and 10-percent loads. MAN publishes engine performance data on their website in the form of the MAN Computerized Engine Application System (CEAS) (MAN Energy Solutions, n.d.). While the MAN data does not contain pre-turbine exhaust temperatures, it does contain the SCR inlet temperature for Tier III applications at 25-, 50-, 75-, and 100-pecent load. The pre-turbine temperature generally trends with load with higher loads having higher temperatures. The temperature can vary somewhat depending on the size of the engine and the engine calibration. A comparison of the available data from the three sources is shown in Figure 58. The left figure illustrates the pre-turbine exhaust temperatures in Tier II operation while the graph on the right shows the SCR inlet temperatures. These graphs illustrate three points. First, general trends for different engines are similar with a maximum spread of 50°C in the mid-load range. Second, different calibrations can have up to a 50°C spread in exhaust temperature for the engine model and this is most apparent in the mid-load range. Third, the Tier III temperatures essentially follow the Tier II temperatures until about 50-percent load (~310°C) at which point thermal management maintains a constant temperature down to 25-percent load.

Of the data shown, the WinGD X52 data with the low load turning (LLT) calibration seemed to be on the low range of temperatures and would be the most conservative choice for further analysis. The X52-LLT series engine was selected to provide the nominal exhaust temperature versus load profile. The Tier II and III exhaust temperatures versus load for this engine are shown in Figure 59. The unknown is the Tier III exhaust temperatures below 25-percent load. Regardless of the technology used to maintain temperature below 25-percent, the best-case scenario would be a gradual decline in temperature to 260°C at 5-percent load., which is illustrated in Figure 59. For analysis purposes, any number of intermediate cases could be selected. An example intermediate case has also been provided in Figure 59.

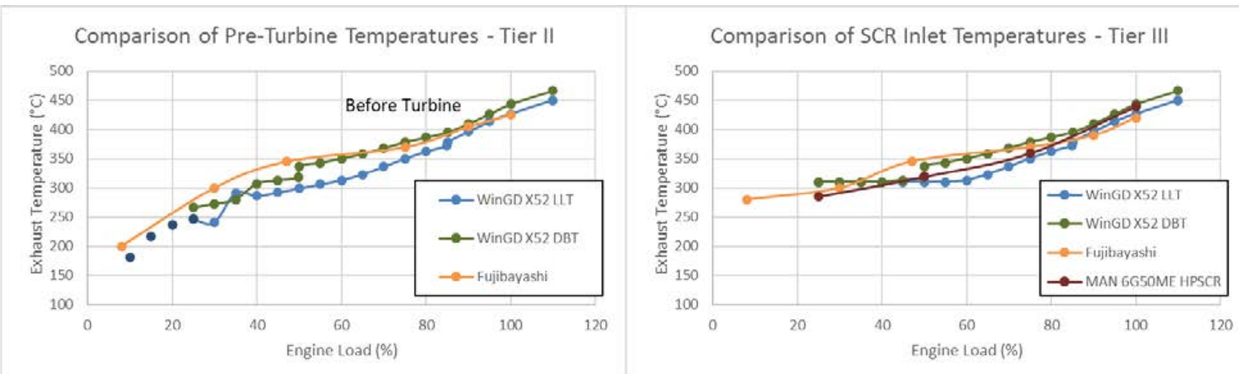


FIGURE 58. EXHAUST TEMPERATURE DATA FOR MARINE TWO-STROKE ENGINE

FIGURE 59. EXHAUST TEMPERATURE VERSUS LOAD PROFILE FOR WINGD X52 SERIES ENGINE

4.2 Brake Specific NO_X Profiles for Marine 2-Stroke Engines

Brake specific NO_X data are provided as an output of the MAN CEAS. For Tier III SCR engines are provided, in Figure 60. The upper and lower ranges from MAN CEAS for the engine out (SCR in) data represent values that would meet the Tier II standard, in the case of the minimum, and meet the Tier I standard, in the case of the maximum. Although these data are nominal values (the same for all engines with HP-SCR), they can be used for calculations looking at the contribution of low loads to NO_X and estimate the benefit if SCR operation were extended to lower loads. For these calculations, the average of the minimum and maximum values was assumed to account for the possibility that the engine calibration might be tuned to produce engine out NO_X higher than Tier II to gain back some energy efficiency when using SCR. Another assumption that was required was the engine out NO_X emissions below 25-percent load. For purposes of calculations, it was assumed to be an extrapolation of the SCR in curve. This is considered a conservative assumption since the brake-specific NO_X may be much higher since operation at these loads is almost always governed by an approved ACD.

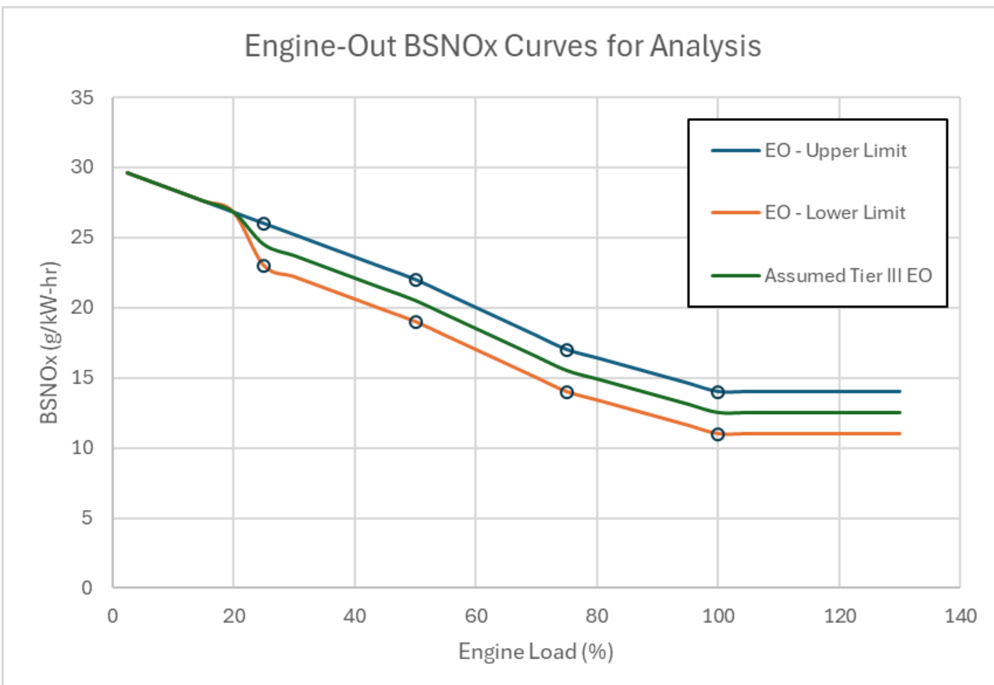


FIGURE 60. BSNO$_X$ VERSUS LOAD FOR A MARINE TWO STROKE ENGINE (MAN ENERGY SOLUTIONS, N.D.)

4.3 Marine NO_X Control Technology Selections for NO_X Estimates

As discussed earlier, there are a variety of approaches the have been taken in the marine industry to meeting the current Tier III regulation. For low-speed, two-stroke diesel engines the primary choice comes down to either EGR or SCR as the primary NO_X reduction lever. As shown earlier in Figure 53, it appears that SCR has been the primary NO_X reduction lever with 65 to 75% of Tier III OGV main engines utilizing this technology, although this portion does vary somewhat by bore size. In addition, it appears that HP-SCR located upstream of the turbine has been the primary choice for dealing with the relatively low exhaust temperatures associated with low-speed two-stroke engines. When examining the potential for further emission reductions, it is useful to make some technology selections as a “best available” approach, although that does not necessarily mean that the selected strategy is the only approach that will work.

For the examination of available emissions reductions discussed later, the decision was made to focus on the use of HP-SCR, along with thermal management options necessary to reach the desired exhaust temperatures. The primary focus of this effort was to examine the possibility of extending emission controls to loads below the current duty cycle floor of 25% maximum power, although the potential some reduction in higher load emissions is also examined in the analysis. For lower load emissions, it was felt that LP-SCR would likely present too much of a thermal management challenge at loads below 20% without very large energy inputs. , EGR would also be able to reach at least some of these targets, but would likely face more difficult technical challenges with condensation risk at very low loads, as well as implementation challenges (and potentially efficiency issues) with higher EGR rates than are currently used at high loads. Note that this does not preclude the combination of EGR and SCR as a strategy, but this would likely be very expensive, complicated, and difficult to package.

In order to reach the necessary temperatures to enable SCR operation, while avoiding ABS condensation risk, current Tier III engines employ a variety of thermal management techniques at lower engine loads. The primary approach to thermal management consists of air-fuel ratio management through the use of either a cylinder bypass valve, a turbine bypass valve, or both. Other combustion changes affecting injection timing and fueling are certainly also part of these strategies, but the change in air-fuel ratio is the primary approach.

For the analysis of potentially available NO_X reductions, SwRI has chosen to develop a set of two different scenarios each for SCR efficiency and exhaust temperature, one aggressive and one more conservative scenario. In general, the more conservative scenarios would represent the potential of the current technology while extending the NO_X controlled envelope down to 10% maximum power while maintaining current performance levels. The more aggressive scenarios involve more aggressive thermal management and would extend the range down to 5% of maximum power, while also pushing SCR conversion to higher levels. The SCR conversion scenarios are discussed in more detail below.

Regarding the engine and thermal management scenarios, the inputs used are shown below in Figure 61 for exhaust temperatures and Figure 62 for engine-out NO_X. Figure 61 shows temperature curves versus load for a Best Projected scenario and for an Intermediate Projected scenario, both in comparison to current Tier III behavior and Tier II temperatures. The current

Tier III behavior was used to establish the baseline tailpipe NO_X levels which were the basis for comparison for calculating emission reductions for the other scenarios. It should be noted that ACD activation does not always occur just below 25%. In practice there is not a sudden switch off of NO_X control just below 25% because some amount of operating margin is needed to ensure NO_X control at 25%. Therefore, the Tier III curve still maintains thermal management and therefore some NO_X conversion down to 20% load. This is important to note because this Tier III curve is the baseline for comparison of other scenarios, and thus sets the "baseline" cumulative NO_X mass for the projections over the histograms.

The Intermediate Projection represents what is likely to be achievable using the current technologies installed on marine Tier III compliant engines, with very little modification. In general, the intent behind the Intermediate Projection was to provide thermal management to enable successful use of SCR down to the 10% load point. It should be noted that there are several examples in the literature of temperatures at or greater than these levels on current technology engines. As mentioned earlier, Hitachi-Zosen showed SCR inlet temperatures at 10% load in the range of 280°C (Fujibayashi, T., et. al., 2013) in 2013, while WinGD described available engine thermal management via a "Tier III mode" available down to 10% load at an exhaust temperature of 310°C (Spahni & et.al., 2023). These data sets indicate the feasibility of reaching these temperatures on real hardware with the current Tier III technology packages, questioning the need for approval of ACDs for all operation below 25% load.

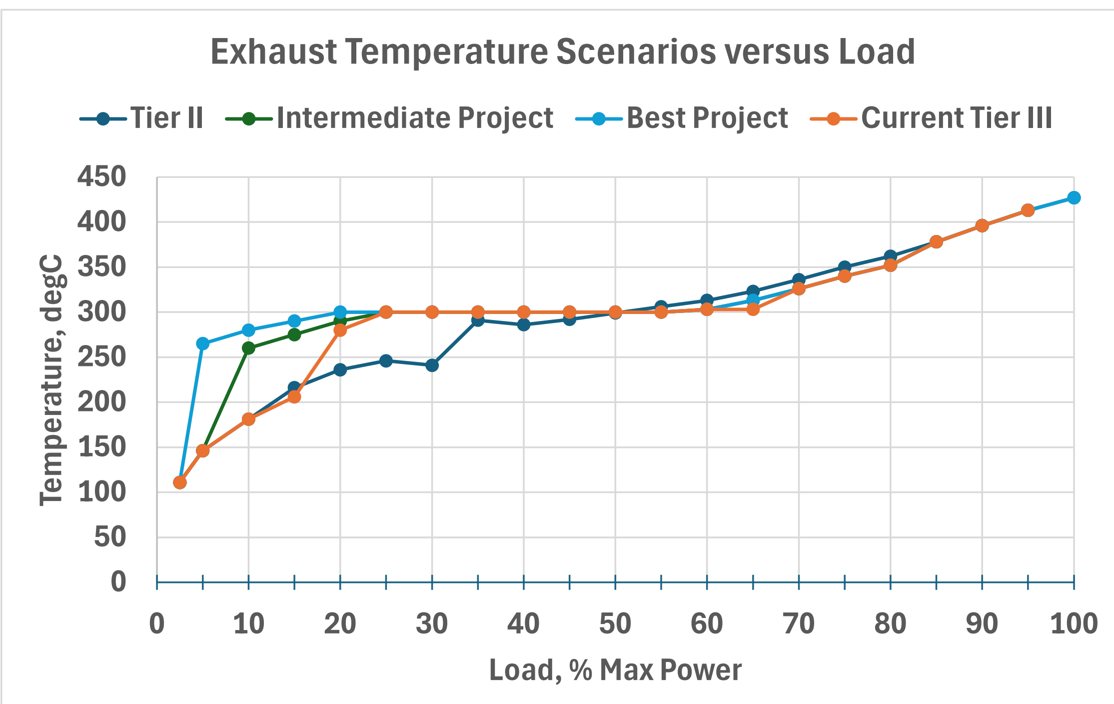


FIGURE 61. SCR REACTOR INLET TEMPERATURES FOR TECHNOLOGY PROJECTIONS COMPARED TO TIER III CURRENT

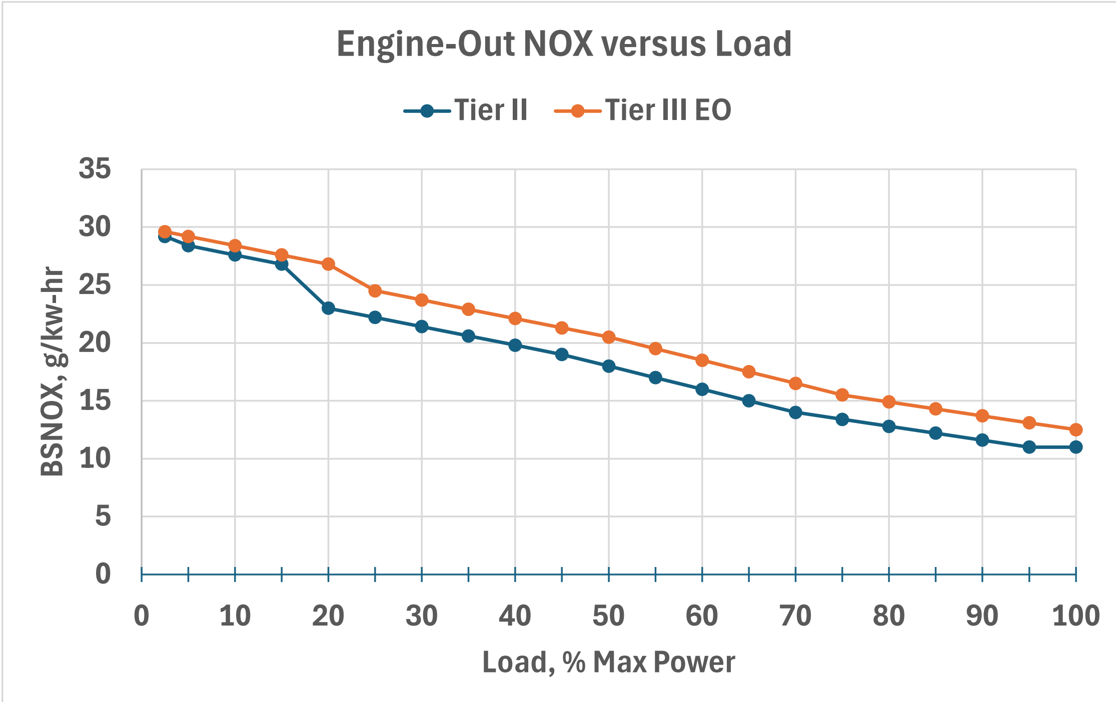


FIGURE 62. ENGINE-OUT NO_X CURVE FOR TECHNOLOGY PROJECTIONS

The exhaust temperature Best Case Projection is designed to push thermal management upwards somewhat, allowing for higher conversion at low load points, but also to extend the range of SCR operation down to 5% of Maximum Power. It is expected that much of this thermal management can be accomplished using the previously discussed technologies, but in the particular case of 5% load, some additional technology may be needed to provide the margin necessary to reach the target temperature, though this heat addition is anticipated to be in the range of an additional 20°C more than what is available from the current air fuel management and other engine calibration parameters. From a technology standpoint, options for this would include combustion phasing, early exhaust valve opening, a relatively small electrical heater or burner, or potentially the process steam-based intake air heating technology described earlier.

4.4 Ammonium Bisulfate Deposit Formation Assessment

One of the primary factors limiting the use of SCR in lower temperature conditions for marine SCR is risk of ammonium bisulfate formation (ABS), either in the exhaust upstream of the SCR reactor, or within the reactor itself. Although small amounts of ABS that do form may be removed under higher temperature conditions, especially in cases with a low ammonia-to-NO_X ratio (ANR), in general prolonged operation under ABS forming conditions will lead to fouling of the SCR reactor, deposit growth on upstream piping systems, and corrosion. Given the high mass rates of engine-out NO_X and therefore NH_3 injection observed in operations covered by the current Tier III requirements, as well as elevated exhaust pressures, the approach to controlling this issue is to maintain exhaust temperatures well above the ABS dewpoint, so as to prevent formation in the first place. Under conditions typical of 2-stroke engines at 25% load and higher, this is generally considered to be around 300°C. Accounting for some heat loss between the exhaust receiver and the SCR this generally implies engine-out temperatures of 310°C, considering the use of exhaust insulation and other design elements of current systems. As discussed earlier, this generally requires the use of various engine thermal management techniques at loads anywhere from 30% to 50%, depending on engine and aftertreatment design. The thermal management requirements become significant around 20 to 25% load and below.

When considering the possibility of achieving NO_X control using SCR below 25% load, the ABS limitation becomes a critical potential limiting factor. Given the limits discussed above, large amounts of thermal management could be required to reach 310°C at very low loads. However, because exhaust conditions are quite different in these low load regions, it is important to examine the behavior of ABS to look more closely at how much thermal management is really needed. Although ABS formation has been studied extensively for two stroke marine diesel engines, much of the literature has focused on higher sulfur fuels, well above the current ECA limit of 0.10% sulfur. In addition, given the propensity of engine manufacturers to pursue ACDs to disengage the SCR system at low exhaust temperatures, the operating range below 25% load has generally not been considered in most of these studies.

The formation of ABS in 2-stroke marine diesel exhaust is a complicated process whose rate is governed by a combination of exhaust temperature, exhaust pressure, and the partial pressures of the primary reactants which are SO_3 and NH_3. Given that the desire is to avoid ABS formation, the critical point to understand is the condensation temperature of ABS, which in turn dictates the "safe" temperature, above which SCR may be used. Muzio et al. summarized a variety of studies of the kinetics of this process as indicated by the chart in Figure 63 (Muzio & et.al., 2017). The authors examined a variety of studies, as well as conducted their own experimental work. They concluded that the combined curve described by the earlier work of Menasha (Menasha & et.al., 2011) and Wei (Wei, 2007) was the best representation of this environment, and therefore this curve (shown on the chart using the light blue and pink data points) is used for the current discussion. However, this curve is relevant primarily near atmospheric pressure.

At pressures above atmospheric pressure, ABS formation rates can increase significantly. This is due to increased partial pressure of the reactants at elevated exhaust pressures. This was described by Sandelin et.al. (Sandelin & et.al., 2016) at the 2016 CIMAC Congress, and an example of this relationship is shown in Figure 64. This example was for 165 g/kW-hr fuel consumption, 8 g/kW-hr NO_X, ammonia at 1000ppm, and assumes 5-10% oxidation of SO_2 to SO_3. The figure shows that elevated exhaust pressures can easily increase the condensation temperature by 20-30°C at even moderate exhaust pressures. While this is a crucial consideration at higher loads, recall that exhaust pressures below 20% load are generally in the range of 1.1 to

1.2 bar, even with scavenging blowers active, and therefore will not contribute significantly to the condensation temperature under low load conditions, especially given the need to limit air fuel ratio to effect thermal management.

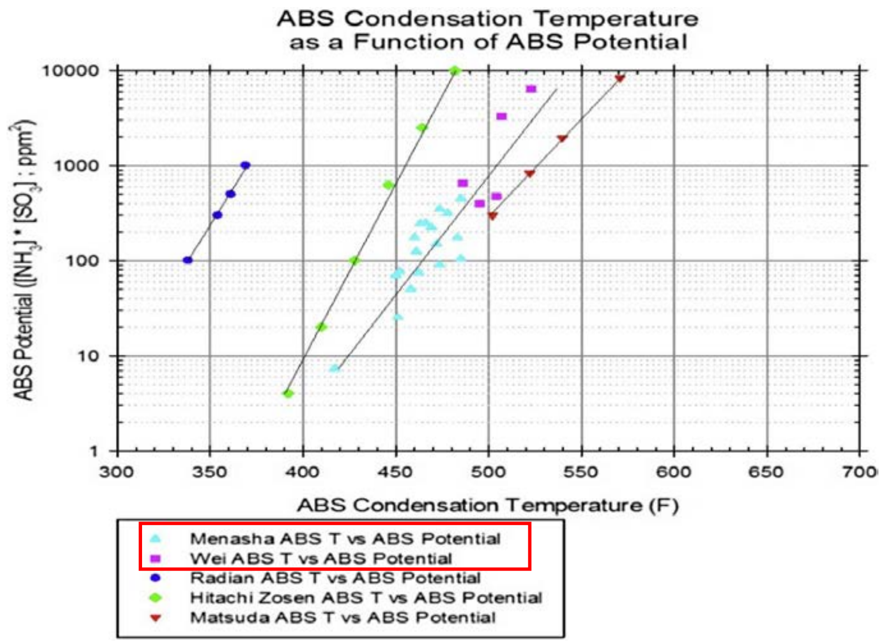


FIGURE 63. ABS CONDENSATION AS A FUNCTION OF REACTANT CONCENTRATION (MUZIO ET. AL)

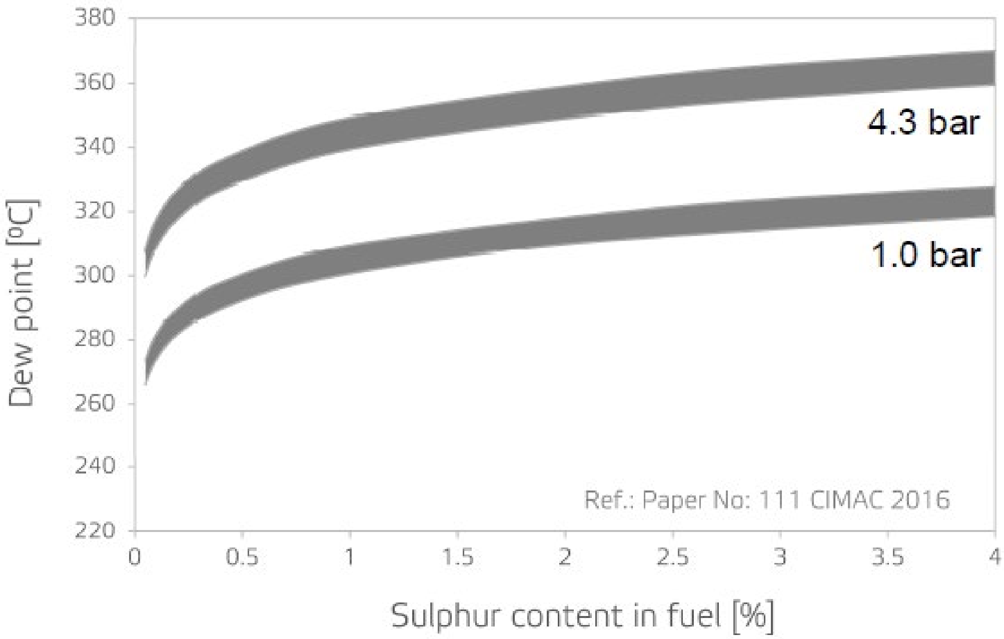


FIGURE 64. INFLUENCE OF PRESSURE ON ABS CONDENSATION TEMPERATURE

The oxidation of exhaust SO_2 to SO_3, is an important reaction step in the formation of ABS, because the reactions are driven by the availability of SO_3 and other oxidized SO_2 byproducts (at lower temperatures it is more likely that H_2SO_4 is the reactant given the reaction with water in the exhaust). This has been studied extensively in the literature, and SO_3 ranges of 3% to 10% of total SO_2 have been reported, though the majority of the reported levels have been 5% and lower. An assumption of 5% to 10% was used by Sandelin to generate temperature ranges above in Figure 64, and is documented in other literature. For purposes of examining ABS risk in this study, an SO_2 oxidation rate of 7% was utilized to provide a conservative estimate.

As noted earlier, low load operation below 25% of maximum engine power tends to be characterized by higher air fuel ratios which will reduce the partial pressure of SO_3, an important driver of the ABS formation temperature. In addition, given the lower catalyst temperatures, full NO_X conversion rates in the range of 80 to 90 percent are not expected, and thus it is likely the ammonia dosing rate will be lowered somewhat below an ammonia-to-NO_X ratio of 1.

Finally, we can take into consideration that fuel sulfur levels within the ECA will be less than 0.10%. According to marine fuel survey data, the median sulfur concentration for fuels under this specification is 600 ppm (0.06%) (IMO-MEPC, 2024). ABS formation calculations were run at both 600 ppm and 1000 ppm fuel sulfur. Some calculations were also run at 7 ppm fuel sulfur to examine the potential impact of the use of ULSD on ABS condensation temperatures. An ammonia to NO_X ratio of 0.8 was used for these conditions, which is also factored into later SCR conversion curves.

Given these parameters, and the operating parameters typical of low load operation below 25% of maximum power, it is possible to estimate the risk temperatures for lower load operation in the range of 10% and 5% of maximum power, as compared to 25%. Assuming that thermal management is accomplished using typical currently-applied methods of air-fuel ratio management, such as cylinder bypass and/or turbine bypass methods, it is expected that air-fuel ratio would have to be controlled in the range of 50-60 to achieve SCR temperatures in the range of 260°C to 300°C. For performance projections, we are expecting NO_X values under these light load conditions in the range of 25-28 g/kW-hr. Considering these inputs and using the ABS condensation rate kinetics in Muzio (Muzio & et.al., 2017), estimates are given for the “low load” ABS condensation temperatures in the range of 5-10% of maximum power.

TABLE 7. ABS CONDENSATON TEMPERATURE ESTIMATES AT 5-10% LOAD

Fuel Sulfur, ppmw	SO_2 Concentration, ppm	NH_3 Concentration, ppm	ABS Condensation Temperature, °C
600 (ECA median)	8 – 9	~ 1200	255 – 260
1000 (ECA max limit)	14 – 15	~ 1200	260 - 265
7 (ULSD median)	~ 0.1	~ 1200	~ 225

These projections indicate that under very light load conditions (i.e., below 20%), and only under light load conditions, it appears feasible to extend the range of ABS-limited SCR operation to temperatures lower than the 300°C limit that is generally used at higher loads. Again, it must be noted that this is not feasible at higher loads (i.e., 30% and higher) due to the combination of higher pressures, lower air-fuel ratios, and higher ammonia dosing rates.

4.5 Selective Catalytic Reduction Performance Curves

As noted earlier, for the NO_X projections, high-pressure SCR is the technology of focus. For the Tier III marine diesel engine market, the primary technology of choice is vanadium-based catalysts. Given the size and scale, as well as the need to be resistant to relatively high levels of sulfur compared to many other applications, vanadium is the generally considered to be the best overall choice for use in Tier III marine diesel engines. The SCR reactor arrangements often use technologies similar in many ways to those used for stationary power generation, where SCR was first applied. More specifically, the catalyst is a V_2O_5 material on a WO_3/TiO_2 support. These applications typically use an extruded catalyst rather than a washcoated substrate which is typical of smaller applications. In addition, given the application a relatively low cell density substrate is used, often on the order of 64 cpsi. This allows the catalyst assembly to have low backpressure, while being less susceptible to fouling from soot, ash, and ABS exposure. The reactors are typically assembled using multiple layers of individual catalyst blocks that are assembled to create the required reactor cross sectional area for the application. These assemblies are usually designed so the blocks can be removed and replaced if there is a need to service the reactor. Although design guidance does vary, the reactors are typically sized for space-velocities (a global measure of catalyst size relative to flow rate in catalyst volumes per hour) on the order of 10,000 1/hr to 15,000 1/hr. This sizing is changing somewhat as reactor designs continue to improve, but this will be used for the current projections.

Although NH_3 can be used directly as the reductant, it is more typical that urea-water solution (UWS) similar to DEF and/or AdBlue that is used in land based applications will be used for ease of storage and safe handling. The concentration of urea in the water in marine applications, however, is typically 40% urea by mass, which is higher than the 32.5% used for land based applications. The SCR system includes dosing and metering systems for the UWS, and arrangements for mixing are necessary in the exhaust upstream of the reactor inlet to ensure the uniformity of the reductant distribution. Although problems with deposit formation can be issues in other applications, for an HP-SCR marine application dosing is generally not done below the ABS formation temperature, which is typically much higher than the temperatures where UWS related deposit formation can be an issue (typically at temperatures below 210°C). Nevertheless, proper mixing design is important to prevent any such issues.

The active temperature range of these catalyst systems for peak conversion is generally in the range from 300°C to about 425°C. It should be noted that these catalysts are generally capable of NO_X conversion efficiency in excess of 90% at these temperatures under the right conditions, but in the case of Tier III certified marine diesel engines, they are generally controlled to a nominal peak NO_X conversion on the order of 80% or slightly higher. Parasitic ammonia oxidation on the surface of the catalysts begins to compromise performance starting around 425°C to 450°C, although initially this can be mitigated somewhat by increased dosing at temperatures below 500°C. At temperatures below 300°C, NO_X conversion capability typically begins to drop, although this does not become pronounced until the temperature drops below 250°C. The driving force, however, that limits low temperature conversion in Tier III marine diesel engine applications is the need to avoid ABS condensation, as described earlier.

Three conversion curves were generated in support of the NO_X mass emission projections in Section 5. These NO_X conversion curves are expressed as NO_X reduction efficiency as a function of catalyst temperature (in this case SCR reactor inlet temperature). The SCR conversion curves used for these projections are given in Figure 65.

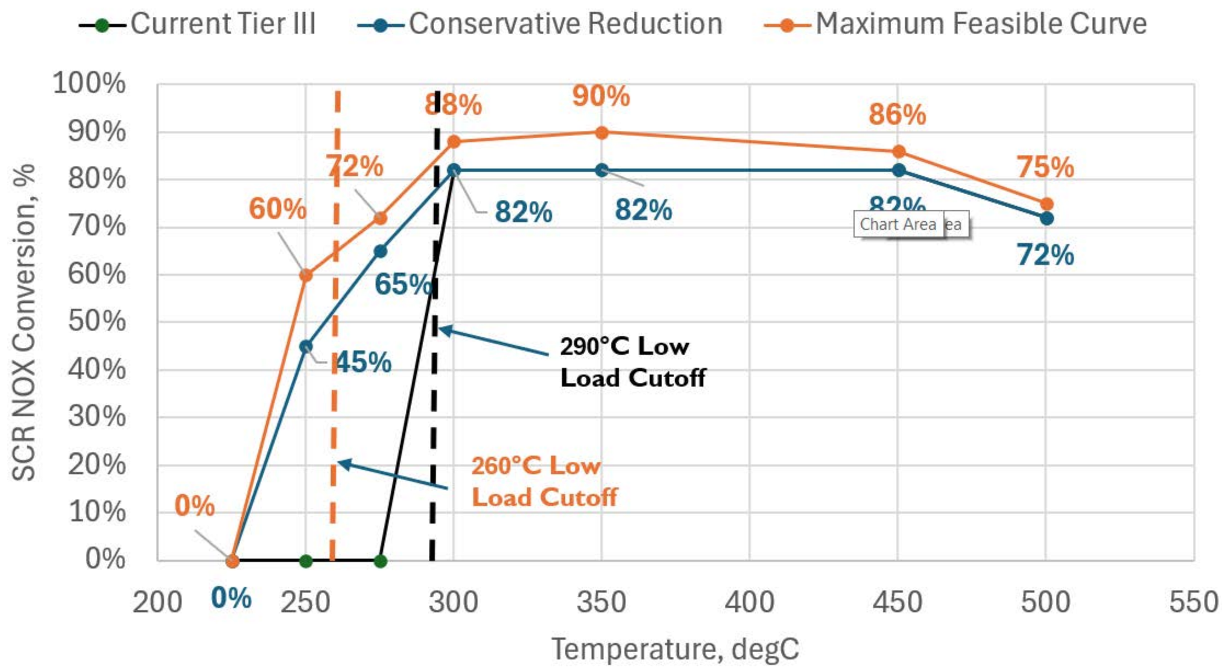


FIGURE 65. SCR NO_X CONVERSION CURVES USED FOR TIER III MARINE DIESEL ENGINE NO_X PROJECTIONS

The SCR conversion curve labeled as Current Tier III (black line) is intended to reflect a current system calibrated to meet the Tier III requirements for a NO_X ECA assuming the engine-out NO_X levels and temperatures curves shown previously. As noted earlier, peak conversion on these catalysts is typically controlled to about 80%, which is sufficient to satisfy the Tier III requirements, although it is common for the actual performance to be slightly better than 80%. For the case of this baseline Tier III curve, a peak conversion of 82% was used to reach a tailpipe or stack outlet NO_X level of the 2.9 g/kW-hr for the 4-mode E3 cycle, resulting in a reasonable compliance margin below the standard of 3.4 g/kW-hr. In addition, the individual modes are also below the individual mode cap of 1.5 times the standard. The SCR does not operate below 25% load as engine manufacturers routinely request an ACD with a relatively conservative minimum temperature to avoid ABS formation of 290°C at the reactor inlet is used as the cutoff, below which no UWS will be dosed and thus no conversion will be achieved. At higher loads, the increased exhaust pressure means that a higher minimum temperature of 310°C is used to prevent ABS condensation, although this is typically only a concern during transient operation before the catalyst is fully warmed up. In practice the need for some operation margins means that some conversion is typically achieved down to 20% load, and sometimes at temperatures as low as 275°C for a short time.

The SCR conversion curve labeled Conservative Reduction (blue line) represents a system that has been calibrated to continue dosing at temperatures below 290°C but only under light load conditions. A minimum temperature of 260°C is utilized, representing the minimum temperature that can be allowed to avoid ABS, specifically under very light load conditions. Note that for the purposes of ABS control at higher loads, the minimum temperature of 310°C is retained, but again, this is only significant during transients. It is important to note that the target conversion performance is significantly reduced below the capability of the catalysts at higher loads, in part to assist with the avoidance of ABS formation at these temperatures, resulting in UWS dosing rates well below an ammonia-to-NO_X ratio of 1. This curve represents an intermediate approach that retains the high load conversion targets of previous systems, and instead focuses only on extending the low temperature conversion range, but in a conservative fashion.

The SCR conversion curve labeled Maximum Feasible Curve represents what we believe to be a more aggressive, yet still achievable performance level given the current technology. It focuses on both a further improvement step in low temperature performance, and an increase in high load conversion to better reflect the actual capabilities of the catalyst under reasonable UWS dosing conditions. For high load, performance can be increased by increasing the UWS dosing rate to reach an ANR closer to 1. However, this is still limited by the tendency for the generation of increasing amounts of ammonia slip at higher space velocity conditions. An example of this relationship, taken from Sandelin (Sandelin & et.al., 2016), is given below in Figure 66 for a condition of 350°C at 1.3 atmospheres. For the Maximum Conversion curve, the NH_3 slip limit was set at 10ppm. This effectively limits peak conversion at the upper end to a range of 85 to 90% depending on the temperature.

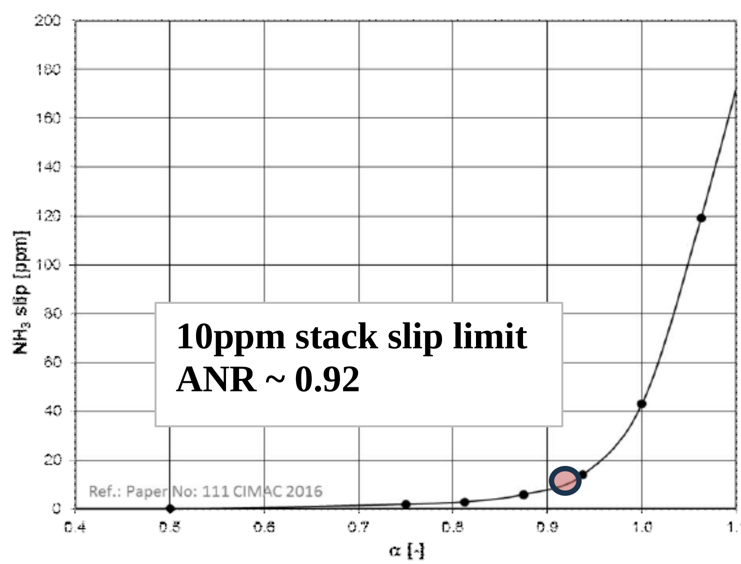


FIGURE 66. AMMONIA SLIP CURVE EXAMPLE VERSUS STOICHIOMETRIC AMMONIA-TO-NOX RATIO

For the Maximum Conversion curve, the minimum ANR was increased somewhat to bring NO_X conversion closer to the capability of the catalysts at these temperatures, while still representing a reduced rate to help manage ABS formation. The minimum dosing temperature of 260°C at the reactor inlet is retained for light load operation, but given the higher ANR this is only feasible at the lowest loads when operating only on ECA fuel with less than 0.10% sulfur content. Even at 10% load, a higher minimum temperature would likely need to be used, scaling upwards with drop in load. Careful coordination of thermal management and dosing rates would be needed to realize reasonable ABS avoidance under these conditions. It is possible that there will be some ABS formation in catalyst pores at the lightest loads at and around 5% load depending on conditions. However, overall the very low NO_X mass rates would likely result in very low actual ABS condensation rates that should be manageable as long as there is some operation in the ECA at higher loads (above 25%).

Other scenarios are possible as well, such as the combination of an SCR conversion curve that increases high load conversion like the Maximum curve, while retaining the more conservative low load behavior of the Conservative curve. However, these were not examined as part of this analysis.

5.0 ASSESSMENT OF POTENTIAL FOR ADDITIONAL NO_X REDUCTIONS IN EMISSION CONTROL AREAS

The ultimate objective of this study is to estimate the NO_X reduction potential of applying various technology modifications to Tier III compliant marine diesel engine designs that are equipped with SCR. Using information contained in the histograms from Section 2.6 showing the load duty cycle data from vessels in the fleet as derived from AIS position data, the technology selections discussed in Section 4.3, and the projected SCR performance curves in Section 4.5, it is possible to estimate potential NO_X reductions from Category 3 engines on a mass basis for various technology scenarios. These projections then can be used to examine different changes to the standards, and estimate the NO_X reduction potential available from those scenarios. These tools can be used to examine the impact of changes using any engine load histogram as an input, and the technology assumptions can be adjusted to examine other technology scenarios.

This analysis is focused on the engine load histogram for the fleet of ships with Tier III main propulsion engines that operate in the ECAs, as presented in Section 2. While scenarios were also generated by individual vessel type, the following discussion focuses on the ships included in the complete Tier III ECA data set.

The engine load histogram that is used as the input for this analysis is repeated below in Figure 67. This figure shows the total frequency counts in each of the designated load bins, along with the cumulative distribution for each bin as a percentage of the total operation data set. It should be noted that the data indicates that 43% of all operation in the ECA is below the 25% load point, indicating that a significant portion of operation is below the emission control window of the current Tier III certification requirements. Even assuming that emission controls are still operational down to 20% load as discussed earlier, this still results in 35% of all operation not being considered when assessing compliance with the Tier III NO_X limits.

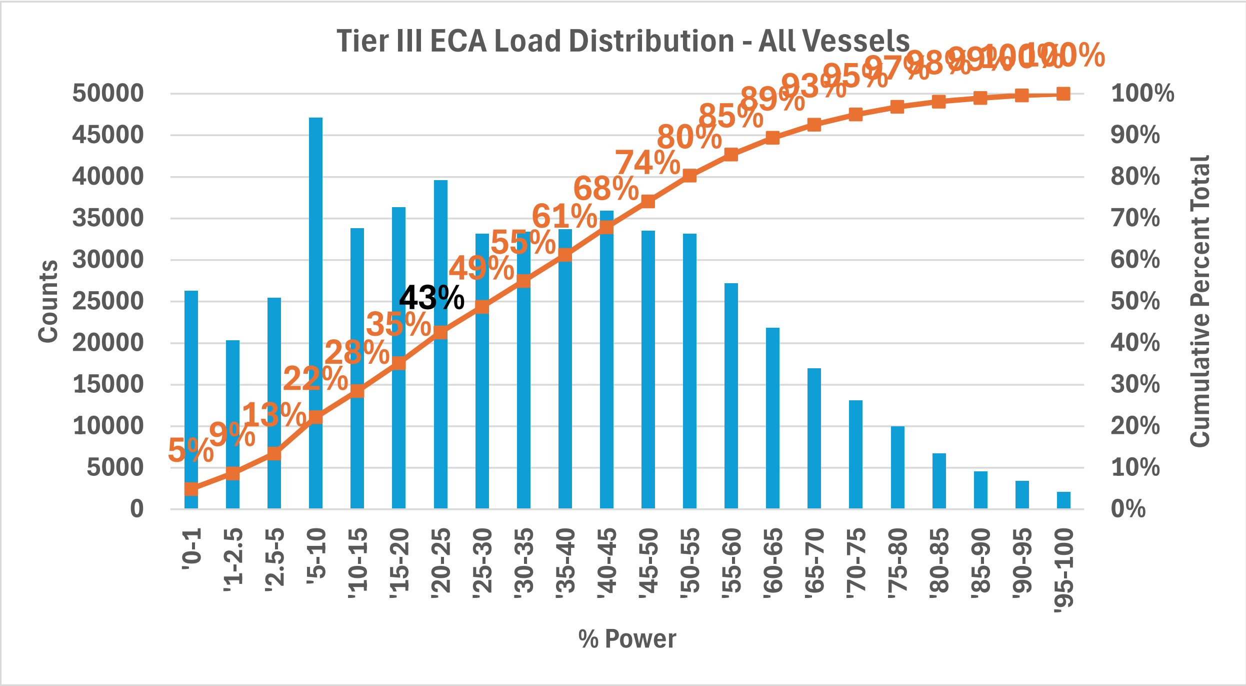


FIGURE 67. TIER III ECA LOAD DISTRIBUTION – ALL VESSELS

A comparison of the actual load histogram for the North American ECA to the weight factors that are currently assigned in the IMO E3 test cycle is shown below in Table 8. Note that the operational weighting factors are calculated as the sum of operation between a given mode and the next lower mode, with the 25% mode encompassing operations from 0-25%. From the table, it is clear that the IMO weighting factors are far from accurately representing actual operation in the ECA.

TABLE 8. ACTUAL OPERATIONAL DISTRIBUTION IN ECA VERSUS E2/E3 TEST CYCLE WEIGHTING

IMO Mode Number	% Power	IMO E2/E3 Weight Factor	Actual Operational Weighting in ECA
1	25	0.15	0.43
2	50	0.15	0.32
3	75	0.50	0.21
4	100	0.20	0.05

The basic methodology we used to estimate NO_X emissions was to combine the temperature data and SCR conversion curve along with Tier III engine-out NO_X data to generate a tailpipe brake-specific NO_X level for each load bin in the histogram. That brake-specific NO_X level is then multiplied by the duty cycle total counts in each load bin and by the average percentage of maximum power for that bin, producing a load weighted NO_X "mass rate," in units of mass counts per kilowatt of engine power. Although this could be translated to an actual NO_X mass rate by considering the time associated with each count and the individual vessel power for each data point, this additional transformation was not necessary because the object of this study was not to estimate a NO_X inventory. Rather, the objective of the study was to provide a relative assessment of NO_X reduction potential for certain technologies when applied to 2-stroke Category 3 engines.

To allow for relative comparisons, a baseline scenario for NO_X performance was needed. Using the ECA duty cycle histogram in Figure 67 and the Tier III NO_X baseline performance curves established for the technology scenarios, a baseline NO_X mass scenario representing the current Tier III fleet was generated. This baseline scenario is shown in Figure 68. This data shows the relative amount of NO_X mass emitted during all operations recorded in the ECA for each load bin. Table 9 also shows this baseline in terms of the exhaust NO_X results over the current E3 cycle. The composite value of 2.7 g/kW-hr indicates compliance with the 3.4 g/kW-hr IMO Tier III standard with an expected 20% compliance margin.

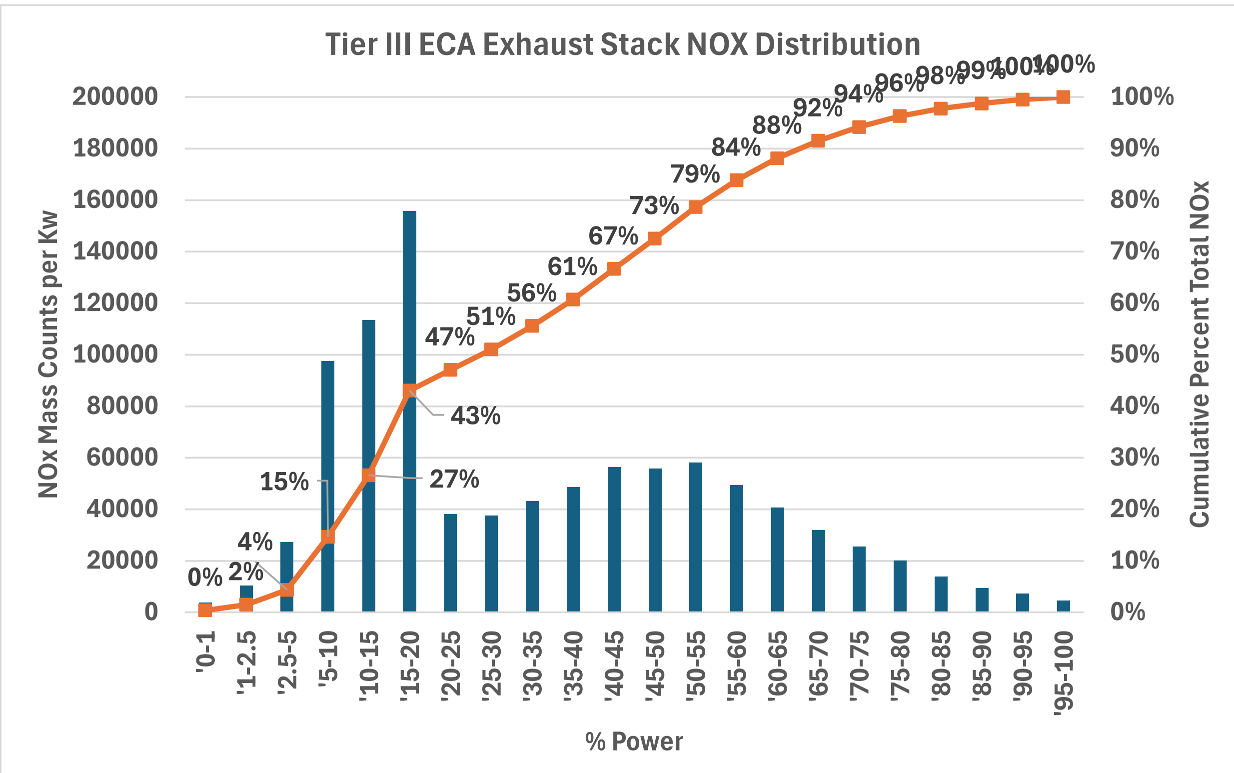


FIGURE 68. TIER III BASELINE NO_X DISTRIBUTION FOR THE ECA

TABLE 9. IMO E3 TEST CYCLE RESULT FOR BASELINE TIER III SCENARIO

IMO Mode Number	Percent Load	Exhaust NO_X, g/kW-hr	E3 Weighting Factor
1	25%	4.2	0.15
2	50%	3.3	0.15
3	75%	2.6	0.50
4	100%	2.3	0.20
Composite	**n/a**	**2.7**	n/a

The data indicates that 43% of all NO_X mass emitted in the ECA occurs in the load bins below 20% maximum power. In particular, the load bins between 5%-10%, 10%-15%, and 15%-20% contribute the majority of that NO_X. Although the relative power at these lower bins is low, the high brake-specific NO_X combined with the large amount of operation in those bins results in a significant contribution to the total NO_X mass emissions. The contribution to NO_X mass below 5% maximum power is very small and can be ignored. There is also still a considerable amount of the total NO_X mass emitted in the higher bins, indicating potential for further reduction by applying the technologies discussed above.

Figure 69 shows the contribution of operation in the various load bins to overall total NO_X mass, in comparison to the amount of time spent in each load bin as a percentage of total operation. These results indicate that the load bins between 5% and 20% have a NO_X mass emission contribution that exceeds their relative amount of operation time, while bins below 5% are not significant. In addition, bins in the 35% to 65% range still contribute significant NO_X mass, although those values are more proportional to their relative amount of operation time.

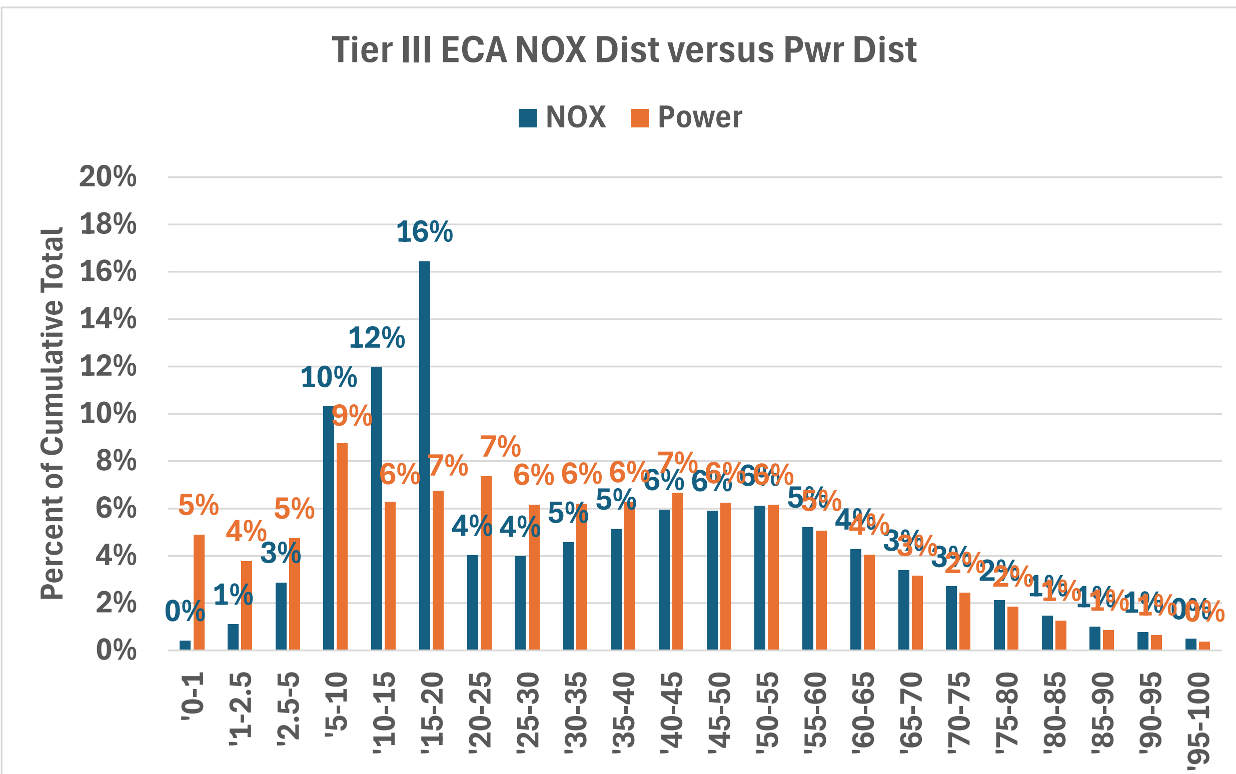


FIGURE 69. COMPARISON OF POWER DISTRIBUTION VERSUS NO_X DISTRIBUTION FOR TIER III VESSELS IN THE ECA

One potential countermeasure to address the disproportionately high NO_X emissions at low load is by re-weighting the current test cycle modes for engines certifying to IMO Tier III NO_X standards, to more accurately reflect the actual duty cycle of engine operation in ECAs. While this would not address the low load emissions, it is still useful to examine what could be gained from this relatively simple change. Using the current Tier III NO_X baseline performance shown in Figure 68, a revised set of weighting factors would produce the results indicate in Table 10 below. Using revised weighting factors, the composite NO_X level rises from 2.9 g/kW-hr to 3.2 g/kW-hr, which would still pass the current standard, but with less compliance margin. To achieve a sufficient compliance margin, SCR efficiency would have to be increased on some of the test cycle modes. The current peak efficiency in these modes under the current Tier III scenario is 82%. Increasing this to 85% efficiency, apart from at 100% load which could be problematic on some engines, would again achieve 2.7 g/kW-hr NO_X, which would meet the NO_X standard, assuming a 20% compliance margin.

TABLE 10. NO_X RESULT FOR TIER III BASELINE USING ACTUAL TIER III ECA DUTY CYCLE WEIGHTING, ASSUMING 82% PEAK EFFICIENCY

IMO Mode Number	Percent Load	Tailpipe NO_X, g/kW-hr	Revised Weighting Factor	E3 Weighting Factor
1	25%	4.2	0.43	0.15
2	50%	3.3	0.32	0.15
3	75%	2.6	0.21	0.50
4	100%	2.3	0.05	0.2
Composite	**n/a**	**3.2**	n/a	n/a

The impact of this change on the NO_X distribution is show in Figure 70 below. As can be seen, this adjustment does not address the low load NO_X mass emissions, and ultimately results in only a modest 9% reduction in total NO_X mass emitted. It is clear that other changes would be needed to achieve a more meaningful reduction in total NO_X mass.

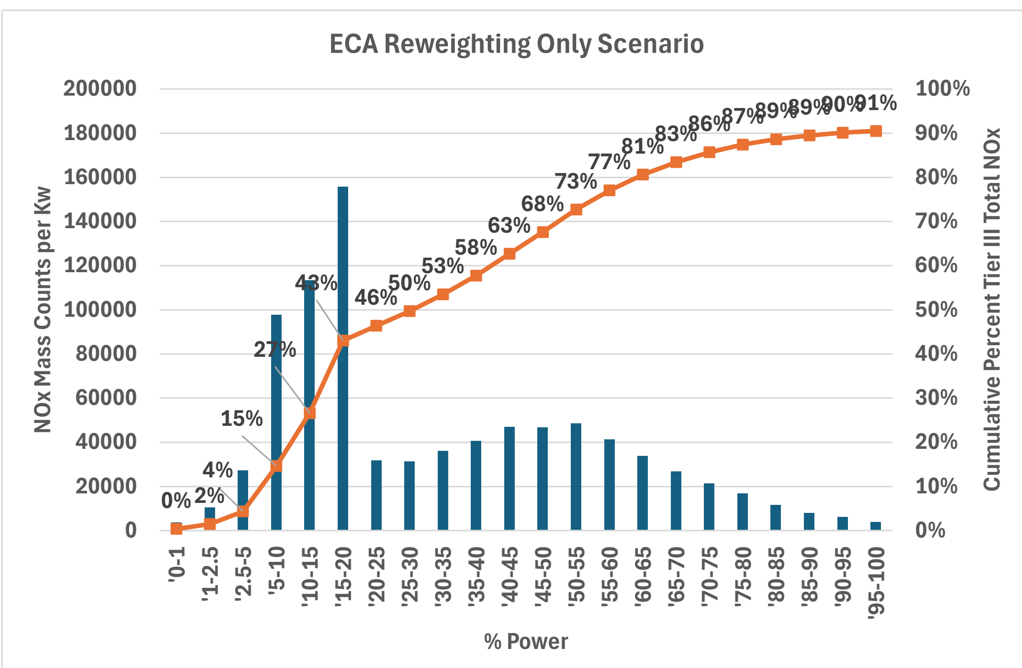


FIGURE 70. NO_X MASS DISTRIBUTION FOR ECA MODE REWEIGHTING SCENARIO AND CURRENT STANDARD

To examine if more significant emission improvements are available, we examined the impact of the various technology scenarios on the overall NO_X histogram. It is also possible to examine different NO_X standards that those scenarios could support. Many of these scenarios involve extending NO_X control to the load range below 25%, to address the considerable amount of NO_X mass emitted in the load bins between 5% and 20%. Recall, the assumption that the current NO_X standard at 25% still results in at least some emission control in the 20-25% range due to the need for operating margin. In addition, some of the scenarios described below also examine improvements to SCR performance at loads above 25%.

Figure 71 shows the NO_X mass distribution for what is considered a "Best Case" scenario. This scenario combined both the Best Case temperature curve, which results in NO_X control down to 5% load, with the Maximum Feasible SCR performance curve. The results indicate a total reduction of 53% in NO_X mass emitted in the ECA if these technology measures were implemented. $BSNO_X$ levels and NO_X conversion by load bin for this scenario are given in Figure 72. This scenario involves leveraging existing thermal management technologies available on marine diesel engines, pushes high load SCR conversion to what is considered to be the best performance feasible for the current vanadium SCR catalysts, and requires an additional thermal management technology to extend emission control down into the 5-10% load bin.

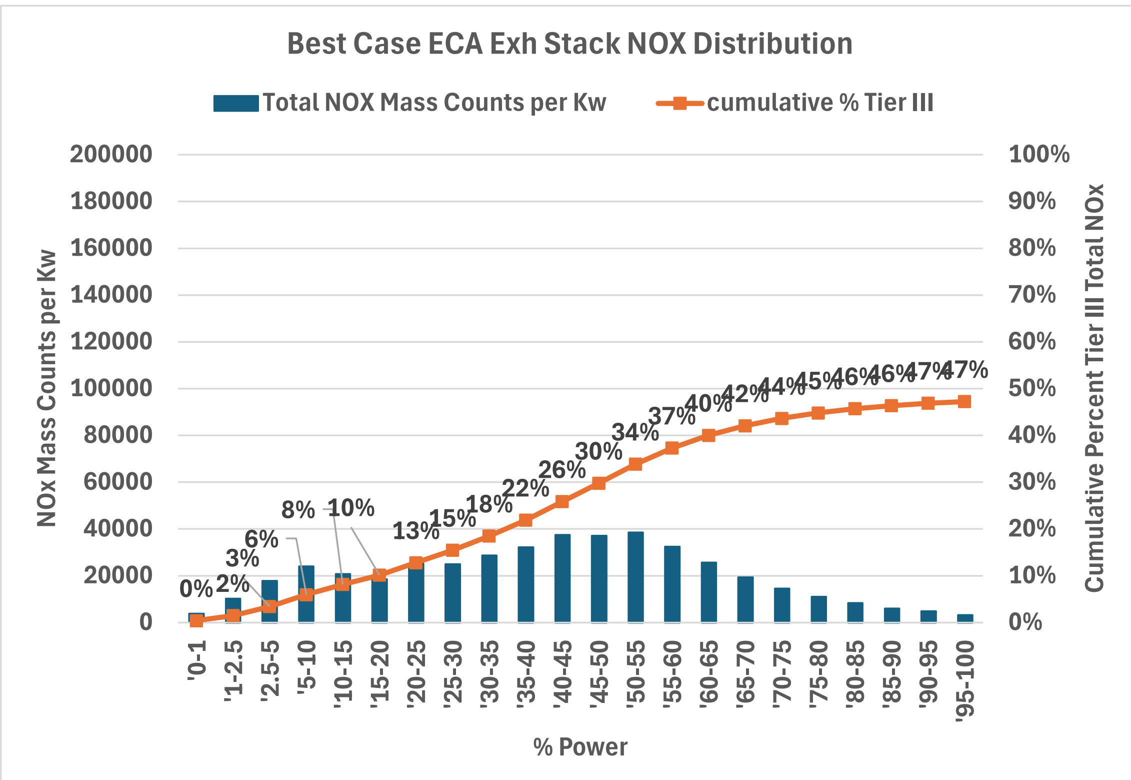


FIGURE 71. NO_X MASS DISTRIBUTION FOR BEST CASE TECHNOLOGY SCENARIO

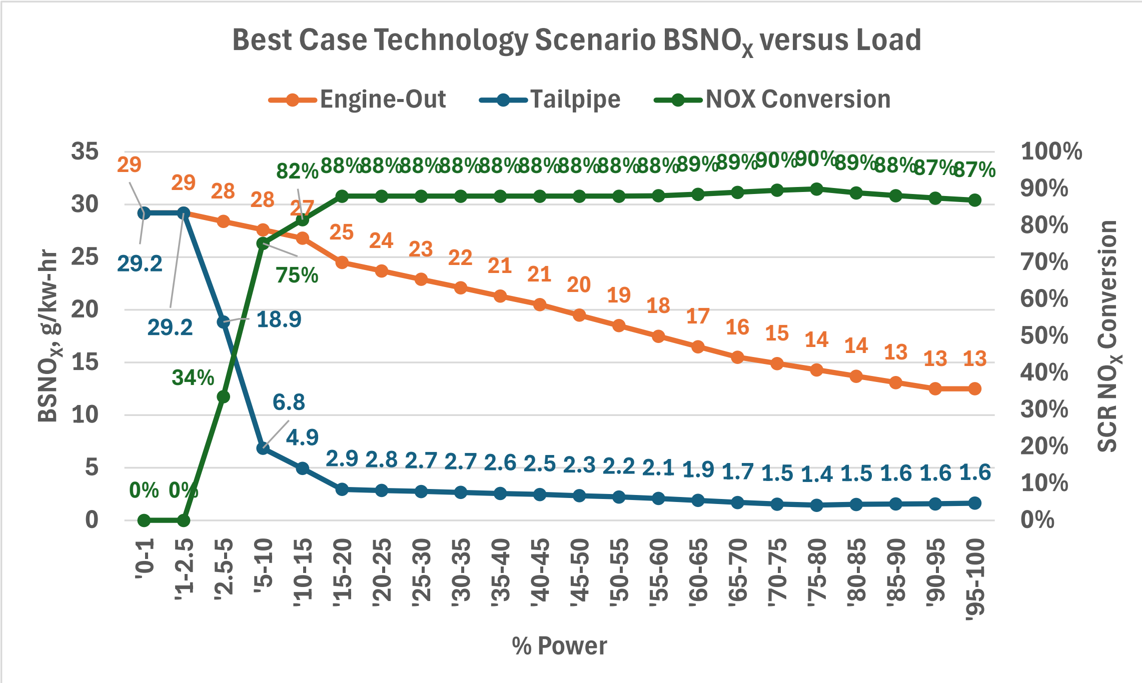


FIGURE 72. BSNO$_X$ AND SCR NO$_X$ CONVERSION BY LOAD BIN FOR BEST CASE TECHNOLOGY SCENARIO

A comparison of NO$_X$ histograms between the Tier III Baseline scenario and the Best Case technology scenario is shown in Figure 73. Although NO$_X$ reductions are evident in most bins, and there is a small spillover in the 2.5-5% bin due to operating margin, the majority of NO$_X$ mass reduction, on the order of 60% of the total reduction, occurs in the load bins between 5% and 20% of maximum power. In the higher load regions, a reduction on the order of 20% is observed, although this is somewhat lower at the highest loads due to limits associated with ammonia oxidation. The contribution at high load is still responsible, however, for one-third of the total NO$_X$ mass reduction, and therefore improving NO$_X$ reduction at high loads is still a significant change to consider. Recall that this higher load change involves pushing high load conversion in the range from 300°C to 425°C from 82% up to 88%.

Table 11 shows the cycle weighted result for the E3 test cycle modes using both the IMO weighting factors and the ECA duty cycle weighting factors. The table provides potential composite NO$_X$ standards that this best case scenario could support, assuming a 20% NO$_X$ compliance margin to account for production variance. These test cycle weighting factor adjustments would capture the benefits of the Best Case scenario in the higher load region. Additional changes would still be needed to realize the large gains achievable at low loads. This can be done by extending the E3 test cycle to include lower load mode points and mode caps below 25%, which would prevent the use of ACDs in the low load operating range.

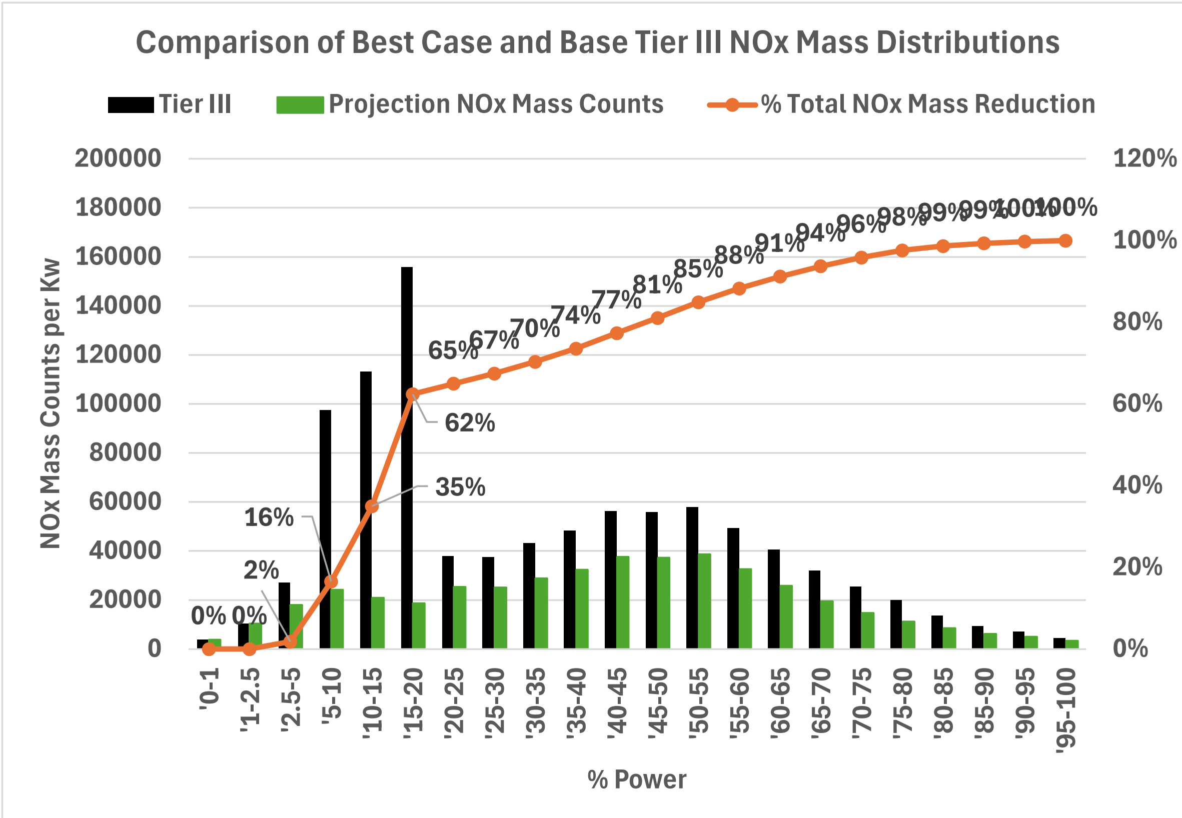


FIGURE 73. COMPARISON OF NO_X MASS DISTRIBUTIONS FOR BEST CASE TECHNOLOGY SCENARIO AND TIER III BASELINE SCENARIO

TABLE 11. CYCLE WEIGHTED EMISSION RESULTS FOR CURRENT MODES FOR BEST CASE TECHNOLOGY SCENARIO

IMO Mode Number	Percent Load	Tailpipe NO_X, g/kW-hr	IMO E3 Weighting Factor	Updated Tier III Weighting Factor
1	25%	2.7	0.15	0.43
2	50%	2.2	0.15	0.32
3	75%	1.4	0.50	0.21
4	100%	1.6	0.20	0.05
Composite, g/kW-hr			**1.7**	**2.1**
Potential standard with 20% Margin			**2.0**	**2.5**

Including lower load emission control could involve the use of cycle mode changes, low load mode emission caps, or possibly both. Table 12 shows one such scenario involving the addition of a 10% mode to the duty cycle, combined with the further addition of a mode cap at 5% load, to capture the 5-10% load bin. Based on the data in the scenario, this mode cap would need to be set at 2.5 times the standard for that mode only. The mode cap at 10% would also need to be set at 1.8, which is somewhat higher than the 1.5 currently allowed for all other modes. With the addition of a new mode, this scenario includes new weighting factors. These were assigned on the basis of the ECA duty cycle weighting for the 5 modes used in the composite calculation. Under this scenario, the test cycle and compliance with the standard for operation outside the ECA could remain at the current Tier II level utilizing the IMO 4-mode E3 test cycle, since the objective is to capture some of the emissions during Tier III operation that are occurring due to lack of SCR control at low load.

TABLE 12. CYCLE RESULTS ON BEST CASE TECHNOLOGY SCENARIO FOR UPDATED TEST CYCLE WITH 10% MODE AND UPDATED DUTY CYCLE WEIGHTING FOR TIER III

Mode Number	Percent Load	Tailpipe NO_X, g/kW-hr	Updated Tier III Weighting Factor	Mode Cap
n/a	5%	6.8	n/a	3.2
1	10%	4.9	0.22	2.3
2	25%	2.7	0.20	1.5
3	50%	2.2	0.32	1.5
4	75%	1.4	0.21	1.5
5	100%	1.6	0.05	1.5
Composite, g/kW-hr			**2.2**	
Potential standard with 20% Margin			**2.6**	

As noted earlier, the extension of the NO_X curve to 5% load will likely involve the addition of thermal management technology that is not currently included on most Tier III compliant marine diesel engines. Therefore, it is reasonable to examine a less aggressive scenario in which the NO_X curve is extended down to 10% load rather than 5%. This was examined using the Intermediate Temperature curve and the Maximum Feasible NO_X curve. The NO_X histogram for that scenario is shown in Figure 74. Figure 75 shows $BSNO_X$ levels and NO_X conversion by load bin for this scenario. This scenario results in a 45% reduction in NO_X mass, with the primary changes being the loss of NO_X reduction on the 2.5-5% and 5-10% load bins (note some NO_X reduction still occurs at 5-10% load due to the need for operating margin near 10% load). This scenario involves applying less thermal management at both 5% and 10% loads, resulting in lower temperatures and somewhat less NO_X performance at the 10% mode point, as well as only a small amount of conversion below that. NO_X reductions for this scenario are shown in Figure 76.

This scenario still includes increasing NO_X conversion at temperatures in the range between 300°C to 425°C from 82% to 88%. In addition, it still requires increased thermal management utilization at low loads compared to the current Tier III, but to a lesser extent and only actively down to a little below the 10% load point for operating margin. As noted earlier, there are several examples in the literature of current hardware on vessels that can achieve the temperatures required to support conversion down to 10% load without additional hardware.

Table 13 shows potential changes to the standards that would realize these reductions. These changes would extend the range of emission control down to the 10% load point. This scenario is similar to the previous scenario shown in Table 12 but without the inclusion of the 5% load point that provides NO_X emission control via a mode cap. Instead, the cycle and caps are only down to 10%. Functionally, the only cycle change is that performance on the 10% load point is not quite as good, and as a result, the cycle composite NO_X result is higher. This assumption would result in a slightly higher NO_X limit, and also requires a mode cap at 3 times the emission standard for the 10% load point, while leaving the others at 1.5.

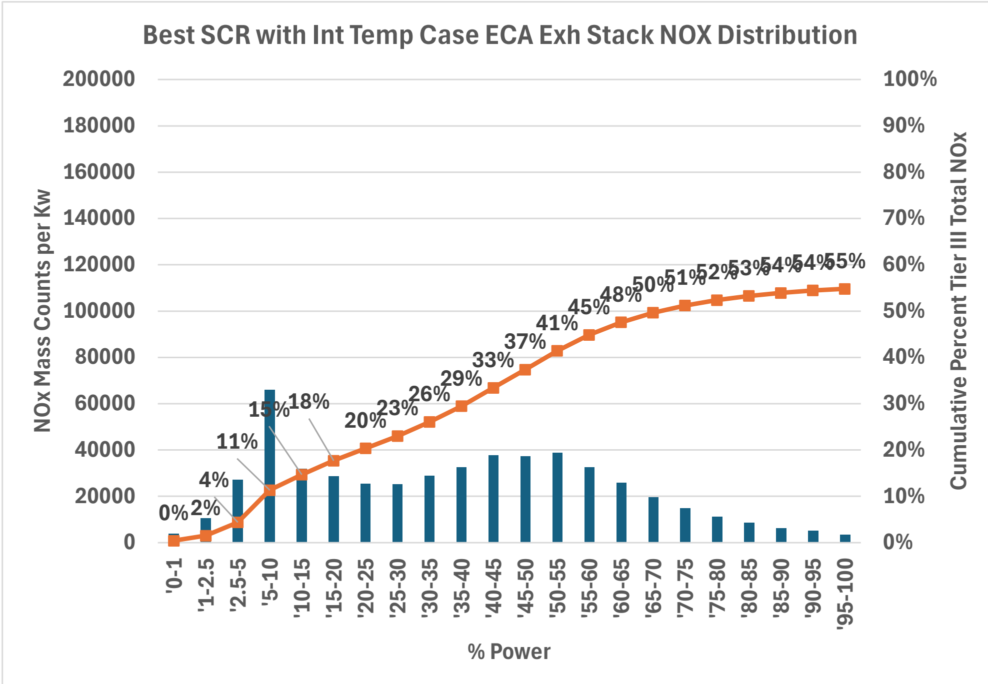


FIGURE 74. NO_X MASS DISTRIBUTION FOR MAXIMUM FEASIBLE NO_X CURVE, INTERMEDIATE TEMPERATURE CURVE, TARGETING CONVERSION DOWN TO 10% LOAD SCENARIO

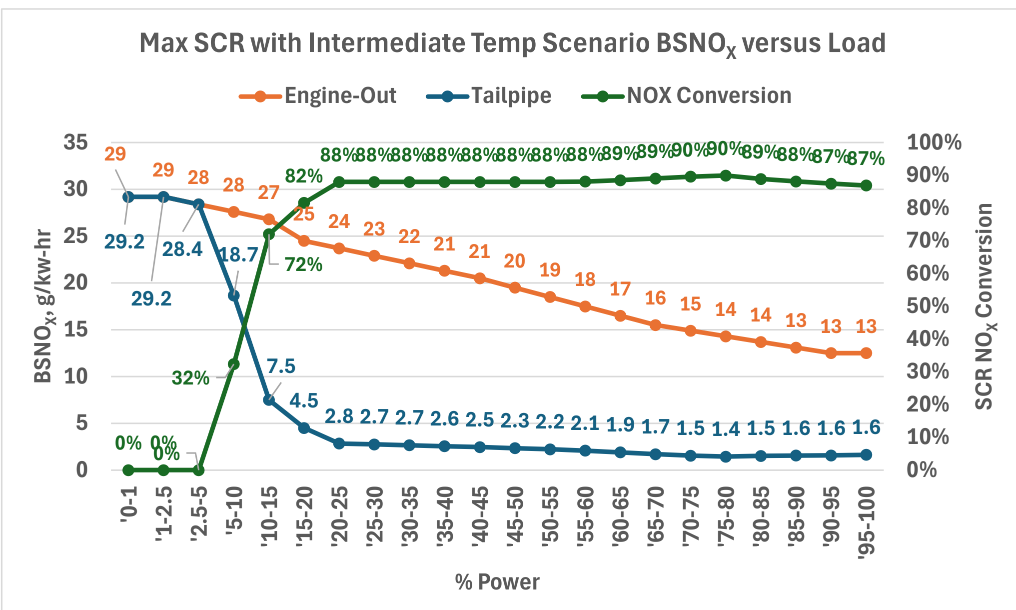


FIGURE 75. $BSNO_X$ AND SCR NO_X CONVERSION BY LOAD BIN FOR MAXIMUM FEASIBLE NO_X CURVE, INTERMEDIATE TEMPERATURE CURVE, TARGETING CONVERSION DOWN TO 10% LOAD SCENARIO

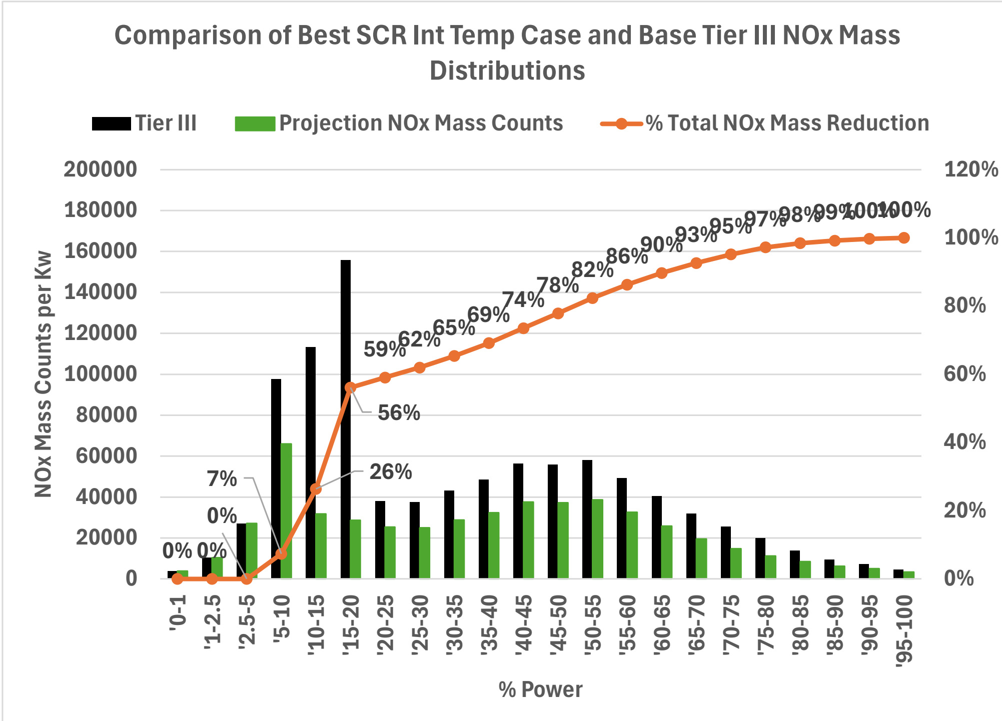


FIGURE 76. COMPARISON OF NO_X MASS DISTRIBUTIONS FOR MAXIMUM FEASIBLE NO_X CURVE, INTERMEDIATE TEMPERATURE CURVE, TARGETING CONVERSION DOWN TO 10% LOAD SCENARIO AND TIER III BASELINE

TABLE 13. CYCLE RESULTS ON MAXIMUM FEASIBLE SCR CURVE, INTERMEDIATE TEMPERATURE CURVE, TARGETING CONVERSION DOWN TO 10% LOAD FOR TEST CYCLE WITH 10% MODE AND UPDATED DUTY CYCLE WEIGHTING FOR TIER III

Mode Number	Percent Load	Tailpipe NO_X, g/kW-hr	Updated Tier III Weighting Factor	Mode Cap
1	10%	7.5	0.22	3
2	25%	2.7	0.20	1.5
3	50%	2.2	0.32	1.5
4	75%	1.4	0.21	1.5
5	100%	1.6	0.05	1.5
Composite, g/kW-hr			**2.5**	
Proposed standard with 20% Margin			**3.0**	

Another possible scenario is to focus only on improvements in the low temperature region, while leaving the current performance levels intact at higher loads. We examined this case using the Intermediate Temperature Curve and the Conservative SCR Conversion Curve. This would limit the low temperature improvements at the 10% load point like the previous scenario, but in addition would not push the high load conversion. The resulting NO_X distribution associated with this more conservative scenario is shown in Figure 77 below, and the $BSNO_X$ and NO_X conversion are given in Figure 78. A comparison with the Tier III baseline is shown in Figure 79.

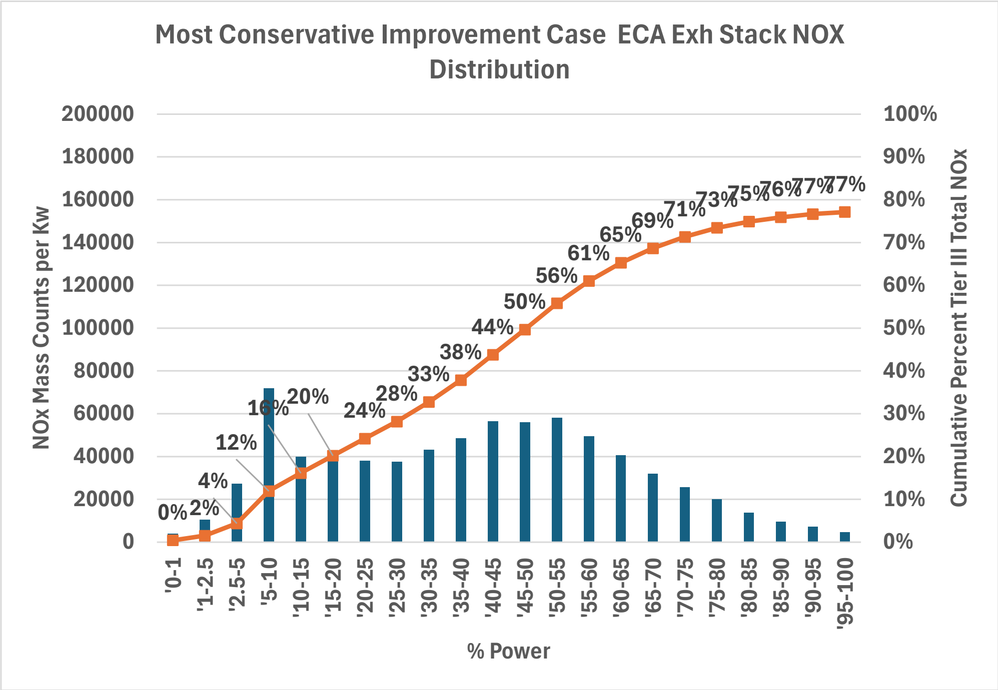


FIGURE 77. NO_X MASS DISTRIBUTION FOR MORE CONSERVATIVE (LOW-TEMP ONLY) NO_X CURVE, INTERMEDIATE TEMPERATURE CURVE, TARGETTING CONVERSION DOWN TO 10% LOAD SCENARIO

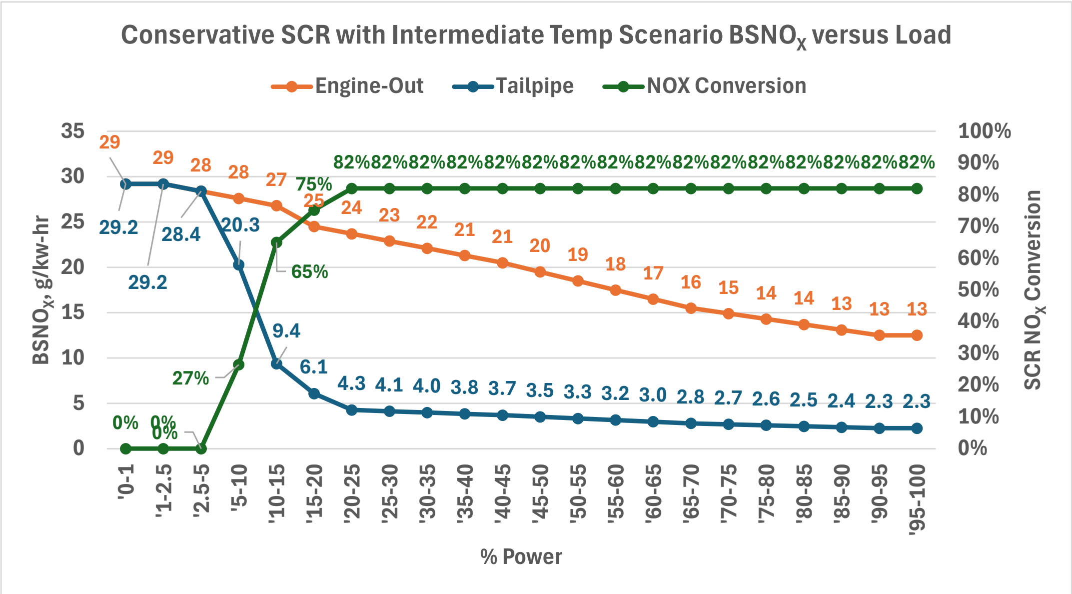


FIGURE 78. $BSNO_X$ AND SCR NO_X CONVERSION BY LOAD BIN FOR MAXIMUM CONSERVATIVE NO_X CURVE, INTERMEDIATE TEMPERATURE CURVE, TARGETING CONVERSION DOWN TO 10% LOAD SCENARIO

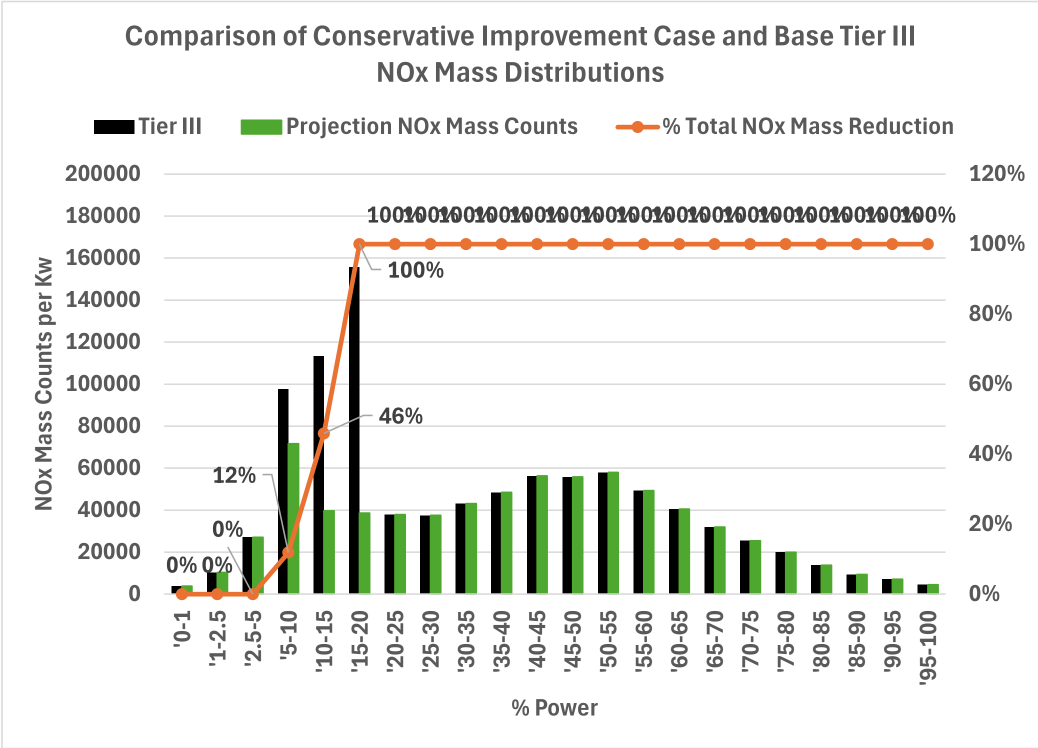


FIGURE 79. COMPARISON OF NO_X MASS DISTRIBUTIONS FOR CONSERVATIVE (LOW-TEMP ONLY) NO_X CURVE, INTERMEDIATE TEMPERATURE CURVE, TARGETING CONVERSION DOWN TO 10% LOAD SCENARIO AND TIER III BASELINE

This scenario results in an overall 23% reduction in NO_X mass, 88% of which occurs in the 10-20% load range, and a small amount below 10% due to operating margin. This is about half of the NO_X mass reduction observed for the previous scenario that had a similar extension to cover low load operation but also included higher load reductions.

Table 14 shows a set of results for a 5-mode regulatory scenario with ECA duty cycle weighting for this scenario. The higher resulting standard does somewhat reduce the multiplier needed for the 10% mode cap to 2.6, but the NO_X limit would be 50 percent higher than the previous scenario.

TABLE 14. CYCLE RESULTS ON CONSERVATIVE (LOW-TEMP ONLY) SCENARIO TARGETING CONVERSION DOWN TO 10% LOAD FOR TEST CYCLE WITH 10% MODE AND UPDATED DUTY CYCLE WEIGHTING FOR TIER III

Mode Number	Percent Load	Tailpipe NO_X, g/kW-hr	Updated Tier III Weighting Factor	Mode Cap
1	10%	9.4	0.22	2.6
2	25%	4.1	0.20	1.5
3	50%	3.4	0.32	1.5
4	75%	2.6	0.21	1.5
5	100%	2.3	0.05	1.5
Composite, g/kW-hr			**3.7**	
Proposed Standard with 20% Margin			**4.4**	

Figure 80 shows a scenario which involves maximizing low temperature reductions while leaving higher temperature performance at 82% conversion where it currently stands. This scenario involves combining the more conservative SCR reduction curve with the best case temperature curve, pushing NO_X conversion down to 5% of maximum power. $BSNO_X$ and NO_X conversion for this scenario are given in Figure 81. As can be seen, pushing the range of NO_X conversion down to 5% from the previous scenario (at 10%) without any change in high load performance only results in a small added improvement of an additional 7% reduction in NO_X compared to the Tier III baseline scenario, to a total of 30% reduction for this scenario. Figure 82 shows that all of the improvements occur only at low loads as expected, with the additional gains coming from improvements in the bin from 5-20% maximum power.

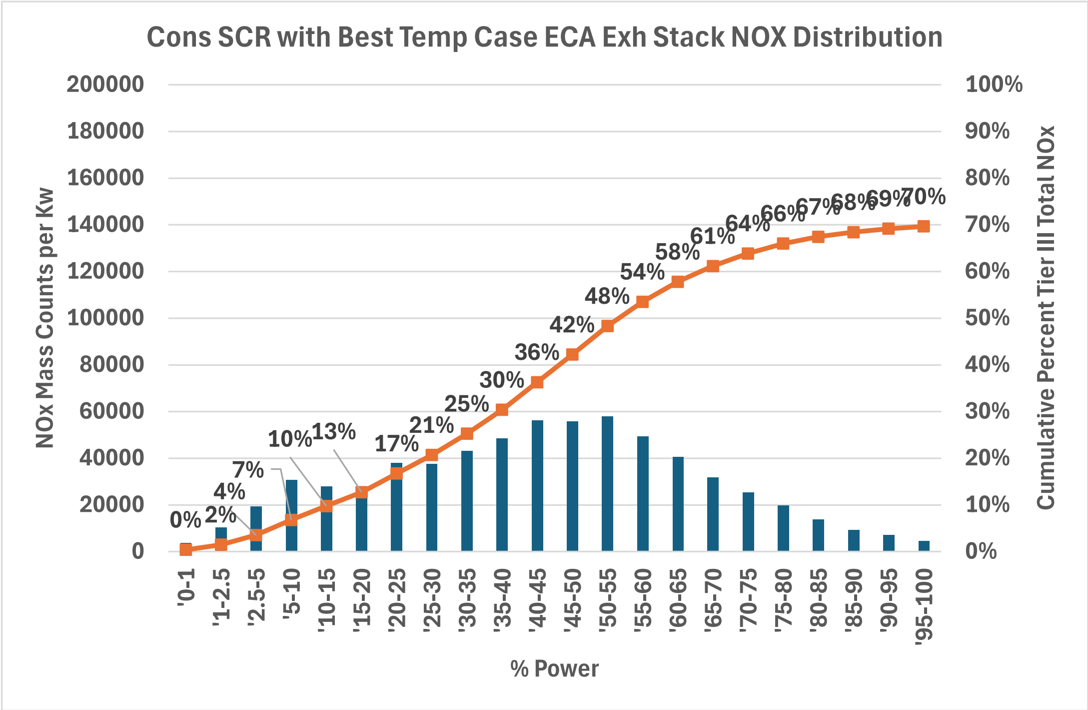


FIGURE 80. NOx MASS DISTRIBUTION FOR CONSERVATIVE SCR WITH BEST CASE TEMPERATURE DISTRIBUTION, TARGETING CONVERSION DOWN TO 5% LOAD CONVERSION ONLY

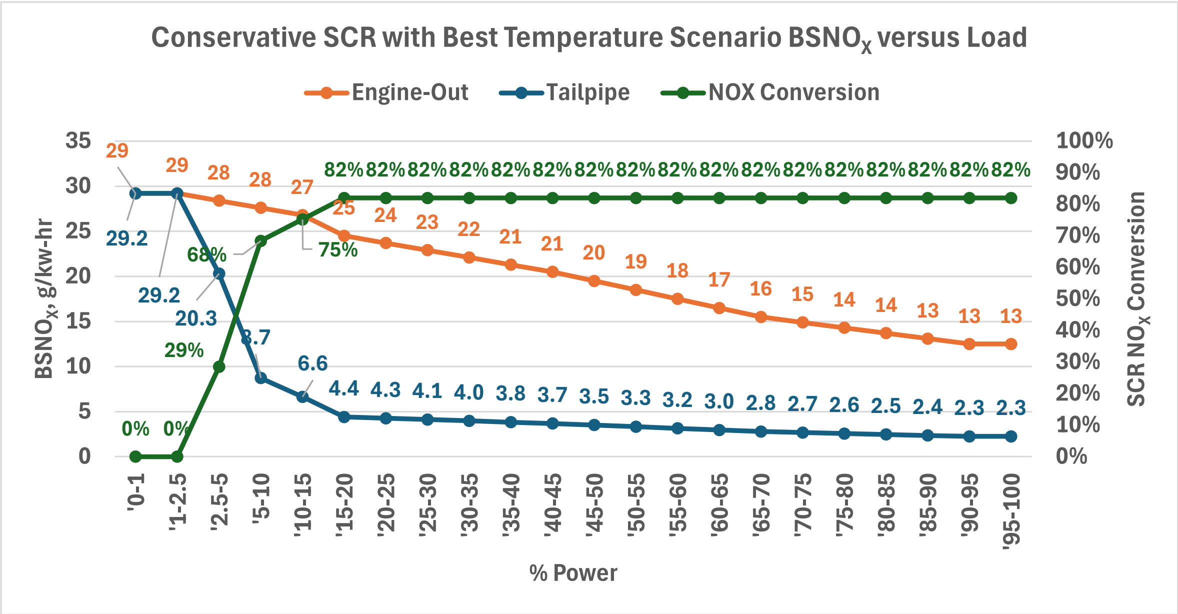


FIGURE 81. BSNOx AND SCR NOx CONVERSION BY LOAD BIN FOR CONSERVATIVE SCR WITH BEST CASE TEMPERATURE CURVE SCENARIO, TARGETING CONVERSION DOWN TO 5%

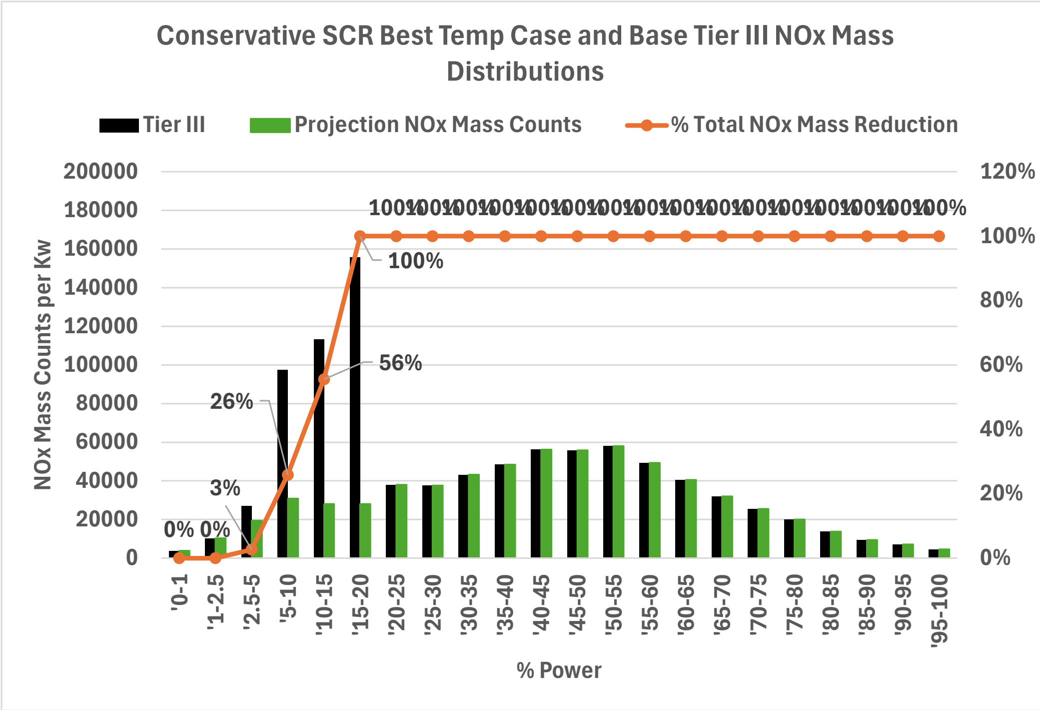


FIGURE 82. COMPARISON OF NO_X MASS DISTRIBUTIONS FOR CONSERVATIVE SCR WITH BEST CASE TEMPERATURE CURVE SCENARIO, TARGETING CONVERSION DOWN TO 5%

Table 15 shows changes to the standard to implement this scenario. A similar set of structural changes to the Best Case projection is used for this scenario, with a 10% mode point added to the test cycle, and an additional mode cap added for emissions at the 5% load point. However, without any high load reductions, calculated composite emissions for the updated cycle would be somewhat higher than the current Tier III standard, due to the re-weighting of the cycle. How, this still results in a net 30% reduction in NO_X mass emitted.

TABLE 15. CYCLE RESULTS ON CONSERVATIVE SCR WITH BEST CASE TEMPERATURES SCENARIO FOR UPDATED TEST CYCLE WITH 10% MODE AND UPDATED DUTY CYCLE WEIGHTING FOR TIER III

Mode Number		Percent Load	Tailpipe NO_X, g/kW-hr	Updated Tier III Weighting Factor	Mode Cap
n/a		5%	8.7	n/a	2.6
1		10%	6.6	0.22	2.0
2		25%	4.1	0.20	1.5
3		50%	3.3	0.32	1.5
4		75%	2.6	0.21	1.5
5		100%	2.3	0.05	1.5
	Composite, g/kW-hr			**3.3**	
	Potential Standard with 20% Margin			**4.0**	

6.0 SUMMARY AND CONCLUSIONS OF NO_X IMPROVEMENT TECHNOLOGY SCENARIO ANALYSIS

A summary of the various technology scenarios is given in Table 16 below. The scenarios are presented in the order they were examined. The table the total reduction in NO_X mass projected for each scenario, along with details of each scenario. These include the minimum load point for active NO_X reduction by SCR. As noted earlier, each scenario projects some NO_X reduction in the next lowest power bin as a result of the need for operating margin, so the scenarios that include 10% minimum load assume some NO_X conversion in the 5-10% load bin, and scenarios that include 5% minimum load assume some NO_X conversion in the 2.5-5% load bin. Any scenarios that include the 10% mode to the duty cycle will require reweighting to account for this new mode, and these weight factors are based on the weighting for the Tier III ECA power histogram.

TABLE 16. SUMMARY OF NO_X IMPROVEMENT TECHNOLOGY SCENARIOS EXAMINED

Scenario	Details	Minimum Load	NOx Reduction	Standard Notes
1	Current Mode Reweights	25%	9%	Reweigh to ECA Duty Cycle
2	Max SCR, Best Temperatures	5%	53%	Add 10% Mode and 5% Mode Cap, Tighten Standard Limits
3	Max SCR, Intermediate Temperatures	10%	45%	Add 10% Mode, Tighten Standard Limits
4	Conservative SCR, Intermediate Temperatures	10%	23%	Add 10% Mode
5	Conservative SCR, Best Temperatures	5%	30%	Add 10% Mode and 5% Mode Cap

Scenario 1 is included for comparisons, but generally indicates the simple measure of re-weighting the test cycle modes to more accurately reflect ECA operation will achieve only modest NO_X reductions without additional changes. All of the remaining scenarios involve extending the range of NO_X control for the test cycle down to at least the 10% load point, which would likely require adding a new test cycle mode at 10% (it may be possible to achieve the same result using carefully designed mode caps). As noted earlier, existing literature indicates that this is likely achievable using technologies already deployed on many Tier III OGVs. It should be noted the scenarios 2 and 5, which extend NO_X control down to the 5% load point will likely require some additional thermal management technology, such as intake air heating or variable valve actuation, to achieve. Technology scenarios marked as "Max SCR", include reductions in the NO_X standard that would require improving high load SCR NO_X conversion efficiency to 88% from the current 82%.

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