ISTANBUL TECHNICAL UNIVERSITY ★ GRADUATE SCHOOL

ENERGY EFFICIENCY ORIENTED MODEL BASED INVESTIGATION OF MARINE DIESEL ENGINE AND AUXILIARY SYSTEMS

Ph.D. THESIS

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Department of Maritime Transportation Engineering

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<u>ISTANBUL TEKNİK ÜNİVERSİTESİ ★ LİSANSÜSTÜ EĞİTİM ENSTİTÜSÜ</u>

ENERJİ VERİMLİLİĞİNE YÖNELİK GEMİ DİZEL MAKİNELERİ VE YARDIMCI SİSTEMLERİNDE MODELLEME TABANLI ARAŞTIRMA

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To my family in the present and future,

FOREWORD

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April 2021 Çağlar DERE
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ABBREVIATIONS

AER	: Annual Efficiency Ratio
AIS	: Automatic Identification System
DCS	: Data Collection System
DWT	: Deadweight tonnage
EEDI	: Energy Efficiency Design Index
EEOI	: Energy Efficiency Operational Indicator
EGC	: Exhaust Gas Cleaning
ERS	: Engine Room Simulator
GHG	: Green House Gases
FW	: Fresh Water
HFO	: Heavy Fuel Oil
HRR	: Hear Release Rate
HTFW	: High Temp. Fresh Water
IMEP	: Indicated Mean Eff. Pres.
IMO	: International Maritime Organisation
JW	: Jacket Water
LNG	: Liquefied Natural Gas
LMTD	: Log Mean Temperature Difference
LTFW	: Low Temperature Fresh Water
MARPOL	: International Convention for the Prevention of Pollution from Ships
MCR	: Maximum Continuous Rating
ME	: Main Engine
MEPC	: Marine Environment Protection Committee
MDO	: Marine Diesel Oil
MRV	: Monitoring Reporting and Verification
MVEM	: Mean Value Engine Model
RMS	: Root Mean Square
SEEMP	: Ship Energy Efficiency Management Plan
SFOC	: Specific Fuel Oil Consumption
SW	: Sea Water
TDC	: Top Dead Centre
T/C	: Turbo Charger
TG	: Turbo Generator
VSP	: Variable Speed Pumps

SYMBOLS

A : Area [m^2]
a : Vibe constant
$\mathbf{C_i}$: Constants
$\mathbf{C_P}$ **-** $\mathbf{C_V}$: Specific Heat Capacity [kj/kg.K)
CD : Combustion duration [“ϴ” crank angle]
D : Piston Bore Diameter [m]
E : Energy [kJ]
H : Head (pressure) [bar]
$\dot{H}$ **- h** : Enthalpy [kJ/kg]
$\mathbf{h_c}$: heat transfer coefficient [W/m^2K]
L_R : Load Ratio of Engine [%]
LHV : Lover Heating Value of Fuel [kJ/kg]
$\mathbf{M_{air}}$: Molar weight of air [g/mol]
m : Vibe form parameter
$\dot{m}$: Mass flow [kg/s]
m_{fuel} : Fuel mass (per cycle) [kg]
N : Engine Speed [rpm]
Ni : Pump Speed [rpm]
n : Polytrophic constant
P : Pressure [bar]
$\dot{P}_T$: Power [kW]
Q : Heat [kW]
$\dot{Q}$: Heat Transfer Rate [kj/s]
R : Individual Gas Constant [kJ/K]
$\mathbf{R_u}$: Universal Gas Constant [kJ/kg.K]
T : Temperature [K]
U : Internal Energy [kJ]
$\bar{U}$: Heat Transfer Coefficient [$kJ/m^2.K$]
V : Volume [m^3]
W : Work [kJ]
w : piston mean speed [m/s]
X_{fr} : Burnt fuel fraction [%]
α : Speed coefficient
Δ : Difference
μ : Motor efficiency
λ : Air-Fuel Ratio
η : Efficiency
ϴ : Crank Angle
τ_C : Torque [kN.m]
v : Actual speed of ship [knot]

Subscribts

Actual : Actual value
air : Air
Base : Base-designated value
bp : Break power
c : Combustion
comp : Compressor (T/C)
cool : Cooling
cyl : Cylinder content
d : Diffusive
design : Initial design value
dis : Discharge
Exh : Exhaust
Elec : Electrical
FW : Fresh Water
gas : In-cylinder gas
ID : Ignition Delay
ig : Ignition
in : Inlet stream
JW : Jacket Water
L1 : Maximum Operating Point
lm : Log-Mean
lub : Lubricating Oil
M : Actual Operating Point
mech : Mechanic loses
out : Outlet Stream
p : Premixed
r : Reference
Ratio : Ratio
rad : Radiation
SW : Sea Water
TC : Turbocharger
T : Turbine (TC)
w : Wall
0 : Motoring condition

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ENERGY EFFICIENCY ORIENTED MODEL BASED INVESTIGATION OF MARINE DIESEL ENGINE AND AUXILIARY SYSTEMS

SUMMARY

The fact that transportation of goods by seaways is the most efficient method in international transportation and the increasing trend in world trade leads to an increment in the volume of sea trade and raises the number of merchant ships. There is a directly proportional relationship between industrial production and transportation of goods. As a result of growth in production, the transportation of goods has been increasing with the share of carbon emission volume of shipping in total global anthropogenic emissions. Since the merchant fleet propulsion is heavily depended diesel engines, carbon-contained fuels are in use, particularly Heavy Fuel Oil (HFO) with 79% dominancy. Carbon emissions, the combustion product, as a greenhouse gas, has severe effects on environment, like climate change. Marine Environment Protection Committee (MEPC) as a sub-committee of International Maritime Organisation (IMO) has agreed on putting forward efficiency measures to decrease carbon emissions from shipping.

Efficiency Design Index (EEDI) and Ship Energy Efficiency Management Plan (SEEMP) are two main mandatory regulations. Additionally, Energy Efficiency Operational Indicator (EEOI) was proposed to standardize the calculation method for ships' energy efficiency during operation; however, the type of calculation method will be superseded by Carbon Intensity Indicator (CII) as a new standard. With the new standard, not only the annual fuel consumption data "Annual Efficiency Ratio" (AER) will be asked but also the efficiency plans for the next three years for the ships will be required. Furthermore, the ships which cannot meet the minimum efficiency requirements, must re-new the SEEMP plans, with a procedure how will they achieve, their measuring, self-evaluation and improvement methods.

IMO GHG Strategy aims to mitigate carbon intensity of international shipping by 40% until 2030 and at least 50% up to 70% abatement by the year 2050. The reference line was determined according to the year 2008, emission levels. In the short term it can be achieved by less fuel consumption, which results in mitigation of carbon emissions. For the upcoming mid-term solutions that reducing the fuel consumption of ships by reducing energy consumption and increasing main engine efficiency. These all studies are called energy efficiency studies in maritime sector and the efficiency issue has been becoming a decisive challenge for the future shipping industry.

Although, alternative fuels without carbon are just initiated to integrate to the shipping sector, transformation of the primary power fuels to the not carbon-contained fuel will take a long time for shipping sector.

A significant amount of emission reduction was achieved after 2008 to 2015 because of slow steaming operations, the average engine loads of the ships decreased significantly because of global volume trade, efficiency regulations. However, the significant reduction comes from intrinsic relation between ship hull and speed. There

is a cubic relationship between the ship speed and required power for propulsion. Ships are designed for optimized range of operational speed. Their propeller and main engine are chosen according to planned rpm and propulsion power in the range of operational load. However, while in the slow steaming operation ships are not operated in their designated operational range. Although, operating out of the optimized operational range leads to loss of efficiency, reduced drag force with reduced speed enables lesser fuel oil consumption for per nautical mile. Nevertheless, the solution to be questioned is how long speed reduction will be a solution considering lowering speed is limited. After 2015 the efficiency increment trend has slowed down, because the ships had not reduced their speeds anymore, slightly increase and decrease can be observed at the trends.

In the light of operational trends in recent years, a brief summary was prepared for the state of the art developing technologies to reduce both fuel consumption and emission, search on the new approaches to increase the efficiency of the engines, combined with engine room equipment. It can be seen that the most type of measures were developed for reduced loads, particularly.

There are some methods applied on the shipping operations such as using limited engine power via slow steaming, fitting the ship with energy recovering installations, exchanging some equipment with more energy-efficient ones, using alternative fuels and optimisation of trim, ballast or voyage course etc.

The study of the thesis aims to decrease energy consumption used by auxiliary systems or reduce the consumption of the fuel by main engine of the ship through the implementation of efficiency enhancement methods proved by model-based approach. The approach was applied on container ship. The container ships tankers and bulk carriers represent the 65% of world fleet. Due to the relatively higher speed of container ships, slow steaming is much more popular among in container ships. With high power demand of the ships, and the increment in the sizes of the ships results in increment in installed main engine power. Increment in both capacity and main engine power, the auxiliary power demand of the ships increases. Therefore, not only the fuel consumption of the main engine but also energy demand of the auxiliary systems must be analysed. The study is carried out by three main studies, which are published articles, stated in chapters 2-3-4.

In the thesis, the energy management methods, which are applied to main engine and auxiliary systems, will be introduced with their quantitative results section by section.

Chapter one issues the main engine operational efficiency under different operational loads. The model-based approach was carried out with together with the combination of mean value engine model approach and zero dimensional model. Mechanical and thermodynamic principles are utilized to represent engine load-rpm relation and determination of in-cylinder and after processes, including, combustion, cooling, turbocharging. With the help of the developed engine model, cooling phenomenon is studied in order to calculate the potential of energy efficiency improvement in a container vessel. The effect of cooling on in-cylinder processes is discussed under variable engine loads, reduced loads, particularly. The outputs of the model is in-cylinder pressure and temperature diagram in a crank angle domain, Scavenge pressure, Exhaust pressure and temperature, turbocharging power, cooling losses, injection timing. All engine conditions are evaluated at steady state conditions. The simulation results showed that with elevated liner temperatures, as in the range of maximum continuous rating temperatures, at reduced loads, has favourable effects on generated power. If the power generated by the engine fixed with constant rpm for the ship brake specific fuel oil consumption reduces with lesser heat loss to the cylinder walls. As a consequence of keeping some part of the heat in the cylinder, increased pressure in the cylinder can be converted to mechanical energy through piston head and piston rod and that energy can be transferred to the motion of the propeller shaft via crank shaft rotation. However, not all the power can be transferred to a mechanical power, around 30% of the unrejected heat could be utilized as shaft power.

Chapter two treats the operational case of seawater and freshwater cooling system in the engine room. As can be calculated in case study one, the main engine liner cooling demand reduces significantly in reduced loads. The goal of the study is to minimize electrical power consumption during slow steaming operation, hence equivalent fuel consumption and emissions could be decreased. A computational study was carried out in order to clarify potential savings with respect to main engine load. An optimized operation of auxiliary machinery load is proposed to increase the energy efficiency aspect of the main diesel engine operation. The study quantifies the energy savings in main engine cooling system which has significant potential to reduce electrical power consumption in slow steaming operations. The values obtained from the simulator were verified using the technical manual of a diesel engine, which have similar output power range. The results of calculations show that there is great potential to improve energy efficiency when variable pumps are used. Based on the results, from the main engine cooling system, 60% of electrical power demand reduction can be achieved. The power reduction saved by pumps, decreases considerable amount of marine diesel oil as 296.2 tons used by diesel generators or oil fired boiler which corresponds 924 tons of CO_2 emission reduction and $207,300 cost saving.

The Chapter three issues the compressed air system used as a vital sub-system for the operation of main engine and its components. The compressed air is a valuable energy source in operational manner, by the reason of intrinsic lack of efficiency in pressurization process. Operational pressure and leakage rate are the major variables which affect operational efficiency of the system. This study aims to reveal potential energy saving for the compressed air system. To this end, several pressure ranges, 29-30 bars to 14-18 bars, and different leakage rates 2.4% to 45% are evaluated. After the data was obtained from ships, thermodynamic calculations had been carried out. Optimization of pressure saves 47.3% in daily power requirement, 58,2% in compressed air unit cost, 18.4 and 57.4 tons of reduction in fuel consumption and CO_2 emissions in a year respectively. High leakage rates can cause 2.7 times more power and fuel consumption. Finally, operating load, as an important indicator of compressor, makes imperfections identifiable. A detailed graphical evaluation of achieved improvement at electrical consumption need by suggested design pressure ranges was presented from both economic and environmental concept.

In the conclusion, evaluation of energy management methods in the past to present and relation with the current studies are discussed with model based approaches. Further studies can be proposed by modelling method by integrated the models, developed in this studies. Model-based approach will be a prominent solution for near future shipping to prepare more realistic and accurate SEEMP plan and observing the future obstacles for subject ship.

ENERJİ VERİMLİLİĞİNE YÖNELİK GEMİ DİZEL MAKİNELERİ VE YARDIMCI SİSTEMLERİNDE MODELLEME TABANLI ARAŞTIRMA

ÖZET

Gemiler yüzer yapılar olarak kendi kendine yetebilen sistemlerin oluşturduğu bir bütündür. Geminin sevki için gerekli gücün üretilebilmesi, yüksek güç avantajları, verimlilikleri, sistemin operasyonel güvenilirliği ve bakım tutum avantajları sayesinde ağırlıkla dizel makineler ile sağlanmaktadır. Gemi dizel ana makineleri ağır yakıtlar ve kısmen daha hafif dizel yakıtlar kullanırlar. Kullanılan bu yakıtlar %85-%87 oranlarında karbon ihtiva ederler. Ağırlıkla karbon bazlı bu yakıtların yakılması sonucu karbondioksit gazı (CO_2) ortaya çıkmaktadır. Ortaya çıkan CO_2 gazı sera gazı olarak adlandırılmakta ve küresel ısınma ve iklim değişikliğine sebep olduğu belirtilmektedir. Yapılan çalışmalar göstermektedir ki insan kaynaklı üretilen CO_2 emisyonlarının %2,2 si deniz ticareti tarafından kaynaklanmaktadır. Deniz taşımacılığı diğer taşımacılık türlerine kıyasla, birim yük başına en çevreci az emisyon salınımına sahip taşımacılık türüdür. Ancak, Uluslararası denizcilik örgütünün (IMO) yaptığı çalışmalara göre deniz ticaret hacminin artması ile birlikte salınan CO_2 gazı miktarı 2050 yılına kadar %90 ile %130 arasında artması öngörülmektedir. Ancak CO_2 gazı karbon kaynaklı yakıtların kullanılmasının kaçınılmaz bir ürünüdür. CO_2 gazı üretiminin azaltılması alternatif yakıtlara, yenilenebilir enerji kaynaklarına ya da karbon içeren yakıtlar kullanıldığı durumlarda teknolojik yenilikler ve operasyonel iyileştirmeler yardımı ile yakıt tüketiminin azalması ile mümkün olacaktır. Operasyonel iyileştirmeler, geminin gövde yapısında yapılan iyileştirmeler, seyir optimizasyonu, makine-ekipman bakım tutum, atık ısının yönetilmesi ve hız düşümü ile sağlanabilmektedir.

Uluslararası Denizcilik Örgütü (IMO), MARPOL EK-VI sözleşmesinde, gemi kaynaklı hava kirliliklerinin azalması için alt komitesi olan Deniz Çevresi Koruma Komitesi (MEPC) yardımı ile ek önlemler getirmiştir. Enerji Verimliliği Dizayn Endeksi (EEDI) ve Gemi enerji verimliliği yönetim planı (SEEMP) regülasyonları ile gemilerin daha verimli operasyonlarının sağlanması amaçlanmaktadır. Ayrıca gemilerin operasyonel verimliliğini ölçmek amacı ile Enerji Verimliliği Operasyonel göstergesi (EEOI), gemilerin yakıt giderlerinin yıllık bazda taşınan yük ile ölçmek istenmiştir. Bu operasyonel gösterge, yürürlüğe girecek olan karbon yoğunluk göstergesi olarak güncellemeye uğramıştır. 2023 yılında, gemilerin sağlamaları gereken enerji verimlikleri sınıflara ayrılacak (A-F) ve D derecesini 3 sene ardarda, F derecesini 1 sene alan gemilerin SEEMP planlarını tekrar hazırlamarı ve bu planlarda istenen enerji verimliliğini nasıl sağlayacaklarını, nasıl ölçeceklerini, değerlendirme ve geliştirme planları için ayrıntılı bir plan hazırlanması istenecektir.

Gemi operasyonlarında, gemi tipine, ekipman tipine, geminin büyüklüğüne ve operasyon yüküne göre enerji verimliliği çalışmaları her gemi için ayrıca yapılmalıdır. Genel olarak izlenilen operasyonel verimlilik uygulamaları verimliliği belirli bir ölçüye kadar sağlayacaktır. Ancak her geminin kendine uygun SEEMP bulundurması ve planların o gemiye göre yapılması gerekmektedir. SEEMP verimlilik döngüsünde

planlama, uygulama, ölçme ve düzeltici faaliyet olarak dört ana aşama vardır. Geminin operasyon özelliklerine göre verimlilik çalışmalarının belirlenmesinde, verimlilik potansiyelinin ölçülmesinde, yatırım/geri ödeme dengesinin değerlendirilmesinde, modelleme bazlı çalışmalar ile çözüm oldukça önemli bir yere sahiptir. Bu şekilde yapılan çalışmalar ile gemi üzerinde ölçülmesi zor olan verilerin dahi verimliliğe nasıl etki ettiğini anlamak oldukça kolaylaşacaktır. Enerji verimliliğinin üzerine düşen baskı ileriki yıllarda daha da artacaktır.

Tez kapsamında incelenecek olan çalışma model tabanlı, operasyonel iyileştirmelerin verimlilik üzerine etkilerini görmek için kurulmuştur. Böylece, gemilerin verimlilik çözümlerinde modelleme yöntemi kullanarak ileriye dönük daha kesin tahminler yapılabilmesi, yatırım ve geri kazanım maliyetlerinin hesaplanabilmesi, değişken operasyon koşullarına göre verimliliğin değişikliğinin incelenmesi ve enerji verimliliği yöntemleri seçiminde ön çalışmalar yardımı ile karar verilmesi için etkili bir araç altyapısı oluşturulması amaçlanmıştır.

Gemi bir sistemler bütünüdür ve yakıtın kimyasal enerjisi, ana makine ile mekanik ve ısı enerjisine, ısı enerjisi geri kazanım yöntemleri ile elektrik enerjisine, elektrik enerjisi, pompa ve çeşitli ekipmanlar yardımı ile hidrolik ve pnömatik sistemlerin operasyonu için kullanılır. Enerjinin dönüşümü kimyasal domainden, gaz, mekanik, ısı, hidrolik gibi diğer domainlere geçerken, %100 dönüşümü mümkün olmamaktadır. Sistemlerin verimlilikleri, operasyonel parametrelere göre çalışma dinamiklerinin değişkenliği model bazlı çalışma ile hesaplanmıştır.

Gemilerde bu dönüşümün en büyük ölçekte olduğu sistem gemi ana makineleridir. Günümüz şartlarında en verimli olarak sayılabilecek gemi ana makineleri %50 verimler dolaylarında kimyasal enerjiyi mekanik enerjiye çevirebilmektedir. Egzoz gazının enerjisinin geri kazanımı ile bir miktar ısı enerjisi elektrik enerjisine dönüştürülebilse de %25'lik gibi bir ısı egzoz gazı ile çevreye atılmaktadır. Bunun yanı sıra silindir ceketlerinin soğutulması, pistonun soğutulması, dizel makinelerin çalışması için en temel eleman olan turboşarjer'da sıkıştırılan süpürme havasının soğutulması gibi ısı kayıpları da mevcuttur. Bu ısı kayıplarının da bir kısmı geri kazanım potansiyeline sahiptir.

Son dönemlerde enerji verimliliği çalışmalarının en başında gemilerin seyir hızlarının düşürülmesi gelmektedir. Ancak gemilerin seyir hızlarının düşürülmesi makine-ekipman odaklı bir verimlilik yöntemi değil, gemi hızının düşmesinden kaynaklı gemi gövdesinin sürtünme direncinin azalmasıdır. Gemi hızı ile makine gücünün arasında üstel (kübik) bir ilişki olduğu düşünülürse gemi hızının belirli miktarlarda düşürülmesi ana makine gücü gereksiniminin ciddi oranda azalmasını sağlamaktadır. Son yıllarda ana makine ortalama yükleri, 2007 yılı ile kıyaslandığında, konteyner gemileri için %40, kuruyük için %25 tankerler için %30 düştüğü gözlemlenmektedir. Bu durum gemi ana makinelerinin dizayn edildiği optimum çalışma aralığından oldukça uzaklaştığını ve üretilen birim güç başına harcanan yakıt "özgül yakıt tüketimi" artışının olduğu yüklerdir. Her ne kadar gemilerin toplam verimliliği artmış olsa da makine verimliliği düşmektedir.

Diğer bir yandan gemilerin sağlaması gereken verimlilik değerleri 2008 den 2015 e kadar ciddi oranda gelişme göstermiştir. Ancak, bu gelişme büyük ölçüde gemi hızının düşürülmesi ile başarılmıştır. 2015 yılından sonra gemi seyir hızlar, artan ticaretin de etkisi ile daha fazla düşmediğinden verimlilik artışları yıl bazında çok sınırlı oranlarda kalmıştır "%1,5". Bu durumda, ilk etapta sağlanan enerji verimlilik yöntemlerinden daha detaylı yöntemlere girilmesi gerekmektedir. Ana makine verimliliği göz önüne alındığında düşük yüklerde operasyonun verimsizleşmesinin birkaç sebebi vardır. Bu sebepler, turboşarjerin veriminin düşmesi, iç dinamikleri yenmek için harcanan kuvvetlerin üretilen güce oranının artması, silindir içi düşük basınç ve sıcaklık ortalaması vs. Bu verimsizliklerin etkilerini azaltmak için birtakım çalışmalar yapılmıştır. Ancak çalışma kapsamında tespit edilen soğutma kayıplarının üretilen güce nazaran artması, silindir içerisinden bağıl olarak daha fazla ısı kaybına yol açar. Silindir duvar sıcaklıklarının gözlenmesi ile, bu kaybın ne kadar azaltılabileceği ve bu uygulamanın ana makinenin verimliliğine ne kadar etki edeceği çalışma kapsamında, ana makine modeli oluşturularak incelenmiştir. İki zamanlı bir ana makinesinin ceket soğutulması, ceket suyu çıkış sıcaklıklarının kontrolü ile sağlanır ve sabit tutulacak şekilde operasyon sürdürülür. Ancak makine yükünün düşmesi ile ceket suyu giriş ve çıkış sıcaklıklarının arasındaki fark azalır, bu azalma gömlek sıcaklıklarının düştüğünü gösterir. Gemi ana makinelerinin büyük boyutları sayesinde, enstrümantasyon kolaylığı da göz önünde bulundurularak sıcaklıkların ölçülerek ve kontrol edilerek yapılacak olan operasyonun katkıları, gemi ana makine modeli ile tahmin edilebilir hale getirilmiştir. Model yardımı ile gömlek sıcaklıkları, operasyon sınırları içerisinde çalıştığı maksimum sıcaklıkta tutularak, silindir için ısı kayıplarının ne oranda azaltılabileceği, bu ısı kaybının azaltılması ile makine gücünün ne kadar artacağı tahmin edilmiştir. Bu ısının tamamı işe dönüşemez, kalan kısmının egzoz gaz sıcaklığına etkisi, turboşarjer operasyonuna etkisi, skavenç basıncına ve döngüsel olarak hava/egzoz debisine etkisi araştırılmıştır. Son olarak, geri kazanım yöntemleri ile ne kadarının geri kazanılabileceği ve ne kadarının kullanılmadan ısıl güç olarak çevreye atıldığı, değişken düşük yük operasyonlarında koşturulmuştur. Sistemdeki tüm alt-modeller ile birlikte ne kadar yakıt ve emisyon tasarrufu sağlayacağı hesaplanmıştır.

Gemilerde seyre ve ana makineye yardımcı, havalandırma, aydınlatma ve yaşam mahalli için kullanılan elektronik cihazlar, pompalar, kompresörler, iklimlendirme-soğutma, fanlar, gibi birçok elektrik tüketen yardımcı makine mevcuttur. Bu ekipmanların elektrik tüketimlerinin takip edilebilmesi, tüketim potansiyellerinin belirlenmesi enerji verimliliği için oldukça önemlidir. Bu sebeple Enerji verimliliği dizayn indeksinde, gemilerde elektrik gücü tüketen tüm ekipmanların tüketimlerinin, çalışma sürelerinin de belirlenerek Elektrik Güç Tablolarının (Electrical Power Tables/EPT) hazırlanması gerekir. Bu tablo için ekipmanın kullanıldığı alan gruplandırılır ve anlık çektiği güçten günlük çalışma miktarına bağlı bir dizi hesaplama ile 1 gün içerisinde saatlik tükettiği ortalama güç hesaplanır. Enerji verimliliği açısından bakıldığında oldukça kalabalık olan bu listenin, daha sistematik ve derinlemesine bir çözüm için anlaşılabilir ve standartlaştırılmış bir tüketim tablosu ortaya koyar. Bu yardımcı makine gruplarından ana makine ve yardımcı sistemlere ait olan gruplar arasında en fazla güç tüketen grup olan soğutma suyu pompaları ve çalışma prensibinin yapısı gereği oldukça verimsiz çalışan hava kompresörleri incelenmiştir. Çalışmalar üç ana başlıkta farklı ekipmanları incelemiş, üç ayrı makale olarak basılmıştır. Tez bölümleri ilgili makalelerden oluşmaktadır.

Düşük yüklerde çalışan ana makinenin, soğutma yükü ve düşük skavenç basınçları sonucu giriş havasının soğutma ihtiyacı önemli ölçüde düşmektedir. İkinci çalışmada gemilerin değişken düşük yük operasyonlarında soğutma ihtiyacı analiz edilmiş, tatlı soğutma sisteminin, gerçek soğutma ihtiyacına oranla oldukça fazla çalıştığı görülmüştür. Bu durum kullanılması gereken deniz suyu ihtiyacını da ciddi ölçüde azaltmaktadır. Tatlı su ve deniz suyu pompaları sisteminde eş zamanlı devir ayarlayıcılar kullanılarak, düşük devirlerdeki soğutma talebi karşılanmıştır.

Pompaların devir basınç ve güç hesaplarında, literatürdeki verimlilik ve sistem limitleri de kullanılarak yapılan modelleme çalışmasında pompaların tüketmiş olduğu gücün çok büyük oranda düşürülebileceği görülmüştür. Ayrıca sistemde kullanılan ısı değiştiricisi modeli de pompaların akış debileri ve soğutma sularının giriş sıcaklıkları kullanılarak, çıkış sıcaklığının hesaplanmasında, sonucuna göre pompaların devir sayısının azaltılıp arttırılmasında kullanılmıştır.

Diğer çalışmada, basınçlı hava sistemi ekipmanı olan hava kompresörlerinin operasyonu üzerinde durulmuştur. Dört farkı gemiden hava kompresörlerinin çalışma kaydı ölçülmüş, hava kompresörünün çalışma sıklığı, çektiği akım, hava tüpünün basınç değişimi (doldurulurken ve operasyon sırasında boşalırken), depolanan basınç aralığı ve gemilerin tahmini hava tüketimi gözlemlenmiştir. Depolama basınçları en yüksek olan gemi seçilmiştir. Kompresörün operasyonu gereği hava sıkıştırılır ve sıkıştırma işlemi soğutma ile tamamlanarak basınçlı hava depolanır. Kompresörün havayı sıkıştırması büyük ölçüde sıcaklık artışına sebep olduğunda soğutma sonucu bu enerji sistemden geri çekilir. Harcanan enerjinin %10 u kadarı basınçlı hava tüpünde kullanılabilir durumda olduğu söylenebilir. Basınçlı hava tüplerinin maksimum basınçları 30 bar olmakla birlikte operasyon sırasında, 9 bar'a kadar düşürülebilmektedir. Hava kompresörünün çalıştığı basınçlar kompresör verimliliğine etki edeceği gibi, hava soğutulması sebebi ile yüksek basınçların elde edilmesi hava maliyetini (kWh/kg.air) oldukça arttırır. Ayrıca hava kaçakları da sızıntının tespit edilmesinin zorluğu, bir iz bırakmaması gibi durumlardan ötürü verimliliğe önemli bir etkendir. Operasyon aralığı güvenli aralıkta kalmak kaydı ile 30 bar ve 14 bar aralığındaki farklı aralıklarda belirlenmiş ve kompresör verimi, hava tüketimi, hava maliyeti göz önüne alınarak, ve sanal olarak belirlenen %2,5 baz olarak kabul edilip %45 kaçak/sızıntı miktarının model ile koşturulması ile, değişken operasyon basınçlarına ait maliyetler çıkarılmıştır. Çalışma sonucunda, operasyonel basınç değişikliğinin elektrik enerjisi ihtiyacında %50 civarında azalmaya imkân verdiği görülmüştür.

Gemilerin elektrik ihtiyacının tamamen atık ısıdan kazanıldığı durumlarda elektrik tüketiminin değişikliği sisteme direkt olarak bir katkı sağlamasa da, son yıllarda en önde gelen verimlilik uygulaması olan hız/yük düşümünde düşük egzoz debisi ile atık ısıdan kazanılabilen enerji potansiyeli azalmıştır. Isıl yükün bir kısmı, liman kazanından sağlanabilirken, elektrik ihtiyacının bir kısmı dizel jeneratörlerin çalıştırılması ile karşılanabilir. Jeneratörün yükü, düşen elektrik talebi ve ekstra elektrik ihtiyacı ana makinenin değişken operasyonel yüklerinde tespit edilmiştir. Çalışma sonunda, elektrik ihtiyacının düşmesi ile jeneratörün harcaması gereken yakıt miktarının azaldığı görülmektedir. Yardımcı makinelerde yapılan çalışmalar, yardımcı jeneratörlerin yakıt tüketiminde önemli derecede etkiye sahiptir.

Model bazlı çalışmalar, sistemlerin verimliliklerini değerlendirebilecek bir altyapı oluşturmuş, bir gemi için ileriye dönük optimizasyonların yapılması, kazanılabilecek verimliliklerin tahmin edilebilmesi, herhangi bir uygulamanın katkısının hesaplanabileceği, karışık etkileşime sahip gemi makine dairesinin daha anlaşılır düzeyde operasyonel iyileştirmeler için bir araç olacağı görülmektedir.

1. INTRODUCTION

The importance of maritime transportation in world trade has always been great throughout history in world trade. Nowadays, a great part, more than 80% of the products, in circulation in the world trade are transported by seaways.

In 2015, at the United Nations Climate Change Conference, the participated countries agreed on keeping global temperature increase under 2℃. In other words, namely "The Paris agreement", which supersedes the Kyoto Protocol by 2020, called not only participating countries both also International Maritime Organization (IMO) to initiate voluntary/mandatory applications to decrease Greenhouse Gas (GHG) Emissions, the reason of global warming, in shipping sector. Global warming has many severe consequences that results climate change. In order to reduce global warming, some fundamental precautions have been taken to mitigate and to control the amount of CO_2e emissions in both industrial and transportation area.

The rise in CO_2 emissions is highly related with increase in seaborne trade. With the expectation of growing international trade, the growth of emission amount in the future is predicted as increase with up to 50% form 2018 levels and up to 130% increase from 2008 CO_2 emission levels by the year 2050 (IMO, 2020b). The trend of seaborne trade in terms of ton.nautical-mile (tnm) and related emission indicators carbon dioxide equivalent CO_2e with the Energy Efficiency Operational Indicator (EEOI) (g.CO_2/tnm) and Annual Efficiency Ratio (AER) (g.CO_2/dwt.nm), are demonstrated in Figure 1.1, for the recent years.

The global trade and total emission amount in shipping had shown an increasing trend until 2008. The CO_2e (CO_2e emissions represented with CO_2 equivalent) had decreased immediately right after 2008 due to the limited world trade. Then as seen in the figure with the new measures of energy efficiency, a noteworthy reduction could be achieved in spite of the increasing demand for the seaborne trade. After 2015, although an increment seen in CO_2e, limited reduction had continued in carbon intensity indicators (EEOI, AER) with the help of the efficiency measures in shipping.

However, the total amount of CO_2e emissions had increased and therefore, the increment trend calls additional precautions and increasing the pressure on energy efficiency issues.

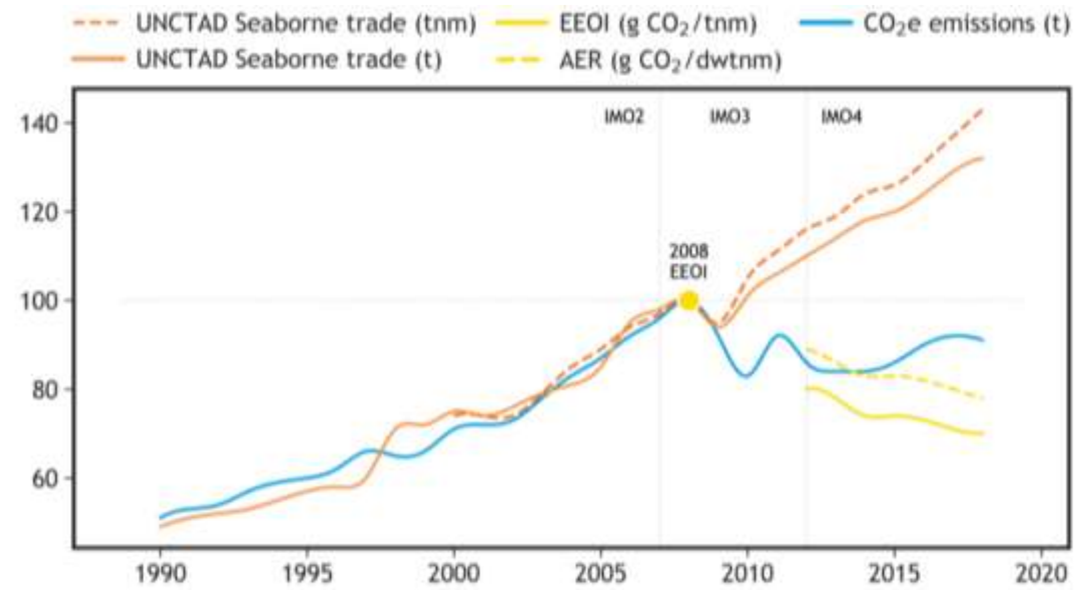

Figure 1.1: Seaborne trade during last three decade (IMO, 2020b).

In recent years, substantial amounts of efforts have been performed to reduce fuel consumption and maintain energy efficient operations in shipping sector. Energy Efficiency Design Index (EEDI) (IMO, 2012a) and Ship Energy Efficiency Management Plan (SEEMP) (IMO, 2011) are two main mandatory regulation amended by IMO and its sub-committee Marine Environment Protection Committee (MEPC). In addition to stringent environmental regulations and policy, also the oscillating freight rates, petrol price fluctuations, and inconstant global volume trade are putting the situation of maritime sector in more challenging conditions.

While the IMO's focus was mainly on maritime safety before the 1954, now, the responsibility scope have been expanded which is safety, security, efficiency and decreasing the marine sourced pollution. Control and mitigation of maritime pollution mainly conducted by Marine Environment Protection Committee, which considers marine pollution issues as a technical body of IMO. The IMO adopted the convention "International Convention for the Prevention of Pollution from Ships" MARPOL in 1973 , to prevent pollution for marine environment by oil and the scope of the convention has been gotten wider and wider in recent years as garbage, sewage, chemicals, air pollution and emissions (IMO, 2020a). The Annex VI, in 1997, was adopted to mitigate air pollution and emissions from ships particularly. The regulations were put into effect in 2005 and after an update in 2008; the more strict limits were put into force in 1st July 2010. As seen in the Figure 1.1 the strict limitations about air pollutions showed their effect as a reduction of CO_2e emissions, and carbon intensity indicators with incentive regulations after entering into force.

Additionally, IMO performs a study about greenhouse gas emissions within a certain periods. The last study (IMO, 2020b) comprehends the years between 2012 and 2018 after 6 years from the last one. In the report, the emission rates the, emission reduction trends, projections about the growth of maritime trade and emissions were estimated. Using AIS (Automatic Identification System) sources, the accuracy of the projections had been improved. The CO_2 emission estimations are performed by three methods that are two "Bottom-up" methods and a "Top-down" method by IMO. The bottom up methods are divided into vessel based and voyage based methods. The fuel consumption of the ships is determined by operational parameters and information provided by ship owners regards to sizes' of the ships and departure/arrival ports of the voyage, respectively. The top-down method is performed with the data provided for fuel oil consumption "Data Collection System (DCS)" or for European Union's Monitoring Reporting and Verification (MRV) Scheme. A voyage-based data is collected by these systems as; the ships subject to provide information about its fuel consumption, voyage duration and distance, and the cargo carried during voyage on an annual basis. The estimations of CO_2 emissions show good agreement between bottom-up methods and top-down method. IMO projects CO_2 emissions for the future, thorough these gathered data. In this regard, IMO put Initial Strategy for the reduction of GHG emissions. As an comprehensive strategy, time table consist of three step approach with the initial strategy, short term (2018–2023), mid (2023–2030) and long term (2030–2050) measures to be on the way of low carbon shipping. The global implementation of energy efficiency regulations will have taken a process.

IMO GHG strategy aims to decrease carbon emissions from ships by 40% until the year of 2030 and the further targets is that the 70% decrement in the year 2050 (IMO, 2018a). Prevent and control pollution in order to lessen the effect on environment with the zero carbon emission perspective is planned to be achieved in this century as soon as possible. Among the methods currently used, the energy efficiency enhancement methods are in the foreground, and the additional researches are being carried out for new efficiency solutions and techniques along with the methods whose preparation phase has completed.

Adaptation of alternative marine fuels are promising and applicable method for diesel engines. The most prominent alternative fuel application is using LNG as a marine fuel with lesser carbon content than HFO. Additionally using methanol is the fourth most used fuel in maritime sector but, it must be underlined that dominancy of HFO usage is around 79% in the market (IMO, 2020b). Obviously, the application of using environmental friendly fuels in the marine transport cannot be brought into play in a short period and that requires a long transition time. Hereof, using alternative fuels and non-carbon fuels are considered as long-term solutions to be applied all the shipping industry. Considering that the use of zero-carbon fuels will take time, the prominent solution for the next years is to decrease of the consumed fuel in ship power plants to mitigate the CO_2 emissions. This can be possible with decreasing the energy demand of ship, and auxiliary machines or increasing the efficiency of the main engine.

In short, this thesis focuses on the proposing solutions for the reduction carbon emissions as well as the reduction ship fuel oil consumption with the help of modelling tool. The modelling tool was applied on the main engine and its auxiliary systems in the ship power plant. The concept of the thesis was determined and conducted with the help of the regulation indicators, which will be explained in the following subsections and the understanding of energy efficiency concept in marine engine room. In an engine room, the efficiency elements can be classified in to two main groups as power generation and power consumption, the main subsections of the two elements are represented in Figure 1.2, which will be narrowed down in the following subsections.

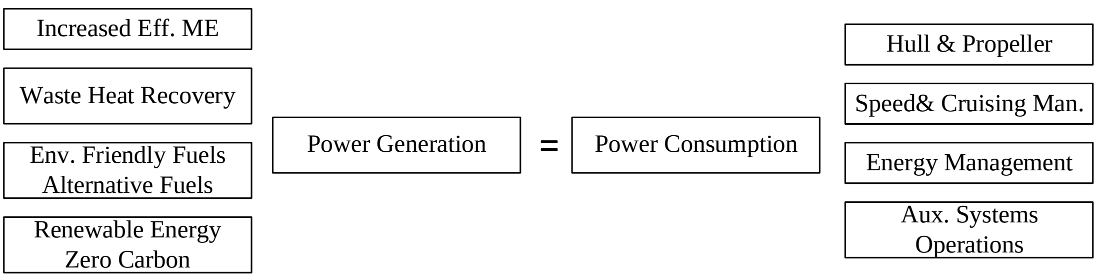

Figure 1.2: Two main elements in engine room, to be followed for the energy efficiency solutions.

The fuel consumption of a ship is described by the given equation 1.1. In the given equation, the total fuel consumption is written by the help of main engine and auxiliary engines fuel consumptions.

$$Fuel\ Consumption = \sum_{i=1}^{nME} P_{ME(i)} SFC_{ME(i)} + (P_{AE} SFC_{AE}) \qquad (1.1)$$

In cases where energy efficiency increasing equipment are used, as like power take in units (PTI), the fuel consumption formula is updated by considering the effects of the units on consumption. Besides, EEDI value is obtained by multiplying carbon dioxide factor of the fuel used in the engine, with energy saving equipment addition to the formula, which will be given in the regulatory background sub-section.

Energy efficiency regulations have been guiding in both the determining the scope of the thesis and the direction of the study carried out. The straight effects of the equipment's fuel consumption, included in the calculation of SEEMP and EEDI within the rules, on the efficiency are indicators that the results of the studies to be carried out on these equipment will make a favourable contribution to energy efficiency. Electric Power Tables (EPT) included in EEDI calculation, indicates the potential electric usage of all auxiliary machines for the any kind of ship in 24 hours' time interval. All the electrical consumers are put in the document and there cannot be left out of listing. Accordingly, since the energy demand of engine room systems is provided by auxiliary generators, their effect on energy efficiency is as important as the fuel consumption of the main engine. Although electrical energy can be recovered from the waste heat of exhaust gases, ship operational conditions and energy needs differs from designed parameters. In recent years, new operational solutions have come up as slow steaming, and it seems the reduced loads will remain as a permanent solution in the following years. Manufacturers and operators are proposing new solutions for new operational trend as explained in the technical efficiency literature in the following sub-section. After reviewing of proposed technical solutions, the thesis proposes some additional applications for the both ME and auxiliary systems in terms of energy efficiency for actual operating trend of the ships.

The results of the proposed methods were obtained with the help of models developed in the thesis. Main engine, the pump with the heat exchanger and the compressor are investigated in the thesis. The modelling tool from which the results are obtained also allows for additional studies with integrating sub-models. The modelling study, carried out in the thesis helps to planning and execution of efficiency applications by giving time advantage and capability of estimating saving potentials.

1.1 Statement of the Problem

Although ship transportation is the most economical and environmentally friendly type of transportation for per unit load in the world trade, it has a large share in environmental pollution due to the size of the transported volume. UNCTAD reported that, the commercial world fleet has more than 95,000 ships with 1.97 million dead-weight tons (DWT) carrying capacity. Furthermore, the capacity of world fleet is increasing every year. As indicated in the report in 2019 by UNCTAD that, the carrying capacity had increased by 2.6% while trade volume increased by 2,7% which one was 4.1% in previous year 2017 (UNCTAD, 2019). The expectations for annual increase in trade is 3.4% annually. The fastest grow belongs to the container and dry bulk carriers and it is expected that the container trade will have been 6,0% growing each year until 2023 (UNCTAD, 2019). IMO is trying to take precautions and legislating some limitations to reduce emissions while shipping capacity is increasing. In greenhouse gas study of IMO showed that the 2,2% of global CO_2 emissions has been derived from shipping sector and an estimation that it can increase by 80% to 130% by the year 2050 (IMO, 2020b). Although the shipping sector has relatively lesser effect on air pollution compared with the land-based transportation, however the impact of shipping emissions on total anthropogenic emissions have been increasing. Total CO_2 emissions had increased 34,793 billion tonnes to 36,573 billion tonnes, from 2012 to 2018. In these years the contribution of shipping on the CO_2 emissions had increased from 962 million tonnes, with the share 2,76%, in 2012 to the amount of 1056 million tonnes, with the 2,89% share in 2018 (IMO, 2020b). The worldwide anthropogenic CO_2 emission change and the contribution of the shipping to the emissions during recent years are given in Table 1.1 (IMO, 2020b). There is both increasing trend in share and total amount in emissions. The objective of the regulations is to diminish the CO_2 emissions at least 50% of the emission volume of the year 2008, in this century.

Table 1.1: The variation of CO_2 emission in recent years (million tonnes).

Year	Global Anthropogenic CO_2 emissions	Total Shipping CO_2 emissions	The CO_2 emissions Ratio of Shipping to Global
2012	34793	962	2,76%
2013	34959	957	2,74%
2014	35225	964	2,74%
2015	35239	991	2,81%
2016	35380	1.026	2,90%
2017	35810	1.064	2,97%
2018	36573	1.056	2,89%

The mitigation of the carbon emissions will continue to be one of the biggest challenges in future shipping sector. Consuming carbon-contained fuels in ship engines produces CO_2 emission as an inevitable combustion product. The amount of carbon emission is mainly depended on carbon content of the fuel and how much fuel consumed by the ship's power plant. The dominance of fuel consumption is heavily based on HFO in the market by 79% of total fuel consumption which is followed by marine diesel oil (MDO), Liquefied Natural Gas (LNG) and limited amount of methanol (IMO, 2020b). The main relationship between energy efficiency and environmental concerns is the CO_2 emission, which causes the global warming as an greenhouse gas (GHG).

The market of engines is dominated by diesel engines, the use of slow-speed diesel engines is preferred because of their high-power output and efficiency advantage. The dominancy of the engine types used in ships is given in Table 1.2. Although the share of slow-speed diesel engines is in decreasing trend in recent years, it is still the most common type of engine used in ships. Dominancy of the diesel engines, using carbon based fuel show the carbon emissions will continue to be a serious problem in shipping sector. The slow-speed diesel engines has relatively higher power output than the other type of engines. Therefore, it is preferred that to install in the big size ships.

The fuel is not only consumed by the main engines. The operation requires additional electrical power or steam depending on the ship type or operation. While electrical energy can be supplied by shaft generators or a waste heat recovery system, in some cases, auxiliary boilers and auxiliary diesel generators are the main solutions to meet the ship's demand as there is not enough resources to be converted or such a lack of equipment efficiency equipment.

Table 1.2: The distribution of engine types installed in ships.

Engine Type	2012	2013	2014	2015	2016	2017	2018
Slow-Speed Diesel Engine	42,3%	40,7%	39,8%	39,2%	39,1%	38,8%	38,6%
Medium-Speed Diesel Engine	34,7%	34,7%	34,2%	33,9%	34,0%	39,9%	33,6%
High-Speed Diesel Engine	21,6%	23,2%	24,5%	25,4%	25,4%	25,7%	26,2%
LNG-Otto Slow Speed	0,002%	0,006%	0,009%	0,009%	0,016%	0,025%	0,036%
LNG-Otto Medium Speed	0,116%	0,147%	0,198%	0,237%	0,276%	0,306%	0,348%
LNG Diesel	0,002%	0,002%	0,002%	0,006%	0,024%	0,046%	0,076%
Lean Burn Spark Ignited	0,038%	0,042%	0,051%	0,060%	0,057%	0,060%	0,057%
Methanol	-	-	-	0,003%	0,013%	0,013%	0,014%
Gas Turbine	0,113%	0,110%	0,110%	0,106%	0,112%	0,103%	0,098%
Sail	0,281%	0,273%	0,276%	0,286%	0,290%	0,295%	0,305%
Steam Turbine	0,549%	0,503%	0,489%	0,463%	0,425%	0,417%	0,420%
Batteries	0,002%	0,002%	0,002%	0,003%	0,003%	0,007%	0,023%
Non-Propelled	0,245%	0,299%	0,315%	0,308%	0,275%	0,273%	0,311%

The fuel consumptions of the ship types, respect to the main engines, auxiliary generators and the boilers are shown in Figure 1.3. as the end users of the fuel. The main engine energy is used for the propulsion, auxiliary engine consumption is dedicated for electrical energy and boiler consumption is for heat use in a ship. The power demand of auxiliary machineries can be identified with equation 1.2 and 1.3. In the Figure 1.3, it can be noticed that three main types of ships as; bulk carrier, container ships and oil tankers are the dominator of fuel consumers in the shipping trade market. These types of ships represent 65% of fuel consumption. And also it must be emphasized that, both the ship sizes' and installed main engine power have increased in these type of ships (IMO, 2020b).

$$P_{AE(MCR>10000kW)} = \left(0.025x\left(\sum_{i=1}^{nME} MCR_{MEi} + \frac{\sum_{i=1}^{nPTI} P_{PTIi}}{0.75}\right)\right) + 250 \qquad (1.2)$$

$$P_{AE(MCR<10000kW)} = \left(0.05x\left(\sum_{i=1}^{nME} MCR_{MEi} + \frac{\sum_{i=1}^{nPTI} P_{PTIi}}{0.75}\right)\right) \qquad (1.3)$$

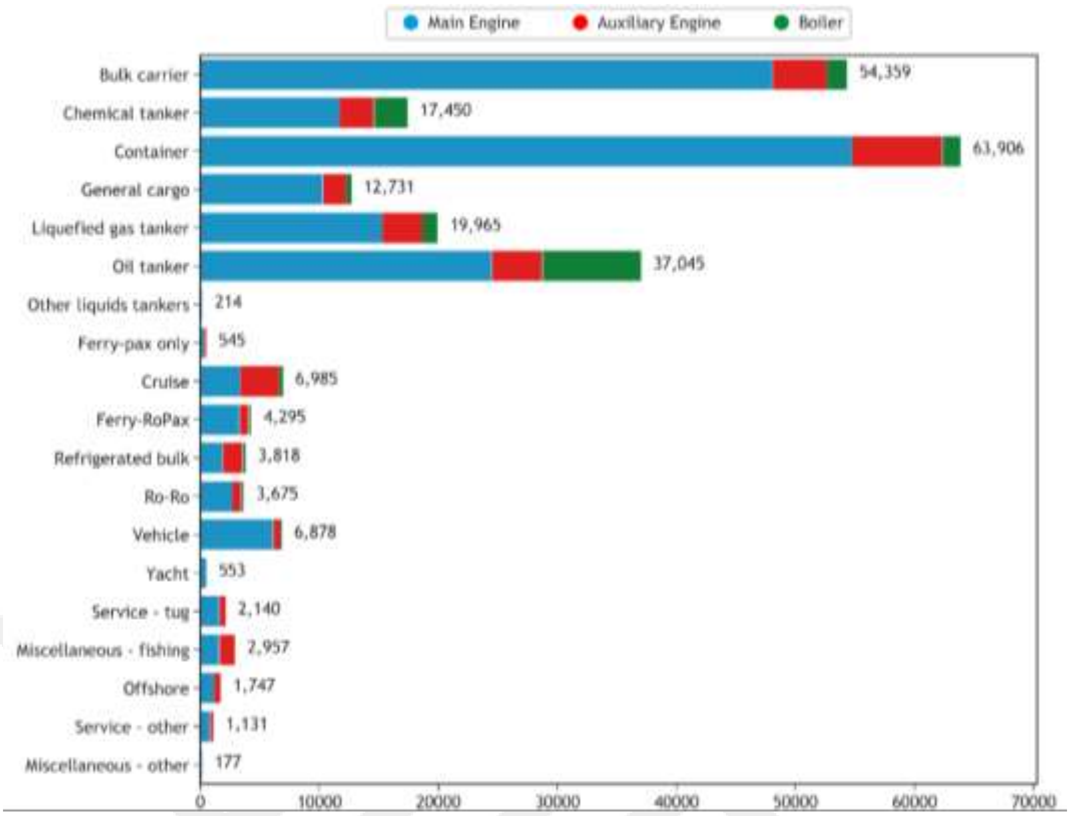

Figure 1.3: Distribution of ships' HFO-equivalent fuel (million.tonnes) consumption of machineries (IMO, 2020b).

In the duration of 2012 to 2018, the trend of operational speed had fluctuated. While 2015-16 years were the highest operational speed in this period, in 2018 the average operational speed was close the minimum speeds. Thereby, the CO_2 emission rates had increased in the period 2015-16 which shows, the amount of the trade and market dynamics are still the main drivers for the sailing speeds and associated fuel consumption with the emissions (IMO, 2020b). In the Figure 1.4, it is shown that average operating speed fluctuation of the ships, dominator of the total fuel consumption.

The reference value "1" was set in the year 2012. The change of the values over the years to 2018 are represented in the Figure 1.4. The size of the ships and accordingly installed main engine power had increased. Although, the design speed of the new built ships had remained constant, the average operational speed increased in 2016 and after decreased to lower rates. A noticeable decrease in fuel consumption had come out with both the operational speed reduction and enlargement of ship sizes notably in 2014 and 2015. However, as mentioned before, reduction in speed is highly depend on volume of the trade and market dynamics and the trend changed reversely in last years.

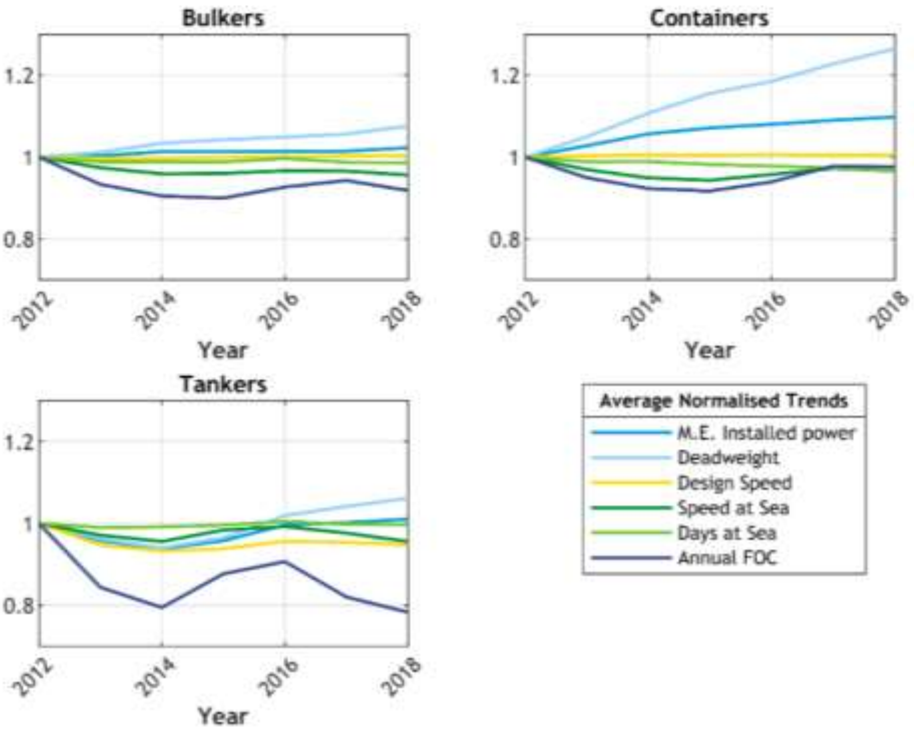

Figure 1.4: Trends indicator graph for the three type of ships(IMO, 2020b).

In the beginning of the last decade, the speed reduction is the most accepted way to reduce fuel consumption and increase energy efficiency. However, the level of achievable energy efficiency via speed reduction is limited. Therefore, in the early years of energy efficiency applications the slow steaming, which is initially seemed to be a prominent and successful method, has lost its effect in recent years and the momentum in increasing energy efficiency has slowed down. The change of energy efficiency indicators, which are used to describe of Carbon Intensity Indicator (CII) are demonstrated in Table 1.3 along with years since 2008(IMO, 2020b).

Table 1.3: Four main indicator of carbon emissions by ships (IMO, 2020b).

Year	EEOI (g.CO_2/t/nm)			AER (g.CO_2/DWT/nm)			DIST (kg.CO_2/nm)			TIME (t.CO_2/hr)		
	Value	Variation vs 2008	Variation vs 2012	Value	Variation vs 2008	Variation vs 2012	Value	Variation vs 2008	Variation vs 2012	Value	Variation vs 2008	Variation vs 2012
2008	15,16	—	—	7,4	—	—	350,36	—	—	4,38	—	—
2012	12,19	-19,6%	—	6,61	-10,7%	—	387,01	10,5%	—	4,74	8,1%	—
2013	11,83	-22,0%	-3,0%	6,4	-13,5%	-3,2%	380,68	8,7%	-1,6%	4,57	4,1%	-3,7%
2014	11,29	-25,5%	-7,4%	6,2	-16,1%	-6,1%	382,09	9,1%	-1,3%	4,54	3,5%	-4,3%
2015	11,3	-25,5%	-7,3%	6,15	-16,9%	-6,9%	388,62	10,9%	0,4%	4,64	5,8%	-2,2%
2016	11,21	-26,1%	-8,1%	6,09	-17,7%	-7,8%	397,05	13,3%	2,6%	4,77	8,7%	0,5%
2017	10,88	-28,2%	-10,8%	5,96	-19,5%	-9,8%	399,38	14,0%	3,2%	4,79	9,2%	1,0%
2018	10,7	-29,4%	-12,3%	5,84	-21,0%	-11,5%	401,91	14,7%	3,8%	4,79	9,2%	1,0%

The data is the result of aggregated information from ships. The demonstrated values represents voyage based information of the 88% of the world merchant fleet, first six main type of ship, which is responsible for 98% of CO_2 emissions of the world shipping. From the year of 2008 to 2012, a noticeable reduction could be achieved in EEOI, AER, DIST and TIME. In spite of a good rate in those 4 years, the reduction rate could not be kept as in the same trend. While in the first 4 years 2008 to 2012, the EEOI could be reduced by 19,6% but, at the end of the following six years period carbon emission mitigation could be achieved 10% more with 29,4% from reference values in 2008. Annual Efficiency Ratio (AER) is the indicator of CO_2 emitted, for per DWT per nautical miles. This is the new feature of energy efficiency indicator. Naturally, AER is highly affected by reduction of speed, but it is influenced by another parameter, the ship size, while it can be decreased by enlargement of the ships. In the Figure 1.5, it can be seen that the effect of ship size on AER. In the axis of X represents the size groups of the ships "1" is the smallest ship size. From the year of 2008 to 2012, 10,7% reduction level could be reached, then in the following six years period 11,5% more improvement could be achieved with the 21% reduction from 2008 reference data. As can be seen that the enlargement of vessels became an efficiency solution for the new ships.

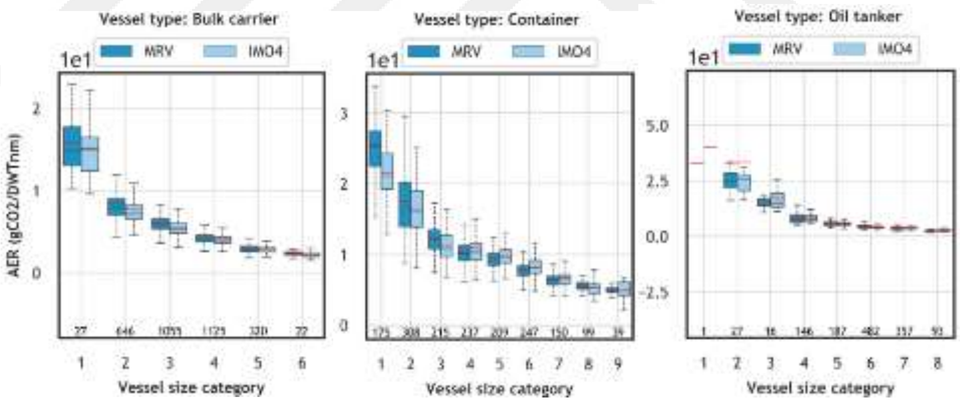

Figure 1.5: The variation of AER (g.CO_2/DWT/nm) as regards to ship size (IMO, 2020b).

On the other hand, DIST and TIME had showed reversed response to the reduction in first two indicators. As a result of increasing ship dimensions, these two indicator had increased. Although, speed reduction puts decreasing effort to the decreasing of these values, enlargement in ship sizes is the key driver of the trend. As can be seen in the Table 1.3 the achieved reduction rate have been proceeded with a good trend between 2008 and 2012 when the decreased trade volume allowed slow steaming. Unfortunately, the reduction amount in carbon intensity had not showed linear trend after 2012 and the reduction rate slowed down. After 2015 with the help of the technological advancement in engineering systems 1-1,5% reduction rate could be achieved (Rutherford, 2020), howbeit, higher reduction rates are required for the environment. For this reason, authorities agreed on there is a need putting more pressure on energy efficiency applications to force shipping with restricted emission limits.

IMO Initial Strategy comprises slow steaming, course optimisations, implementation alternative fuels (Joung et al., 2020). Slow steaming had been accepted as a reliable and permanent solution for shipping industry than ten years (Wiesmann, 2010). The average speed reduction rates, comparison with the year 2007 are given in Table 1.4 for container ships.

Table 1.4: The average engine loads of container ships respect to year and size.

Ship type and size (DWT) category	Average main engine loads at sea							
	2007	2012	2013	2014	2015	2016	2017	2018
Container								
000-999	0.62	0.55	0.54	0.52	0.51	0.51	0.5	0.48
1,000-1,999	0.58	0.5	0.48	0.46	0.46	0.47	0.46	0.46
2,000-2,999	0.58	0.42	0.37	0.35	0.35	0.37	0.39	0.39
3,000-4,999	0.59	0.41	0.36	0.33	0.32	0.33	0.33	0.33
5,000-7,999	0.63	0.37	0.34	0.32	0.31	0.31	0.34	0.34
8,000-11,999	0.69	0.35	0.34	0.34	0.33	0.38	0.4	0.4
12,000-14,499	0.67	0.36	0.34	0.35	0.36	0.4	0.42	0.41
14,500-19,999	-	0.26	0.29	0.35	0.54	0.6	0.6	0.56
20,000-+		N/A	N/A	N/A	N/A	N/A	0.34	0.51

In addition to environmental concern, there is also economic reasons for the energy efficiency as an incentive reason for the companies. Although oil prices are not high in recent years, fuel cost have always constitutes a significant part of ship operating costs. Increasing energy efficiency level of the ship will provide a considerable amount of reduction in the operational expenses. To provide competitive freight rates and increasing the profitability rate for companies will also be possible by investing in energy efficiency applications. There are many energy efficiency methods that have been applied in the maritime sector until now, to mention them briefly under main headings, they can be gathered as like cruise-based (course optimising, weather etc.), educational/training-based (efficiency - maintenance trainings etc.) and machinery-based methods (waste heat recovery, etc). Some of them requires additional concerns to reach sustainable development.

The ship's engine room and auxiliary systems are interconnected and include highly interacting systems. While it is possible to handle some systems which are operating independently, also there are systems that dependent on each other should be considered in a whole concept. For the efficiency applications, in the planning stage, which is the first step in energy efficiency studies, the system dynamics should be well understood and planned. Measurement tools and evaluation method should be well determined. Modelling studies are quite successful in offering convergent solutions with the better understanding of the effects of complicated systems and to predict the results. By modelling approach, with the help of the basic operating principles of the systems, highly detailed and high accuracy results can be obtained which are also depending on working conditions of the ship or main engine. Before the efficiency measures to be applied on ships, pre-estimations can be calculated with the modelling tools, as how much energy efficiency can be achieved and the payback period of the investment to be made can be calculated. In addition, modelling, which allows the pre-evaluation of the expected results from the system, provides a pre-correction advantage if necessary. It shortens the implementation period for energy efficiency measures, ensures that corrective actions to be decided as soon as possible, reduces the work load to the ship's personnel and is a very useful tool for the evaluation and re-planning phase in SEEMP.

The need for the study is defined by the following rationales;

- ✓ The share of the anthropogenic emissions of shipping still has been increasing; moreover, the emission reduction trend had slowed down after 2015, when significant amount of reduction had been accomplished.
- ✓ IMO short-term strategy still depend on speed reduction, unfortunately transition of zero-carbon energy will take process. There is still requirement to find aggressive solutions according to operational trends.
- ✓ The Regulation 22 of MARPOL annex VI, which will be explained in sub-section Regulation background, will enter into force in 2023. The ships, which cannot meet the Carbon Intensity Indicator (CII) requirement, should revise their SEEMP plans by defining their efficiency measures.

- ✓ In recent years, as a result of slow steaming, which is an important energy efficiency practice, main engines have been operating at inefficient loads. The practices should be proposed to increase efficiency at reduced loads.
- ✓ Despite the decrease in total fuel consumption during slow steaming, due to the reduced main engine load, insufficient amount of steam and electrical energy supply from waste heat cause additional fuel consumption by the auxiliary machineries. Auxiliary systems' operation must be re-considered with low energy demand perspective.

Implementation of efficiency methods takes time, as a cycle of planning, implementation, observing and re-evaluation. The modelling method overcomes these obstacles its time efficient, high reliability of the results, its applicability for different vessels with different dynamic conditions due to operational differences. Engineering fundamentals will be emphasized, and energy efficiency potential solutions, which will be provided in the scope of the thesis on machinery and auxiliary systems, will be sought. The results of the developed models are compatible with the reference values of the diesel engine operating data.

1.2 Purpose and scope of Thesis

Improving the energy efficiency in ships, which have conventional propulsion, leads reduction in fuel consumption and hence carbon emissions. By modelling the components, diesel engine and its auxiliary systems, the thesis covers the investigation of potential reductions in fuel requirement of ship.

1.2.1 Purpose of the thesis

The purpose of this thesis is to estimate the energy saving potential of the main engine and auxiliary systems of a ship. An engineering approach study is carried out for the ships' energy efficiency problem, encountered in future. The ship is an integrated system; the measurement of each consumer efficiency level while in the operation is quite difficult.

The solution attainability with modelling tool will be investigated in the thesis for marine diesel engine systems, in order to support the low carbon shipping solutions.

To achieve the objective two main questions are answered in studies.

- ✓ What can be improved more with evaluating the operational trends of recent years by considering realistic operating conditions of ship systems instead of only one-design conditions?
- ✓ How can an operational understanding of a system can be improved by modelling method and which kind of operational variables of that element or system must be observed to analyse and improve the efficiency?

In light of answers to these questions, the study searches the potential for increasing energy efficiency and reducing fuel consumption with the help the modelling method considering recent years operational trends. The proposed efficiency measures cover potential solutions that main engine efficiency increment or decrement of the energy requirement of the other systems. The prerequisite to achieve energy efficiency is to estimate the potential savings in the ship operations.

The model-based investigation of energy efficiency of the ship requires enhanced understanding of among main engine system components. Main engine, Turbocharger, pumps, compressors etc. The all system also related with the operational conditions of the ship. Considering that the operational loads of the ship's auxiliary engines and the cooling requirement are depend on the main engine load. The effect of varying operational parameters on energy efficiency and carbon emission is evaluated. Based on the savings achieved and the equivalent fuel reduction, it is calculated that how much fuel is saved on an annual basis and how much less carbon emission is emitted. The calculation saving amount is carried out according to the operational load of the ship, and they made for each load and annually over an operational load profile of the ship.

1.2.2 Scope and delimitation of the study

Energy efficiency methods and emission reduction methods are applied in several ways in shipping sector. The reduction of carbon emissions is possible by reducing fuel consumption and energy demand for the ships that are using carbon-based fuels. In addition, the use of low carbon fuels is also considered as an emission reduction method. For this purpose, the use of alternative fuels can be considered is one of the main methods in achieving emission reduction in addition to other measures.

In the IMO 4th GHG study (IMO, 2020b), it was stated that wide range of alternative emission reduction methods can generally be grouped into four main class. As a first step, 44 different applications, indicated in the study, can be sub-grouped with 16 characteristic groups than can be aggregated within these four main category. The main categories are listed below.

- ✓ Energy saving technologies
- ✓ Renewable energy sources
- ✓ Alternative fuels
- ✓ Speed reduction

The detailed applications about emission mitigation are represented in Table 1.5.

Table 1.5: The list of technologies to reduce carbon emissions and reduction rate.

	Technology	Description	Potential Reduction
	Energy Saving Technologies		
1	Main Engine Improvements	Main Engine Tuning, Common-rail, Electronic engine control	0,25% - 0,45%
2	Auxiliary Systems	Frequency converters, Speed control of pumps and fans	0,87% - 1,59%
3	Steam Plant Improvements	Steam plant operations improvements	1,30% - 2,13%
4	Waste Heat Recovery	Waste heat recovery, Exhaust gas boilers on auxiliary engines	1,68% - 3,09%
5	Propeller Improvements	Propeller-rudder upgrade, winglet, fins, contra-rotating	1,40% - 2,40%
6	Propeller Maintenance	Propeller performance monitoring, polishing	2,20% - 3,95%
7	Air Lubrication		1,35% - 2,26%
8	Hull Coating	Low friction coating	1,48% - 2,55%
9	Hull Maintenance	Hull monitoring, brushing	2,22% - 3,90%
10	Opt. Water Flow Hull Openings		1,64% - 3,00%
11	Super Light Ship		0,28% - 0,39%
12	Reduced Auxiliary Power demand	Low energy lightning etc.	0,40% - 0,71%
	Use of Renewable Energy		
13	Wind Power	Towing kite, sails, wings,	0,89% - 1,66%
14	Solar Panels		0,18% - 0,30%
	Use of Alternative Fuels		
15a	Use of Alternative Fuels with Carbons	LNG, FC, Methanol, Ethanol	5,54% - …
15b	Use of Alternative Fuels without Carbons	Hydrogen - Ammonia - Synthetic Methane, Biomass Methane, Synthetic Methanol, Biomass Methanol, Synthetic Ethanol, Biomass Ethanol	0,10% - 64,08%
	Speed Reduction		
16	Speed Reduction	Speed Reduction by 10%	7,38% - 7,54%

In the table addition to given methods, an estimation for potential carbon emission reduction rate is given in terms of percentage in the associated row. The estimations have been carried out for the years 2030 and 2050 (IMO, 2020b).

The entire table comprises all applications about marine applications, except battery solutions and carbon capture solutions because of lack of certainty of information.

The use of alternative fuels and renewable energy sources need a thoroughly change in the power system and they are out of the scope of the diesel propulsion power plant.

Speed reduction, slow steaming in shipping parlance, was a viable way to reduce fuel consumption of the ship. However, in recent years the trend of the slow steaming had changed due to market conditions. The direct advantage of the slow steaming is that reducing power demand for the propulsion of the ship via relatively low speeds. The required power has a cubic relation with ship speed and by a slight reduction in speed, a significant amount of fuel savings can be achieved. However, this phenomenon can be applied when sufficient time is available for the voyage. If the charter timing is not convenient for speed reduction, the method is not an applicable solution. On the other hand, the volume of the trade and the freight rates are the other two important parameters playing a role in the decisions of shipping companies. In the last years, ships started to sail faster than in 2015-16 levels due to the high freights and high volume rates. Although some successful studies, as a project, were carried out for just in time arrival to the ports, the arrival time and adequate speed reductions were implemented just last 24 hours before arrival to port with an efficient communication with the ship due to the availability of the port.

The reduction rate in speed seems reached their operational limit in last few years, after 2015. Although there is still improvement in total fuel consumption reduction for operating costs of the ships, more slowing down affects the profitability of the companies in merchant fleet market based on nowadays' trade volume and freight rates. On the other hand technological developments yields significant savings in fuel. Considering both the operational trends in recent years and the engine technologies together with auxiliary machineries, additional fuel savings solutions could find out. The model-based solution is a method that integrate systems behaviour with operational conditions. The thesis will search for the additional savings with recent years operational profile with reduced engine loads and reduced auxiliary system requirements because of reduced load, not laden ships as like cruising with less payload or ballast.

1.3 Background and Related Work

In this section, two subhead is explained as, regulatory issues, legislated by IMO and the applications applied for marine power plant for efficiency.

1.3.1 Regulatory background

The control and prevention of pollution by ships to the marine environment implemented by "International Convention for the Prevention of Pollution from Ships" (MARPOL) adopted by IMO, is the primary international regulation in this field and covers 99% of shipping trade (IMO, 2019b). The related regulation for the energy efficiency and reduction of carbon emissions is "Annex VI Prevention of Air Pollution from Ships" which was entered into force 19 May 2005. The MARPOL Annex VI aims to reduce sulphur oxide and nitrogen oxide emissions and ozone depleting substances as well as greenhouse gas (GHG) emissions. Marine Environment Protection Committee (MEPC) concentrated on environment friendly topics, adaptation of regulations, under the responsibility of IMO. Energy efficiency Design Index (EEDI) and Ship Energy Efficiency Management Plan (SEEMP) with the Energy Efficiency Operational Indicator (EEOI) are the efficiency tools for the strategy of reduction of greenhouse gases (GHG) emissions from ships.

EEDI

The goal of the EEDI is consumption of minimum energy (minimum emission) in per mile for transportation of a unit capacity of goods. By increasing the level of strictness in each five-year steps, the regulation forces shipping companies to use more environmental friendly equipment and the more efficient designs to increase their efficiency level of the operation. The regulation does not restrict ship charterers/companies to choose the appropriate efficiency application for their operation. They are free to choose the application, which is economically practical as long as the specified energy efficiency limit (reference EEDI line and also specified reduction rate) is met. The calculation of reference lines of the EEDI for each type of ship are determined by using the ship data build in last 10 years before 2010 for the type of ships as container ships, bulk carrier, gas carrier, general cargo, refrigerated cargo carrier and multi-purpose ship with regression method. These ships cover 85% of CO_2 emission in global merchant fleet (IMO, 2019a). To estimate the reference point for specified type of ship, carbon emission factor "C_F" assumed as 3,1144g CO_2/g.fuel, specific fuel oil consumption of main engine SFC_{ME} 190 g/kWh, 75% of maximum continuous rating power of main engine "P_{ME}", specific fuel oil consumption of auxiliary engines "SFC_{AE}" 215 g/kWh, and no energy efficiency technology are assumed.

The EEDI value was determined by the Eq. 1.4 (IMO, 2013).

$$Estimated\ Index\ Value = 3{,}1144 \frac{190\ \sum_{i=1}^{NME} P_{MEi} + 215 P_{AE}}{Capacity\ . V_{ref}} \tag{1.4}$$

The EEDI calculation for the container ships, the capacity was taken 70% DWT of the ship. The reference values of EEDI for each kind of ship is estimated by regression method with ship data set for each type of ship. The formulation is depended on Deadweight of the ship with two constant parameters.

Each five-year steps, which are 2015, 2020 and 2025 (Exceptional advancing the date as 2022), are the initial date of target reduction rates for that five-year step period. After the year 2020, MEPC agreed on to advance the date of reduction rates for some ships. IMO will have been strengthen the limits with the phase 3 EEDI index by April 2022. This regulation was designated to enter in to force by the year of 2025 but with the new amendment, IMO advanced the date of come into operation. Therefore, the new ships constructed after the phase 3 step must be much more efficient, especially for LNG carriers, and general cargo ships. Additionally, much more stricter limits will come into effect for container ships. The limits of the ships were set according to their DWTs. From the year of 2022, The variation of EEDI reduction limits by the years are as in Table 1.6 for all kind of ships.

In the last MEPC meeting (IMO, 2020), calculation methods were reconsidered for the efficiency level of the ships. According to calculation methods, the ships will be subject to the Annual Efficiency Ratio (AER), the rating of ships in terms of efficiency level. The ratings, which are determined as A to E will be decided according to the Carbon Intensity Indicator (CII) of the ships. Another technical measure that came up recently, is Energy Efficiency for Existing Ships (EEXI), which also aims to mitigate CO_2 emissions from ships. CII and EEXI will be determined as a function of EEDI.

CII will be an operational measure and to satisfy this measure, it is planned that implementation of Carbon Intensity Code before the 1st Jan 2023. In the case of that ship graded with "D" for consecutive 3 years or "E" for 1 year, which means poor levels of energy-efficient operation, SEEMP plans will be required to be re-prepared, inspected, and approved.

Table 1.6: The EEDI reduction rates for all ships according to DWT.

Ship Type	Lower Limit DWT	Upper Limit (less than) DWT	Phase 0 1 Jan. 2013	Phase 1 1 Jan. 2015	Phase 2 1 Jan. 2020	Phase 3 1 Apr. 2022	Phase 3 1 Jan. 2025
Bulk Carrier	20,000	-	0	10	20	-	30
	10,000	20,000	n/a	0-10	0-20	-	0-30
Gas Carrier	15,000	-	0	10	20	30	
	10,000	15,000	0	10	20	-	30
	2,000	10,000	n/a	0-10	0-20	-	0-30
Tanker	20,000	-	0	10	20	-	30
	4,000	20,000	n/a	0-10	0-20	-	0-30
Containership	200,000	-	0	10	20	50	-
	120,000	200,000	0	10	20	45	-
	80,000	120,000	0	10	20	40	-
	40,000	80,000	0	10	20	35	-
	15,000	40,000	0	10	20	30	-
	10,000	15,000	n/a	0-10	0-20	15-30	-
General Cargo Ships	15,000	-	0	10	15	30	-
	3,000	15,000	n/a	0-10	0-15	0-30	-
Refrigerated cargo carrier	5,000	-	0	10	15	-	30
	3,000	5,000	n/a	0-10	0-15	-	0-30
Combination Carrier	20,000	-	0	10	20	-	30
	4,000	20,000	n/a	0-10	0-20	-	0-30
LNG Carrier	10,000	-	n/a	10	20	30	-
Ro-Ro (vehicle)	10,000	-	n/a	5	15	-	30
Ro-Ro (cargo)	2,000	-	n/a	5	20	-	30
	1,000	2,000	n/a	0-5	0-20	-	0-30
Ro-Ro (passenger)	1,000	-	n/a	5	20	-	30
	250	1,000	n/a	0-5	0-20	-	0-30
Cruise Ship	85,000 GT	-	n/a	5	20	30	-
	25,000 GT	85,000 GT	n/a	0-5	0-20	0-30	-

Additionally, the ships graded with A or B levels of CII will be encouraged by the new regulation. The calculation method, reporting, and the plans to adjust the actions to meet the required CII level will be included in SEEMP. The assessment will be based on Data Collection System (DCS) data for each year.

With the implementation of phase three IMO planned that the energy efficiency level of the new built ships will be 30% more efficient than the ships built in 2014.

SEEMP,

The SEEMP is the plan that enables observing the ships' or fleet's performance in terms of efficiency to keep up with the progress in efficiency level while in the operations.

Each shipping company must regulate their SEEMP plan for their individual ship and their fleet. In the program of implementation of SEEMP for a specific ship, the steps which must be followed can be divided as four main parts as; a) planning, b) operation, c) monitoring and d) self-evaluation and improvement (IMO, 2012c). After the evaluation step "d", the planning stage of SEEMP "a" can be reconsidered and sustainable improvement can be satisfied by repeating all steps. Together with the measures taken in the first cycle, a report that predicts how much efficiency will increase should be included in this plan, along with the measures that can be taken in the next stage. In order to execute application step "b", a certain period is determined in the implementation of the plan. After the decision of the start and end dates of the application, records are kept and the results and benefits of the operation are observed. Keeping records is crucial for further developments. However, the observation process must be assessable by shore-side not to put excessive workload to ship personnel. The consideration of the results is handled by the evaluation tool EEOI, which is an effective tool to measure the effectiveness of the plan. The quantitative calculation method is used as a primary monitoring tool. It should be noted that the calculation process must be well determined in the planning stage before implementation. It is very important to collect data for monitoring and evaluation to be continuous and reliable. The evaluation phase, where the results of the application will be seen, should shed light on efficient operation and create meaningful data for the next cycle. For the next planning phase, it is necessary to examine where the problem is and what is working or not.

Generally, the first selected applications are with a high efficiency and initial cost ratio applications by avoiding increased workload.

There are some applications that to improve efficiency in the ships as speed optimization, weather routing, just-in-time applications, well-planned routes, maintenance and optimum design of hull and propulsion system, optimum trim and ballast, using autopilots for steering and rudder control system, waste heat recovery systems, energy management systems, renewable energy usage, alternative fuels, using on-shore energy source etc. (IMO, 2016).

The energy saving equipment is described as which affects the main engine efficiency or auxiliary engine efficiency in the operation. Electrical energy efficient technology or mechanical energy efficient technology. The EEDI formula is in terms of g/t.nm (gram CO_2 emission per carried capacity in per nautical mile) in Eq. 1.5 (IMO, 2018b).

$$EEDI = \frac{\left[\prod_{J=1}^{M} f_j\right]\left[\sum_{i=1}^{nME} P_{ME(i)} C_{FME(i)} SFC_{ME(i)}\right] + \left(P_{AE} C_{FAE} SFC_{AE}\right)}{f_i\, f_c\; Capacity\;\; f_w\, V_{ref}} + \frac{\left\{\left[\prod_{J=1}^{M} f_j \sum_{i=1}^{nPTI} P_{PTI(i)} - \sum_{i=1}^{neff} f_{eff(i)} P_{AEeff(i)}\right] C_{FAE} SFC_{AE}\right\}}{f_i\, f_c\; Capacity\;\; f_w\, V_{ref}} - \frac{\left[\sum_{i=1}^{neff} f_{eff(i)} P_{eff(i)} C_{FME} SFC_{ME}\right]}{f_i\, f_c\; Capacity\;\; f_w\, V_{ref}} \tag{1.5}$$

Where P power of the main and auxiliary engines with $_{ME}$, and $_{AE}$ subscripts. C_F conversion factor for generated CO_2 emission by fuel type, which is seen in Table 1.7.

Table 1.7: Carbon conversion factor (CF) fuel to CO_2.

Type of Fuel	Reference	Lower calorific value (kJ/kg)	Carbon Content	CF ($t\text{-}CO_2/t$-fuel)
Diesel/Gas Oil	ISO 8217 Grades DMX through DMB	42.700	0,8744	3,206
Light Fuel Oil (LFO)	ISO 8217 Grades RMA through RMD	41.200	0,8594	3,151
Heavy Fuel Oil (HFO)	ISO 8217 Grades RME through RMK	40.200	0,8493	3,114
Liquefied Petroleum Gas (LPG)	Propane	46.300	0,8182	3,000
	Butane	45.700	0,8264	3,030
Liquefied Natural Gas (LNG)		48.000	0,7500	2,750
Methanol		19.900	0,3750	1,375
Ethanol		26.800	0,5217	1,913

SFC is specific fuel oil consumption of different engines, $f_{eff(i)}$ is the technological innovation in main or auxiliary engines to decrease engines' fuel consumption, $P_{PTI(i)}$ additional power input to propulsion if there is exist, V_{ref} is the reference speed of the ship. The bottom side of the equation f_w is the speed reduction due to the weather and sea conditions, f_i is the capacity factor, f_c cubic correction factor which varies due to the cargo type of the ship and the Capacity is the deadweight (DWT) of the ship.

Data Collection System

Additionally, Data Collection System (DCS) which is entered into force 2018 March, indicates that the ships 5000 grt or above which represent 85% of total CO_2 emission (IMO, 2018a) should provide the methodology how to collect the fuel oil consumption data (for all fuel types used in the ship) and reporting procedure to their flag state in the SEEMP. Additionally, Monitoring, Reporting and Verification (MRV) system for European Union is also valid in Europe in order to collect data about fuel consumption and carbon emissions. The fuel consumption and emission can be measured by, bunker delivery notes, flow meters, tank monitoring, direct emission measurements or combined. Furthermore, the flag state is asked to declare the fuel oil consumption data to IMO Ship Fuel Oil Consumption Database. The mandatory data collection system will provide a firm database and will help for future findings.

IMO estimates annual CO_2 emission reduction will reach up to 19% to 26% by 2030, which means up to 420 million tonnes CO_2 per year (IMO, 2019a). The environmental advantage and fuel saving cost advantage will encourage the investment to energy efficiency measures in shipping industry.

1.3.2 Efficiency improvement methods in engine systems

In order to ensure energy efficiency in maritime in a complete manner, it is necessary to deal with the all chain of stakeholders from the first point to the end such as design phase then shipyard, main engine and machinery manufacturers, operators, operational planning management and ports. Therefore, it is very difficult to consider the efficiency with a lumped way in all lifelong range; on the other hand, it is also impossible to consider while focusing on one point or on a limited range. It will be possible to practice the entire efficiency frame in shipping by performing the analogy that form part to whole. While determining the spotlight, an extensive analysis of the target should be performed including relationship with related other parameters or systems. The key concept is to reduce the whole frame to a specific spotlight state.

Although, IMO or any other organisation does not address any kind of application, the best practices are stated by MEPC (IMO, 2016) for the reduction of carbon emission applications, carried out in shipping. These efficiency measures were classified in 4 groups in previous section as; Management/Operational Measures, Energy Saving Technologies, Using Renewable Energy and Alternative Fuels.

"Operational level measures" for instance; fleet management, improved voyage planning, weather routeing, just in time arrival, smooth shaft power, optimized ballast and trim, autopilot finally yet importantly, the most popular one "speed reduction. These applicable emission abatement technologies with their emission reduction potential in the 2030 to 2050 given by the IMO 4th GHG study which are listed in Table 1.5.

The measures are; "Energy saving technologies" such as; main engine improvements, auxiliary systems and steam plant improvements, waste heat recovery, propeller maintenance and improvements, hull maintenance, coating, air lubrication, optimizing water flow, super light ship, reduced auxiliary power demand.

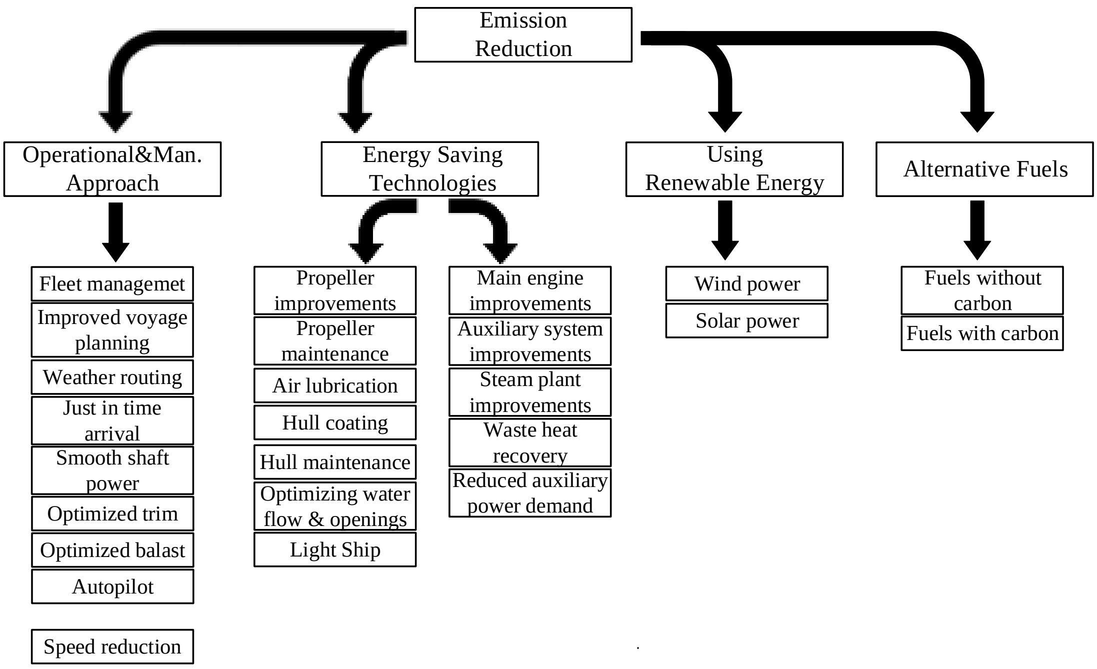

Figure 1.6: Emission reduction strategies, applied in shipping sector.

As for using renewable energy sources, they can be listed as; Wind Power, Solar Panels. On the other hand, Using of Alternative Fuels with and without carbons. The measures which are applicable all measures to reduce carbon emissions in shipping are seen in Figure 1.6.

This chapter will focus on the survey of thesis scope, in practice. The efficiency improvement “Energy Saving” Technologies about the marine diesel engines are reviewed briefly in this section. In Figure 1.6 they were listed in two groups as left and right groups. The left group, as propeller or hull improvements, represents the technologies before in operation while in the construction or design stage which can be implemented by shipyards. The second group (the right side) is the machinery technology group that can be changed after put in operation or designed for the constructed ship by equipment producers. If the groups on the right and left for energy saving technologies are compared, it can be claimed that the right group allows modification after the ship starts its operation life than the left group. The methods, included in technological improvements, are given in detail with several subsections. The ship construction related side (the left side) have not been considered in the scope of the thesis study. While main engine improvements, auxiliary system improvements and reduced auxiliary power demand approaches were studied. The narrowed down category is demonstrated in Figure 1.7.

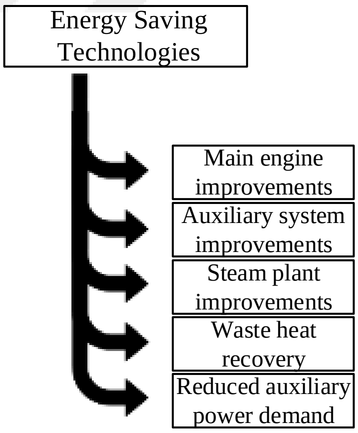

Figure 1.7: Energy saving technologies for marine engine systems.

The detailed literature search will be given in the main sections (the sections 2, 3 and 4) of the thesis for the related research. In this section, state of art technologies were reviewed to find out a gap and light the way of the frame about marine power systems. The technologies explained below.

Data driven performance monitoring and controlling systems;

Real time monitoring of the engine (Performance Monitoring); against the ship performance degradation by along with the years, adjusting of maximum pressures is a solution for recovering performance of main engines. By continuously measuring the cylinder pressure in the real time operation and setting them to an optimum efficient value, which is set by the programmer gives fuel-efficient operation for the ship (MAN Diesel and Turbo, 2021). This technology allows controlling the maximum pressure for each cylinder and allows identification of the problem. Real time monitoring is applied for the main engine cylinders. Both for mechanical and electronically controlled engines have options to be fitted, as in MAN engines they are called as PMI-VIT and PMI-Autotuning, respectively. 0,20-0,25 g/kWh fuel savings could be achieved by adjusting the mean maximum pressure value per bar(MAN PrimeServ, 2020c).

With the monitoring of engine parameters, new digital aided solution ideas have gained importance in recent years. The project "INTENS" was carried out by VTT Technical Research Centre of Finland with help of other stakeholders in industry. The project has pursued toward simulation methods with many studies in the field, as data driven approach (Mia Elg et al., 2020), collecting on-board data (Manngârd et al., 2020), cloud based optimisations (Korvola et al., 2020), dynamic modelling and simulation (Dhanasegaran, et al., 2020), and digital twinning (Hautala, et al., 2020). The simulation of the ship energy systems had been carried out by researchers, in order to estimate the efficiency potential, to find out optimum solutions for the ships. The model-based approach, studied by Guangrong (2017) is based on the data driven approach unifying the dynamic responses of the ship with holistic optimisation. The study, addresses the problem about poor-design at the planning stage because of early decisions, made without the model-based approach while only few information available for design. After the ship put into operation, the model performs as an energy flow simulator, to compare real data in monitoring, diagnostics, controlling purposes (Guangrong, 2017). The energy flow was described with different domains, the distribution of energy is represented in the time domain with the help of subsystems' input data comes from real operation (Guangrong et al., 2013). With this method, dynamic energy balance could be examined. The flow plan of the energy could be demonstrated in a multi domain model, which is built in Matlab Simulink/Simscape environment with the modelling of main components. The electricity consumption, heating and cooling requirements and operating profile were used as an input data for the examination of the energy efficiency technologies as, thermal storage, Organic Rankine cycle, electricity distribution, cooling and electricity distribution in order to reduce the fuel consumption of the ship (Mia Elg et al., 2016).

Turbocharger Adjustment;

Today, diesel engines, while generating the desired high amount of power, turbochargers have the foremost effect in meeting emission requirements and reducing fuel consumption via increasing the volumetric efficiency. Therefore, any improvements to be made in turbocharger technology has a great importance on emission rates. The air amount, required for combustion from scavenge air, which affects the pressure in the cylinder is provided by the most vital engine component, turbocharger. Turbocharger consist of two main components turbine and compressor side, connected via shaft. Naturally, significant technological improvements had been carried out on turbocharger technology. With high efficiency turbochargers, retrofitting solutions, variable turbine area, two stage turbochargers, turbocharger cut of, sequential turbocharging and by-pass solution, due to the operational profile of the engine to obtain optimized fuel consumption.

If the engine fitted with three or more turbocharger units, at the low load operations one of them can be cut out to utilize the lesser exhaust energy with lesser turbocharger (Guan et al.,, 2015). Investigation of cut out of Turbochargers were carried out by many researchers (Hountalas et al., 2014; Kim and Kim, 2019) and significant amount of saving and emission advantages were found at part load operations. Another solution for reduced load is the turbochargers with retrofitted with variable area turbines (VTA), which allow controlling exhaust flow through the turbocharger. By controlling the scavenge pressure with controlling the turbine area, according to mechanical limits and stress of blades, leads provide optimum air to send into the combustion chamber for scavenging and combustion, which leads reduced consumption and emissions (Srivastava et al., 2019). It also gives an advantage during transient operations by controlling VTA (Xiros and Theotokatos, 2011) studied with models (Bahiuddin et al., 2017). Conventional turbochargers are optimised for EEDI reference loads (75% ME Load), according to the manufacturer reports around 2,5% fuel reduction could be achieved by applying VTA (MAN Turbocharger, 2016).

Although, by controlling injection timing and exhaust valve timing up to 3g/kWh saving could be achieved by electronic control unit (ECT) (MAN Diesel and Turbo, 2019). On the other hand, the turbochargers can be adopted to the main engine for reduced operational load profile with high-pressure tuning system (HPT) that gives significant SFOC advantages at low load operations but at higher loads, high amount of exhaust gas is not allowed to pass through turbocharger with exhaust gas bypass system (EGB), not to overload it. In that cases, by-pass way is accepted as a solution with advantage at reduced loads but comes with high SFOC values at higher loads. The samples of conventional turbo-compound and variable area turbine are seen in Figure 1.8. Two stage turbochargers (sequential) are other solution, which are generally available for medium speed and high-speed diesel engines (B. Wu et al., 2019) with extended operational range advantage against to the instrumentation disadvantage. On the other hand apart from the turbine side compressor side optimisation is also prominent solution for medium speed marine diesel engines. Since the turbine side can operate relatively smooth efficiency range than the compressor, the compressor side can be adjusted for reduced loads, separately (Ntonas et al., 2020).

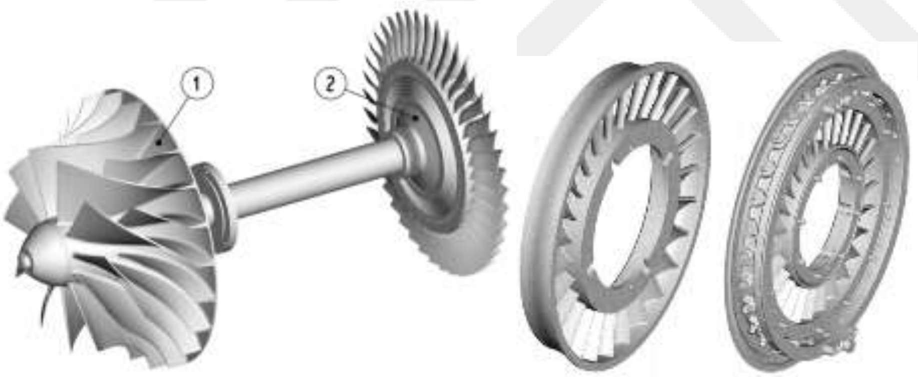

Figure 1.8: a) Compressor wheel and Turbine rotor (left) b) Fixed nozzle ring c) adjustable nozzle ring, variable area turbine (VTA)(right) (MAN Diesel and Turbo, 2013d).

With the all these technologies about turbocharger the main purpose is to perform the most efficient operation to provide required air for the combustion with optimum ignition pressure for specific engine load. Adjusting the optimum air can lead saving in the fuel consumption for diesel engines with the reduced emissions. The all technologies as mentioned above the comparison curves with reference line and with each other are represented in Figure 1.9.

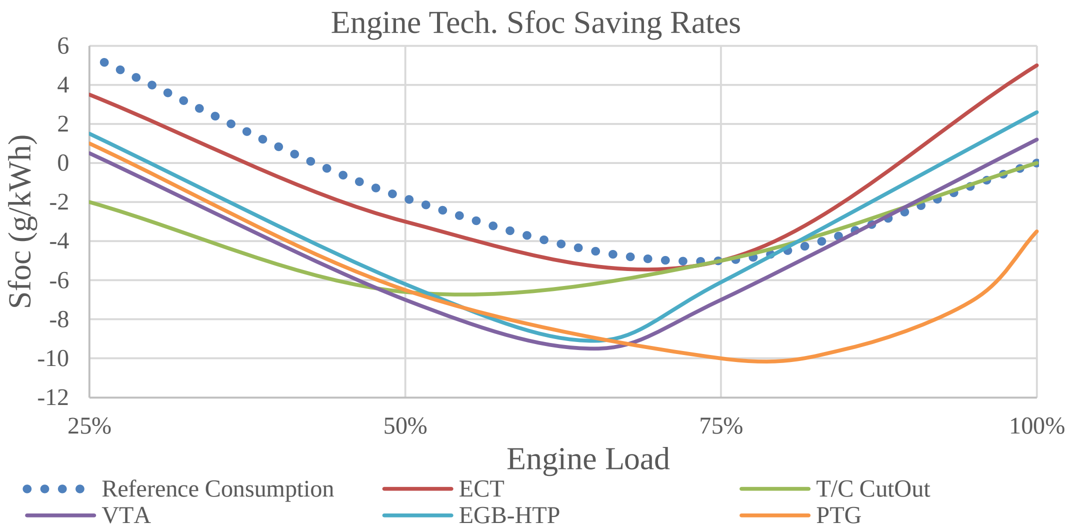

Figure 1.9: Comparison of T/C technologies with the effect of fuel consumption (MAN Diesel and Turbo, 2020a)

Exhaust Valve Optimized Control;

Variable cam profile by hydraulic assisted timing control system. The new exhaust valve control system has advantage at low loads without any disadvantage at high loads. Particularly at slowsteaming 2-3 g/kWh fuel saving can be achieved at 50% main engine load (MAN PrimeServ, 2020a) it can be reduced even in more reduced loads. The control of the cam profile focuses on the lift for the closing of exhaust valve. The valve control sample is demonstrated at Figure l.10.

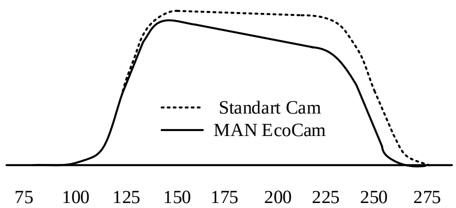

Figure 1.10: Valve lift comparison with hydraulic actuator control (MAN PrimeServ, 2020b).

By closing exhaust valve in early timings, particularly in reduced loads under 60% of ME load leads significant amount of savings in fuel consumption in terms of g/kWh. The representation of the saving amount given by MAN is shown in Figure 1.11 with respect to engine loads up to 6g/kWh is achieved (MAN PrimeServ, 2020b).

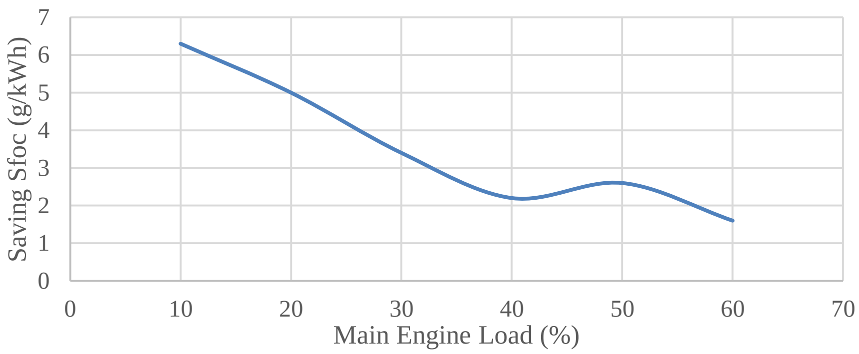

Figure 1.11: The potential saving amount respect to engine load with controlled cam (MAN PrimeServ, 2020b).

As seen in Figure Saving, this method is effective in the power range of 10 to 60% of ME. The saving rate increases as the power load of ME decreases. Additional exhaust valve control at the point of opening time, is applied to emission control and adjustment of exhaust gas temperature with adverse effect in fuel consumption. However, the idea of the latter method is not saving fuel consumption, that is to adjust exhaust gas temperatures for emission control methods as like after treatment.

Derating Engine Power;

The application above are mostly proposed by reduced load of the engines. Ships and their engines are designed to operate specified speed range in order to cover optimal operational range of the ship's most common operational profile along with their operational life. If the operational load of ME and ship speed show decreasing trend as slow steaming or low load operation which also means operational frequency load of the ships falls below the designed speeds, the limitation of the machine power "de-rating" can be proposed as a reasonable option. Additionally as a complementary optimisation, "the retrofit of the propeller" with reduced power and rotational speed is implemented for a holistic approach to operational level. With the help of the combination, engine-propeller match, 4-8% fuel savings are accessible saving rates with 10-15% reduction of main engine power. This optimisation can not be considered with out other mentioned reduced load optimisation methods.

The speed reduction can be considered as both "company management level" and "technological method". While in sailing, just in time arrivals or reducing speed in order to decrease fuel consumption for related voyage is addressed by ship management office. On the other hand, some technical solutions have been put in practice in engine room for main engine or engine room equipment. Although, operating out of the optimized operational range leads to loss of efficiency, reduced drag force with reduced speed enables lesser fuel oil consumption for per nautical mile. Nevertheless, the solution to be questioned is how long speed reduction will be a solution considering lowering speed is limited. For further improvements, an additional significant amount of fuel-saving advantage can be gathered by derating the main engine and revising the propeller according to new operational profile. Derating of the main engine power, compatible with the operational requirements helps to afford energy saving strategies that came in recent years. Slowing down about 10-15% of ship speed, coupled with optimized propeller can reach 10-12% consumption reduction with priory less power demand, and help of engine tuning, optimizing the SFOC range of the engine, refurbished (rematch) turbochargers, new propeller comes with improved propulsion efficiency (MAN Diesel and Turbo, 2013b).

Injection Improvement

The quality of the injection is a vital phenomenon for combustion. Start and the propagation of the combustion have a direct influence on combustion efficiency. With increasing the quality of the fuel spray, injection process, supports for better formation of combustion, by the help of increasing number of injector holes. While increasing the numbers, reduction in the size of the nozzles are also complementary application for the solution. Without any modification on the engine, up to 7g/kWh fuel savings can be achieved (MAN PrimeServ, 2015) by this solution in particular at high load ranges, claimed by MAN, the engine manufacturer. The variation of potential fuel savings according to main engine load is demonstrated in Figure 1.12 in terms of g/kWh fuel consumption.

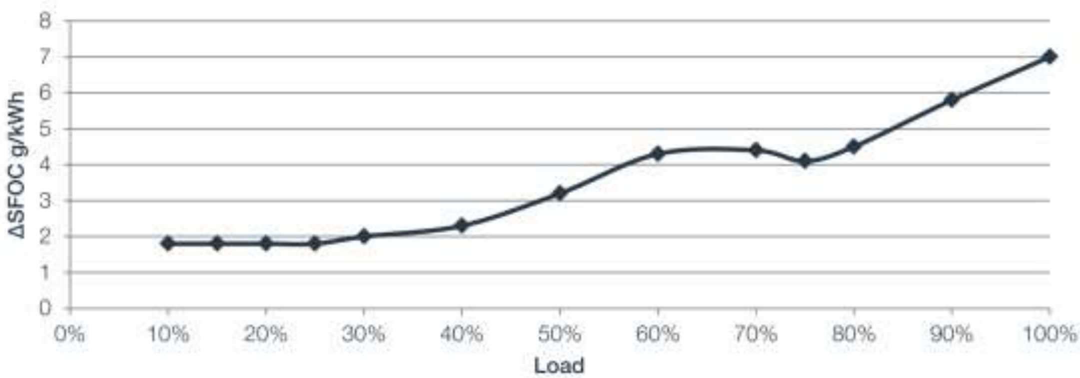

Figure 1.12: The potential saving with nozzle improvement as a function of engine load (MAN PrimeServ, 2015).

Another improvement in injection quality is the better atomisation to increase combustion quality. The enhancement of the spray quality can be achieved by reduction of sac volume in injector designs (US Patent No 6,360,960, 2002). The sac volume is the space that between valve seat and injector needle, thus by eliminating the unfavourable fuel drips from injector holes leads better combustion quality which allows saving in fuel consumption and emissions (Dreeben, et al., 1992). The application of the engine part modification, also called as slide fuel valve, reduces the smoke content and unburned hydrocarbons amount, which causes fouling in burned gas flow while also reducing NO_x emissions particularly at low load operations (MAN, 2010). Additionally, injection pressure is also crucial parameter that maintain injection quality together with nozzle holes, numbers, sac volume. As a common practice in low speed marine diesel engines, unit fuel injection system, supplies fuel to the cylinder separately with injection pump however, the system does not provide the same amount of injection pressure for every load/rpm of the main engine. While the main engine with cam shaft controlled “MC” having relatively reduced injection pressure at low ranges, electronic controlled “ME” type engines have high pressures for all load range (MAN Diesel and Turbo, 2013c). With high-pressure injection line “common rail system” enables injector to work with higher injection pressures with in all injection process and with the help of the high pressure, better atomization control of combustion for emissions (Okubo and Kuwahara, 2019). The last part of the injection is timing which is mentioned in engine tuning/ performance section.

Waste heat recovery systems

Although prime mover of ships, main engines have very high output power ratio to the input energy, substantial amount of energy is wasted through exhaust gas as a heat energy. The amount of heat energy wasted via exhaust gas is around 25% (Theotokatos et al., 2020) of input fuel energy while mechanical power output is around 50%. Recovering the heat energy from the source in order to increase the efficiency level of ship power plant, the waste heat systems are utilized (Theotokatos and Livanos, 2013). As well as thermal demand, met by the waste heat recovery systems, a basic need for the ship, the electrical power demand is also be served by waste heat recovery system, which operates with water/steam working fluid or organic fluids (Larsen et al., 2014). In addition to exhaust heat energy source utilization in general in waste heat recovery systems (Zheshu Ma et al., 2012), scavenge air heat source utilization is another additional high amount of source to increase the efficiency of the system (Mito, Teamah et al., 2018). Due to the complexity of the system, single pressure and dual pressure systems are exist in studies. Although dual pressure system has slightly more advantage, it cannot be considered as s profitable option due to the higher initial investment cost additionally, it can be claimed that during the part load operations, single pressure stage waste heat recovery systems is advantageous with regard to economic, technical, and environmental issues (Theotokatos et al., 2020). Ship size and operational profile is the main parameters for the profitability of the waste heat recovery systems. Steam/water and organic fluids are common practices in existing literature, however waste heat recovery research is not covered by the thesis, there will be some calculations for the main engine efficiency enhancement practices, but the studies about organic rankine cycle have not been taken in to account neither in the thesis scope and the engine efficiency improvement calculations. For the sake of brevity, to observe the general efficiency increment potential for waste heat systems, a study conducted by Inegiyemiema M. (Inegiyemiema, 2018) for a chemical tanker vessel. A mean value approach was utilized for the calculations. Exhaust gas recovery potential was estimated as the heat can be extracted until 45 ℃ for exhaust gases. After WHR system numerical analysis for 100%, 80% and 50% ME loads, the efficiencies were found as 60,7%, 61% (maximum efficiency at 80% ME load) and 60%, respectively. This study have showed maximum recovery capability of WHR system. While the exhaust gas heat is recovered in steam cycle, Jacket cooling water, charge air cooler, and the lubrication oil cooling are the other heat sources for ORC systems.

Thermal Storage Systems

Thermal storage systems are used with the combination waste heat utilization systems in the ships. There is a highly related technique called as thermal storage systems, which are used for recovering low-grade thermal power together with the organic rankine cycle systems (ORC) (Fridolfsson and Kuosa, 2017). In the case of imbalance of heat consumption and waste heat generation in a system, the thermal storage systems provide an advantage to stabilize required heat source for the ORC system. The system is convenient for the ships has inadequate match between heat production and heat consumption. The excessive heat is stored in a material “Phase Change Material” (PCM) that has high latent heat with high thermal conductivity properties (the process can also be called as a charging). When the heat need arises, the stored heat comes to play to compensate the insufficient heat amount. The heat is extracted from the PCM with the help of Heat Transfer Fluids (HTF). By this method, the continuous operation of ORC can be maintained.

Power Take Off system improvements

The electrical power demand of the ships can be meet by shaft generator systems, as a strong alternative for auxiliary generator sets during voyage. Although, the waste heat utilization is the most reasonable and environmental friendly method to generate electricity, in some cases as small sized main engines, low exhaust heat energy or relatively high demand of electricity, the shaft generators are the second advantageous option for electricity production in ships because of using the mechanical energy source's high efficiency level "main engine efficiency". If a two-stroke main engine considered, the mechanical power generated by the engine, has higher efficiency rate (>50%) than the four-stroke gen. sets.

While just installing a shaft generator, "Power Take Off (PTO) can be considered an efficiency enhancement method in ship power plant, because of lower fuel consumption of main engine to generate the same power, comparing with the aux. generators, the equipment need more attention to increase the operational efficiency of running this equipment. During the voyage, the main engine speed is not fixed and it does not operates at a designed constant speed in its operational life. Due to the mechanically connected system of the PTO, the shaft generator has not fixed operational speed naturally. As a solution for the variable speed of the shaft converting the mechanical energy AC-DC-AC is a prominent option for fluctuating speed of the shaft (Yijian, 2007) however the technology allows not a wide range of operational profile. In order to overcome these problems, Induction Machine (IM) shaft generator, which was initially used in wind turbines, were applied to the marine plants shaft generator systems, with their wide range operational speed advantage (Y. Wu et al., 2018). Additionally MAN variable frequency drive technology also enables to use shaft generator connected while in reduced low operations with 50-60 Hz/ 400-440V for the grid with up to 15% fuel savings due to the type of ship and operation(MAN Diesel and Turbo, 2020b).

Auxiliary system improvements and reduced auxiliary power demand

Apart from main engine improvements and waste heat recovery applications, reducing the auxiliary systems' power demand is the most reasonable way to reduce energy consumption of the ship. The ship operational speed had reduced significantly in recent years. The operating profile of the main engines vary from 70% ME load to 40% ME load. Since the cooling system plans, their flow capacities were calculated according to design conditions of the ships and have capability to cope with tropical conditions at maximum load covering margins; the cooling system values satisfy the needs quite well in real operational conditions. The temperature of fresh water is controlled by by-pass line, which is parallel line to the heat exchangers in conventional central cooling system.

The electricity demand of the pumps is accounted for 70% of the ship electrical requirement (Chun-Lien Su et al., 2014). The seawater pumps' frequency controls had been carried out by Su et al. (2014). Kocak and Durmusoglu had also studied sea water pumps' conversion to variable speed based on the sea water temperature variation during ship routes (2018). The control of cooling pumps, which is carried out by evaluating the cooling requirements of components, studied by Elg et al. by using energy flow simulator, developed in Simscape, (M Elg et al., 2015) to support energy efficiency applications in the ships. Another study also carried out by Rahkola and Kinnunen (2017), for ice going vessels. The energy system modelling method had been utilized during, full-speed, half load, manoeuvring, conditions. Variable seawater temperatures were investigated in the study that only focuses on seawater pumps converting to frequency converters. However, limited amount of saving was declared in the study according to variable seawater temperatures. The central water cooling system of the ship comprises two main circuits, which are high temperature and low temperature fresh water circuits in addition to seawater line. Both sea water and fresh water systems can be considered coolant from the view main engine and other component.

To sum up state of art technologies, the energy efficiency practices are directly related to fuel saving and emission-abatement technologies for the marine diesel engines. At a first glance, the optimization of marine diesel engines' operation, via observing the parameters and controlling equipment, as performance monitoring is a prominent way to reduce both emission and consumption. Enhancement of combustion efficiency and quality via performance controlling, injection process, turbocharging solutions and valve train improvements have crucial methods for marine engine control. Recovering and utilizing the waste energy are the prominent ways, to be adopted the specific ship type and operation.

In order to maintain sustainable development of the maritime industry, additional studies have to be adopted to reduce fuel consumption, regardless from the amount of saving and equipment.

The new technological advantages have to be used with practical solutions, to propose applicable solutions as regard to the ship type and operation profile. The significant amount of the reduction strategies are focused on reduced loads of the main engines. Since the engines are designed to operate optimum range which is around 75% Load, during reduced loads efficiency degradations were tried to be overcame via this methods. Considering a significant reduction in operational speeds of the ships in recent years, the improvement solutions for the reduced loads are reasonable. The technologies applied for energy efficiency, are for reduced engine loads, mostly. Table 1.8 represents the mentioned technologies, their related field, and the operating condition, which is more convenient to subject technology.

Table 1.8: Technologies applied in literature within the scope of the thesis.

Design Loads ● Reduced Loads ■		Engine De-rating	Injection Techniques	Power Turbine	Engine Control Tuning	EGB-HTP	VTA	T/C Cutout	Exhaust Valve Control	Variable Speed Pumps	Power Take Off	Waste Heat Steam.	Organic Rankine Cyc.	Thermal Storage Syst.
Main Engine Improvements	●		✓	✓										
	■	✓	✓		✓	✓	✓	✓	✓					
Auxiliary System Improvement	●										✓			
	■									✓	✓			
Steam Plant Improvements	●											✓	✓	✓
	■											✓	✓	✓
Waste Heat Recovery	●											✓	✓	✓
	■											✓	✓	✓
Reduced Auxiliary Power Demand	●													
	■									✓				

1.4 Hypothesis

Energy savings predictions for a system by evaluating the operational conditions of ships and equipment, and estimating their performance under variable loads, shed light on the extent to which ships will be ready for the limitations by rules they will encounter in the future, and how much improvement can be made. The more precise estimations allow the more reliable improvement actions. This is the main idea of how modelling tools helps to the ship operations.

IMO or any other organization does not point out any kind of application to improve the efficiency in ships, specifically. The model-based approach facilitates the determination of the energy efficiency improvement method applied for ships and helps to find reliable solutions.

This study will create an understanding about proposed solutions and their response for main engine systems. The model enables operators to know how much the efficiency potential can be gained, Preliminary prediction of saving, and enables to calculate whether the investment is appropriate to ship type and related ship operation profile. The purpose is to achieve feasible, simple and possible efficiency improvement solutions for the equipment, already exist in the ship with the low investment cost without equipping new system as like ORC or thermal storage systems for subject ship.

In the proposed model based investigation for the energy efficiency in the ships, the new ideas, proposed, comprehend the main engine operation and the auxiliary systems in the operation of engine room. The approach assesses the effect of operating parameters to find cause and effect relationships with each other among components in the frame of all power system.

The Model based approach helps to decide potential solutions to enhance the energy efficiency, by considering the ease of application and low cost advantages with high reliable results and developed cost effective quick response. The simulation with high reliability can represent subjected efficiency application for a given ship in different main engine conditions.

The ship’s power plant has a complex system integration that include, as a mechanical power source main engine; for the electricity auxiliary generators, shaft generators, and turbo generators, additionally electricity consumers as like all pumps, compressors, air conditioning systems, fans etc. To observe the interaction of whole system and observe potential efficiency improvement while in the operation is very difficult and it put some workload to the operators, which is not recommended by MEPC. By modelling way, individual system or combined systems can be modelled and operational characteristics can be simulated in digital environment. Even small improvements can be seen by changing operating conditions, while observing improvement is quite difficult because of less accurate measurements.

In recent years, the efficiency applications considered main lodes; however, the pace of efficiency slowed down. In order to maintain efficiency trend, the measures should focus on main lodes to capillaries. All of the systems in the ship's engine room are systems that affect energy efficiency directly or indirectly. The evaluating of these systems separately or as a whole should be designed to reflect the behaviour of the system. The modelling should be carried out with measurable parameters and outputs. The contribution will change according to type, size and operation of the ship.

The Regulation 22 of MARPOL annex VI, which was explained in Regulation background section, will enter into force in the year 2023. The regulation issues CII that efficiency levels of the ships will be determined and the ships which can not achieve desired efficiency levels will have to update their SEEMP plans. The modelling studies will be an attractive solution to examine the potential solution results. The planning, implementation and measurement phases of the SEEMP results faster, and the possible problems could be detected in a short time, which allows measures will be taken before they are put into practice.

Within the context of model-based investigation, the hypothesis of the study is described as:

- In order to meet the future requirements in the shipping, there is a certain need of tool, having precise estimation capability for the potential saving in ships' power plant. The efficiency applications of the auxiliary systems is at least important as the energy efficiency of the main engine respect to the main engine operational profile. Any saving in electrical consumption of the auxiliary equipment results in saving in fuel consumption for ship too. The model-based investigation, which has precise estimation capability, comprises the subject ships' energy generators and consumers.

1.5 Organization of the Thesis

The challenge of the thesis is not only to identify the problems or potential solutions for the enhancement of the efficiency, but also to develop a model that is designed depending on the operational parameters on which the solution is based and ensures that the results can be verified.

The model-based approach, proposed in this thesis evaluates main engine and its components in the respect of energy efficiency, according to the real operational profile of ships, which is not in the range of the designed operational loads. The real operational profile is different than the optimally conducted energy efficiency studies.

In this respect, studies were conducted according to the gaps, determined as a result of a literature survey. The selected subjects are (a) the cooling need of the main engine according to operational load, (b) optimization of the seawater and freshwater pump load, which are the biggest pumps in ship power plant, and (c) optimization the compressed air cost during transits, by proposing a range of operational pressures, since it is the most expensive energy storage system.

The following sections seek to address how the articles' topics are selected, their scopes, goals, novelty and their contribution into the whole efficiency. The thesis study consists of three parts consisting of three articles in total. All the chapters comprise the subject equipment model, verified by reference data, conducts model based study, to enhance the efficiency of the system according to proposed method.

1.5.1 The first chapter

The article is "Effect analysis on energy efficiency enhancement of controlled cylinder liner temperatures in marine diesel engines with the model based approach.

The scope of the first article is the investigation of excessive cooling effect on marine diesel engine performance with respect to the operational profile of marine diesel engines. In this respect, the study addresses the find out calculating the heat energy amount dissipated by overcooling via jacket water cooling system.

Marine diesel engines have high power output with high operational efficiency. The output power comes with the large dimensions that in the modern type diesel engines, stroke/bore ratio has increased even higher in recent years. The increased thermal efficiency and the large dimensions bring some operational problems as a high thermal load for the components. The mean combustion temperature variation, between around top dead center (TDC) and around exhaust valve open (EVO) causes high variation in temperature pattern along with the liner, vertically. To cope with the safety problems in the all range of engine load, the cooling of the liners is adjusted for challenging conditions. The conventional cooling system for ME jacket water adjusts the inlet temperature of the HTFW to set the outlet temperature after ME. Stabilizes the engine outlet temperature at 85 °C to protect the engine from the excessive thermal load. But the cooling phenomena changes dramatically as a result of engine load. Even in the conditions of reduced loads, the cooling system keeps the same outlet temperature by increasing the inlet temperature to the main engine. Considering the flow regime is the same at the jacket water cooler side, because of fixed speed pumps supply the same pressure to the inlet line, it can be assumed that the liner temperatures reduces significantly. The relatively low liner temperatures compared to the optimised load of the ME, cause higher temperature difference between in-cylinder gases and cylinder liner that result in relatively higher heat loss to the cooling medium. In the marine diesel engines, the cooling need for the liners had not been examined for reduced load conditions. The majority of the studies in the literature, reducing cooling load, focused on small-scale internal combustion engines with gasoline or diesel fuelled with a fixed solutions. Hence, large-scale marine diesel engines need to be considered, to abstain from overcooling, with instrumentation availability advantage by controlling cylinder liner temperatures. The proposed method in this study with similar usage capability but offers fixed-permanent countermeasure to prevent cold corrosion in the liners for slow steaming operations in a MAN service letter (MAN Diesel and Turbo, 2012) had proposed re-designing cylinder liner to increase outlet temperature for the S80ME-C engines.

The reduced ME load conditions, based on different liner temperatures interaction with in-cylinder gases by adjusting the possible higher liner temperatures in all the ME load are examined in the study. The investigation is carried out by main engine model, comprising the liner cooling, in the range of operational profile of the ship.

The marine diesel engine model results had been validated by reference data of the engine, which is gathered by digital twin of container ship main engine, engine room simulator (Kongsberg Maritime, 2019). The engine room simulator is the digital twin of the ship power plant with main engine operations and other auxiliary systems. The data, given by manufacturers is taken into account with the tolerances. According to the differences between reference data and predicted data an Root Mean Square (RMS) error analysis had been carried out and the variation is given in Table 1.9.

Table 1.9: RMS Error Analysis of predicted data of ME model.

	100%	85%	78%	65%	55%	45%	RMSE
RPM	-0,01	-0,02	-0,06	0,02	0,01	0	0,026
Max.P.	-1,51	1,125	2,75	2,36	1,99	0,8	1,743
Scav.P.	0,03	-0,008	-0,024	-0,068	-0,09	-0,109	0,061
Exh. T.	18,12	13,82	11,42	8,84	10,92	17,31	12,799
Exh. P	0,0263	-0,016	-0,017	-0,057	-0,07	-0,09	0,050
Cool.Q	31,6	-12,14	-17,94	-14,347	10,32	39,75	21,910
Bsfc	0,19	-1,14	-1,61	-4,32	-4,43	-2,20	2,593
T/C.P.	238,26	90,22	44,39	-112,56	-160,18	-167,05	137,898

The variation up to 5% can be expected in brake specific fuel oil consumption for the given generated power and pressures while 15℃ variation can be observed in exhaust manifold temperature, additionally the engine loads, lesser than 35%, greater amount of variation can also be observed (Guan et al., 2014). The variation in main operating variables affect the other variables as Turbocharger power and cooling output. The variations, essipecially T/C power and their RMS errors are higher because more than 5% error can be observed at reduced loads as indicated by manufacturer. By considering the variation rate the engine model comparison was carried out for 45% ME load and operational investigation is performed higher than this range.

The mean value engine model is used to simulate the engine characteristics combined with zero dimensional combustion model to eliminate lack of detailed data availability for MVEM about in-cylinder processes. With the combination of zero-dimensional model, the pressure and temperatures could be estimated with higher accuracy. The capability of prediction of the data is important since the study includes cooling loses from cylinder walls. Then the effect of the cylinder wall temperatures on engine is represented. Zero dimensional model provided a good opportunity to estimate the effect of cooling at reduced loads. By applying the modelling techniques together, the mean value of the temperature and the pressure of the in cylinder gases as regard to the crank angle are calculated with the ideal gas law.

The study combines operational practices with environmental concern. There is no governor system model and since transient operation is not investigated, transient response of the engine is not represented in the model. Additionally, the WHR system is equipped to recover waste heat energy from ME exhaust gas flow but, shaft generator, power turbine or organic rankine cycle (ORC) systems are not equipped in the model of power plant. The system comprises the equipment of the ship in case study, no additional component is modelled or is proposed for waste heat recovery increment.

With the help of the developed engine model, a notable understanding could be gained into the effects of the how the operational improvements affect energy requirement and efficiency of the system. A case study is carried out for cylinder jacket water cooling system. The cylinder jacket water cooling “high temperature cooling” extracts heat to keep the liners in safe operable temperature levels. But it must be investigated that the whether it provides both safety and energy efficiency at the same level for each ME load, when it operates at realistic operating conditions which are in accordance with the recent years’ operating trends. The operational profiles of the ships had been changed significantly as slow steaming.

The scope of this study is not to evaluate wasted heat to the fresh water system, that is determining the overcooled heat amount while in fixed jacket water-cooling carried out. The effect of the non-extracted heat distribution on engine output power, T/C, recoverable waste heat and to the environment, which are calculated with model-based approach. The study increases the understanding the effect of liner temperatures on energy efficiency of the diesel engines. Investigation of the exhaust manifold pressure, exhaust manifold temperature and scavenge pressure variation for the loads that under 85% ME load. Because of the lack of detailed information about amount of saving potential or estimation incapability, the modelling studies are required to supply decision-making facility, and to ease of calculating the improvement for the energy efficiency contributions.

1.5.2 The second chapter

The article is “Load optimisation of central cooling system pumps of a container ship for the slow steaming conditions to enhance the energy efficiency”.

The scope of the second chapter is investigation of minimization of auxiliary power demand, the electricity consumption of the pumps. After the optimisation potential of hull-speed and main engine operations, the auxiliary machinery for cooling has the next largest energy saving potential in commercial ships. The pumps’ electric energy consumption accounts for 70% of the ship electrical consumption (Chun-Lien Su et al., 2014). The cooling water pumps for seawater and fresh waterside are the largest electrical energy consumers in the engine room with their continuous operation. The cooling pumps are the vital equipment for a marine diesel engine. They meet the requirement for cooling or heat extraction from any equipment, which need to cools down in the engine room.

The central cooling system of the ship is designed to meet the maximum expected cooling requirement with the 110% conditions of ME. However, the main engine of the ship does not even perform at its designed operational range in recent years. Considering the cooling requirement varies due to the operating conditions, for instance, the operational load of the ME, the cooling system uses an excessive flow of coolant liquid at the operational at the reduced load of operational conditions, which means not economical process most of their operational lifetime. This over pumping causes usage of the excessive power of ship electricity. There is an emerging requirement to investigate the dynamic thermal management of low temperature fresh water (LTFW) system. Speed control of pumps is a reasonable solution that requires control of input speed by frequency.

In the conventional system design, the temperature of the fresh water is controlled by combination of by-pass system and heat exchanger in parallel. Low temperature circulating pumps, seawater pumps and the heat exchange phenomenon are the key elements for the system. The fresh water system set-up is a ring network that comprises low temperature and high temperature fresh water systems together with heat exchanger to provide fixed temperature to low temperature fresh water line. The by-pass system's flow rate increases when the cooling demand reduces for the system components. The excessive flow rate in by-pass line is an indicator of over pumping costs. However, the temperature can be also controlled by flow rates of both fresh water and sea water, based on the operational profile.

The analysis of central cooling water system was performed based on the operational load of the main engine. The fresh water system cooling requirement varies significantly due to the load condition of main engine. The application is done for both sea water and fresh water systems pumps. In this study, both of the coolants are considered operating with varying flow rate with the help of the frequency converter pump controlling system, where the approach, in the literature is for only seawater pumps. The seawater temperature is assumed constant which is not constant in real operations but, since the novelty of the research is the main engine operational perspective that is investigated in the research the seawater temperature is assumed as constant temperature, 20°C. The fresh water systems temperature varies as a function of ME load. Because, the cooling of cylinder liners, scavenge air and lubricating oil is carried out by fresh water cooling system. These cooling requirements are totally based on operational load of main engine, which are represented in Figure 1.13.

The computational effort had been carried out for the estimating the potential saving in a container ship central cooling system together with seawater and fresh water interaction via modelling of cooler and pumping units. This can be accomplished by minimizing the flow of fresh water through by-pass line with the help of the frequency converters.

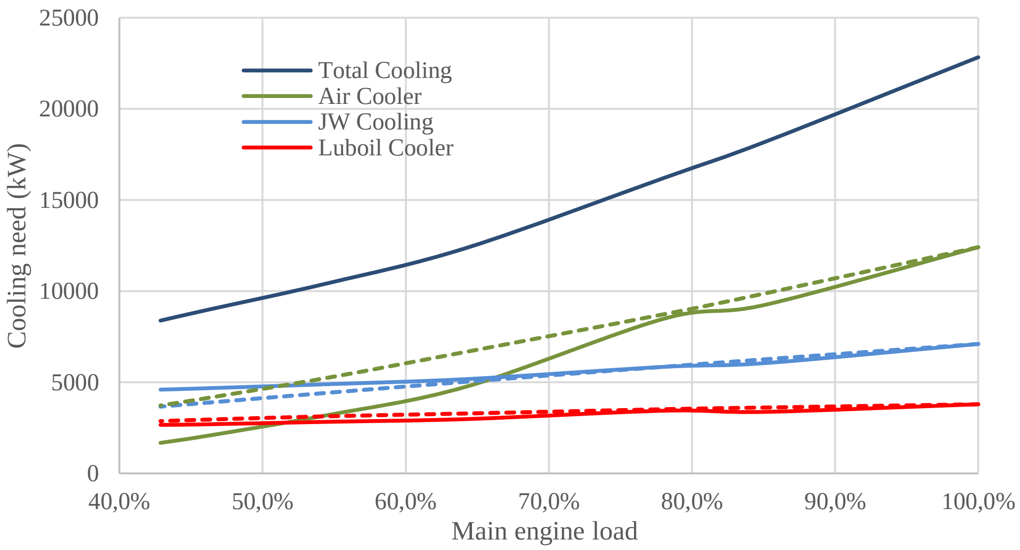

Figure 1.13: The heat dissipation variation as a function of main engine load.

The calculation methods about heat dissipation, given in the engine manufacturers project guide, are used to calculation of cooling requirement of the main engine and connected cooling systems. In addition to the equations, the main engine model also gives the heat dissipation to the jacket water cooling system and with the help of turbocharger sub-model the compressor side outlet temperature and the pressure is used to calculate the cooling requirement of the scavenge air cooler with respect to the engine load. The cooling demand of the components were represented in Figure 1.13. Since the piston-cooling model is not included in the engine model, the cooling requirement variation, which is not also dramatic, for the oil cooling cannot be calculated with the help of the engine model, developed in the article one. However, if it is desired to use the developed model in the calculations, the proportional calculation approach can be used according to the engine load in the calculation of the oil cooler. In the study, the cooling requirements are estimated by comparing of the project guide's formulas algebraic calculation and the reference data used in the validation of the developed main engine model. Three main cooling medium are taken into the account namely, jacket water, scavenge air cooler, oil cooler.

While in the modelling processes, some limitations were put into the model. The limitations are determined according to design of experiments in the literature (M Elg et al., 2015). The temperature of outlet fresh water was adjusted to be 2.5 ℃ above sea water inlet temperature as a parameter of heat exchanger. The efficiency of the pumps modelled by the formula proposed by Sarbu and Borza (Sarbu and Borza, 1998) as regard to the speed for the respective load for the both sea water and fresh water pumps. The minimum flow rate in the pumps were adjusted that the flow rates can be reduced down to 50% of normal load. As another limitation, optimization has been made so that the sea water temperatures after the cooler do not exceed 46℃ and the fresh water temperature entering the cooler does not exceed 50℃. The model aims at predicting the required pump speed by introducing independent variables, as sea water inlet temperature, sea water flow/fresh water flow and fresh water inlet temperatures. The variation of any independent variable as an input parameter for the model, results in change of the output of the model, which is the output temperature of seawater and optimized flow rate of by-pass line where the minimum 10% of fresh water passes through as another limitation. The all limitations are represented in Figure 1.14 in the scheme below.

The purpose of the control mechanism is minimization of by-pass flow rate in order to lessen excessive pumping power. The heat exchange rate in the cooler is controlled by temperature sensor after the coolers, which operate with seawater and fresh water. The design of system has to be converted to flexible operation capability instead of fixed design of extreme conditions. Assumption and mathematical relationships for the heat transfer in cooler between seawater and fresh water system have been adopted.

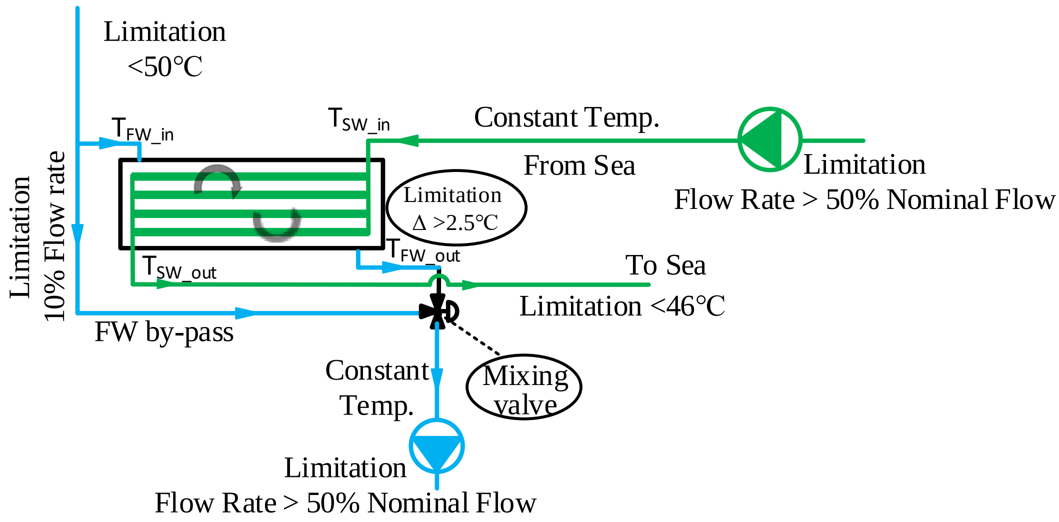

Figure 1.14: Central cooling system modelling control limitations.

Temperatures of water flows at the inlet and the outlet of the cooler for the fresh water side and sea water side is calculated in the study. Additionally, the study includes the mass flows for the pumps (seawater and fresh water) their energy consumptions with respect to affinity laws for the pumps with the efficiency variations calculations and the saving rates are calculated in the study.

Designing the cooling system for the ship power plant must be carried out considering not only safety issues but also, the efficiency issues have to be taken into account while designing and implementation of the piping and equipment. At the design phase, piping is a crucial parameter that, the dimensioning of the system, pressure loses affect efficiency level significantly. However, according to operational profile such as, operation at slow steaming or ballast condition, cooling need of the main engine and the components varies significantly. The cooling pumps are the biggest electricity consumer units that operates full time during operation. The system has been optimized to cut down electrical energy demand of auxiliary systems. Reduced power consumption for real operational conditions instead of challenging conditions, performed in design phase of the ship. The proposed solution can effectively support the energy efficiency enhancement of ship auxiliary energy consumption, also must be considered as a prominent solution at design phase in order to meet the energy efficiency limits of the ships.

1.5.3 The third chapter

The article is “Energy efficiency based operation of compressed air system on ships to reduce fuel consumption and CO_2 emission”

The scope of the third article is the energy consumption minimization of the most expensive energy source, the compressed air system. Although there are some studies carried out for compressed air system in land industries, to the best knowledge of the authors, no discussion on the energy saving potential of the compressed air systems used in the ships. The study was conducted to sustain the energy efficiency enhancement methods for the ships with the proposed compressed air operation. Not enough attention has been paid on the use of compressed air in an energy efficient way, neglecting that the behaviour of the operators has a high degree of influence on energy efficiency.

Since the air is cooled down after the compression process, the significant part of given energy is re-extracted. The great part of given energy during compression process is converted to the heat energy because of the intrinsic process of compression. In order to store higher amount of air in air bottles, specific volume of the air must be decreased via cooling which also prevents pressure drop while in storage because of heat loss. The heat energy extraction explains why the compressed air is the most expensive energy source in the ship.

The compressed air in the ship is in the process for many operations to keep main engine functionality. The first-start of the diesel engines is triggered by compressed air in air bottles. Moreover, the exhaust valve controlling is carried out by compressed air with air spring, no matter the cam controlled or electronically hydraulic controlled exhaust valve system, the closing mechanism is connected to compressed air system function in marine diesel engines. Additionally, for the fuel filters’ self-cleaning mechanism also operates with compressed air system. High pressure and low pressure control airs, safety control airs and other compressed air operations are connected to compressed air supply line. On the other hand, NO_x emission reduction technology uses compressed air namely, Selective Catalytic Reduction (SCR). The urea is injected in to exhaust gas duct with the help of the compressed air. The consumption of compressed air in emission reduction is not only limited for urea injection, but also soot blowing operation and cooling of blowing valves are also carried out by compressed air (Wärtsilä Environmental Technologies, 2017). The Soot blowing operation, instead of using steam in some cases, compressed air is better to use for soot blowing of exhaust gas boilers (MAN Diesel and Turbo, 2013c). The operation of SCR necessitates a smooth pressure supply of air. The air compressors must always run when the SCR system is operating, to prevent fluctuation air pressure, which causes suspend the SCR operation or soot blowing. The cessation of the operation results in clogging of catalyst component or damage, even for a short period of lack of air supply causes "disabled injection of urea". The capacity of the air bottles, operation of supplying compressed air to the storage (air bottles) should be well determined while proposing efficiency measures.

In the compressed air application in industrial plants, there are some the efficiency increment approaches as using efficient electric motors (Kaya et al., 2002), variable speed pumps (Neale and Kamp, 2009), or automatic control of system (Mousavi et al., 2014). Additionally, reducing the inlet air temperature, no excessive supply to the line, properly dimensioned units, using the lowest possible pressure in the system, recovering the heat energy from compressed air, reduction of air leakages, piping design, and monitoring usage of air are the other measures, taken to increase the efficiency of the operation (Abdelaziz et al., 2011; Saidur et al., 2010). All these methods focussed on reducing operational costs of compressed air in industrial areas.

The compressed air system has two main part as supply side and demand side. In the light of end user diversity of compressed air in ships, improving the efficiency focuses on the production side in the study. The usage of compressed air, after storage, is not considered. By analysing the methods used in industry, it is clear that some of the measures are not convenient for application in the ship neither the inlet temperature control is not an effective solution in engine room conditions or harvesting the heat from intermittent compressor operation. The monitoring, leakage analysis and reduction of operating pressures are seem prominent solution for the efficient operation. In order to decrease the energy demand, an investigation is carried out for compressed air system. The data obtained from different size and type of ships. The selection of case study's ship had been done according to efficiency potential and identification of the saving measures. The case study was conducted with selected ships air compressor equipment with their operational data, which is the usage of compressed air amount during operations, and the working schedule of compressors, and it was identified for the efficient operation of compressed air system. The data was gathered by considering energy consumption of compressor and pressure tracking in the bottle as a function of time. Additionally, until the restart of compressor, pressure decrease tracking as a function of time due to the compressed air usage by the systems. By using the volume of the air bottles, and pressure trace as respect to the time, the compressor performance was calculated and the model of the compressor and consumption data was embedded in the model.

As a system dynamics, the efficiency of the compressor, the leakage rate, and air standard compression process is simulated in the model. The air leakage is assumed around 2.5% of air usage where the up to 10% is acceptable in the industrial plants. Additionally, the efficiency of the compressor seemed varies as a function of out-put pressure. Because of both the efficiency variation and intrinsic process of air compression, the cost of per kg of compressed air varies significantly. The maximum pressure for compressed air bottles are 30 bars in ships while the minimum pressure to start the engine is declared as 9 bars in the DNVGL (Germanischer Lloyd) (Lloyd Germanischer, 2015). The limitation of the system is determined by considering class-societies requirement. In the study, different pressure ranges were applied form 30 bars to 14 bars for compressed air system. The cost of per kg air is calculated, as an output of the model. Additionally, the leakage rate is another significant parameter of the system that, 2.5 % to 45% variation of air leakage is simulated and effect on the energy consumption was calculated in the study. To represent the model outputs, graphical representation were applied in different pressure ranges, air cost, air consumption, artificial demand, power consumption of compressor, reduction rate of air consumption as a function of pressure range are given in the study. The output will be a range of observable energy efficiency recommendations based on operators' preference.

The reduction in power with hourly and daily-based results are calculated to make it easy to understand effect on energy efficiency for a ship. Because the electric power tables includes the electric consumption and operating time data of all equipment that used in SEEMP. The effect on power tables, energy consumption data of the equipment reduced significantly, that showed by the model. The energy saving potential and how much the energy saving contribution of the compressed air system for the ship is calculated in this chapter. The model demonstrates the cost of the compressed air production, electrical consumption of air compressor at different pressure ranges and demand of compressed air decrement. In a case study, the total saving in power need for compressed air system could reach 50% of nominal conditions.

1.5.4 The interaction of the studies in the general layout

Ships power plants include complex integration of systems that contains different physical energy domains with various sub-systems make understanding of energy efficient operation in ships difficult. The articles of the thesis investigate different energy domains of the ship's power plant with their interactions to each other under the energy efficiency manner.

In order to represent the transformation of energy to the other forms (domains), the conversion of the physical domain variables to the other ones is carried out with considering inefficiencies of the components via models. For instance, electric motors transform energy from electrical to mechanical domain afterwards, to hydraulic or gas domain in pumps or air compressors, respectively. On the other hand, the internal combustion engines transform energy from chemical domain to thermal and gas then mechanical domain, partially. By considering heat flow to the cylinder walls in thermal domain and exhaust flow with mass and heat variables come up as efficiency rate of internal combustion engine. Transformation of energy with a model-based approach should contain the following domains and variables in ship power plant. The power plant of a ship is delineated in Figure 1.15 where the domains are indicated with different colour lines.

The arrows vertical and horizontal shows the flow of the energy with different domains. Additionally, the numbers 1, 2, 3 numbers are the cases, studied in the articles, respectively. The multiple arrows from the article indicators to other components show the interaction of the system with each other which means a system improvement affects the other systems.

The modelling should meet the requierement the exchange of variables with an acceptible accuracy. The improvements in cooling of main engine or hyraulic power of pumps or power of compressor cause saving in electric and at the end of the process the saving is converted to fuel saving and equivalent CO_2 emission reduction. The domains include in the models are listed in Table 1.10.

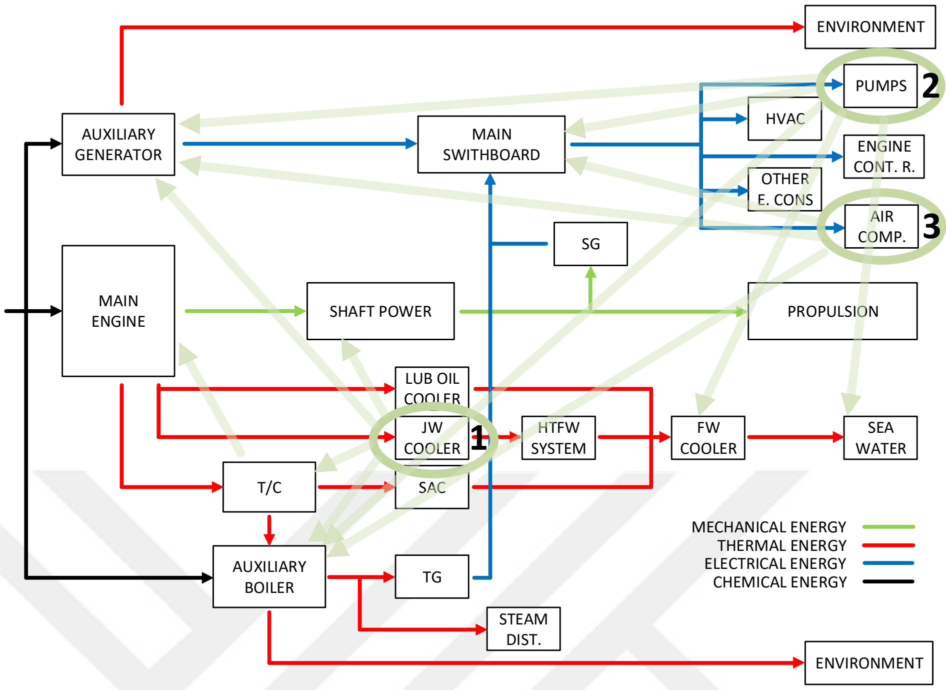

Figure 1.15: A simplified marine power plant for a ship having conventional propulsion.

For the engine fuel consumption, lumped assumptions are made to facilitate transition physical energy systems' results to the chemical domain in order to calculation of fuel saving in boiler and auxiliary generator systems. To calculate the output of fuel saving, the specific fuel oil consumptions are used in boiler and in auxiliary engine regards to the load to represent the efficiency.

The energy systems of the ships are dynamic systems, converting the fuel energy to mechanic, heat and electrical energy. The systems are based on chemical, thermal, hydrodynamic, mechanic, and electrical processes that, the processes are represented by different sub-systems composed of different domain models. The data merge of these sub-systems provides to capture the combination of whole system frame in an energy efficiency perspective. It cannot be really understood that the underlying dynamics of ship power plant energy performance without considering sub-system level based evaluation method.

Table 1.10: The domains of the modelled components.

Physical Domain	Domain Variables: Flow Variable	Domain Variables: Effort Variable	Studied Model	
Mechanical	Torque	Angular Velocity	ME Model	Article 1
Gas	Temperature Pressure	Heat Flowrate Mass Flowrate	ME Model	Article 1
Thermal	Temperature Pressure	Heat Flowrate Mass Flowrate	ME Model Hear Ex. Model	Article 1 Article 2
Hydraulic	Pressure	Volume flowrate	Variab. Speed P.	Article 2
Two-phase fluid	Specific enthalpy Pressure	Heat Flowrate Mass Flowrate	Turbo Gen EGB Model ORC	
Electric	Current	Voltage	Air Compressor	Article 3
Chemical	Molar Flowrate	Chemical potential	Engine Consump.	A 1-2-3

The articles fill the gap in the literature that given in the Table 1.8 before, The following Table 1.11 shows the studies' contribution to the methods given in the literature, while the studies contribute in normal operations the significant advantage provided at reduced loads, particularly. The components are considered at a system-level, to make the calculation processes easy.

Table 1.11: The technologies addition to the literature, studied in the thesis.

Design Loads ● Reduced Loads ■		Engine De-rating	Injection Techniques	Power Turbine	Engine Control Tuning	EGB-HTP	VTA	T/C Cutout	Exhaust Valve Control	Liner Temperature	Variable Speed Pumps	Comp. Air Sys.	Power Take Off	Waste Heat Steam.	Organic Rankine Cyc.	Thermal Storage Syst.
Main Engine Improvements	●		✓	✓												
	■	✓	✓		✓	✓	✓	✓	✓	✓						
Auxiliary System Improvement	●												✓			
	■										✓	✓	✓			
Steam Plant Improvements	●													✓	✓	✓
	■									✓				✓	✓	✓
Waste Heat Recovery	●													✓	✓	✓
	■									✓				✓	✓	✓
Reduced Auxiliary Power Demand	●															
	■										✓	✓				

(1)The main engine, jacket water cooling requirement, analysed and when the excessive cooling is eliminated during low load, the efficiency of the main engine, and tha capability of T/C and waste heat increased. The energy benchmarking is carried out with the same amount of fuel consumptions and the benchmarking can project the point that has energy efficiency potentials in main engine and waste heat recovery capability.

(2) The application of variable speed pumps is carried out by both sea water (as in literature) and low temperature fresh water pumps. The results are compared to the designed fixed speed pumps performances at real operating profiles of ships in recent years, indicates that the variable flow pumps achieved significant amount of energy saving at low loads.

(3)The data, the usage of compressed air, pressure reduction trend as a function of time in air bottles is implemented into new pressure range operations to determine electric energy consumption of the system. Running the simulation in different range of pressures demonstrate the daily energy use of the system. The reduced power demand is converted to auxiliary engines equivalence fuel consumption.

The applications can effect both design phase, and operational measures, the studies are explained in the following sections with the given order in thesis plan.

2. EFFECT ANALYSIS ON ENERGY EFFICIENCY ENHANCEMENT OF CONTROLLED CYLINDER LINER TEMPERATURES IN MARINE DIESEL ENGINES WITH MODEL BASED APPROACH[1]

2.1 Introduction

Two stroke large bore diesel engines are widely used by merchant fleet in maritime sector because of their high power output range and high efficiency. Although the seaborne trade has high efficiency, the total fuel consumption and emission outputs are important in this field because more than 80% of the world trade is carried out by seaways (UNCTAD, 2018). According to the last (third) IMO (International Maritime Organisation) GHG (Green House Gases) study (IMO, 2014) the 2.2% of total anthropogenic CO_2 emissions are derived from maritime sector and it is expected that the emissions have potential to increase between 50% and 250% by the year 2050 as a result of volume of the world trade. For this reason IMO put the regulations into force, in the frame of MARPOL "(International Convention for the Prevention of Pollution from Ships), both to enhance the energy efficiency and to mitigate the CO_2 emissions, resulted from the shipping sector. To this end, EEDI (Energy Efficiency Design Index) (IMO, 2012a), aims a continuous growth for the energy efficiency as, 10% improvement for each 5 years step. On the other hand, EEOI (Energy Efficiency Operational Indicator) (IMO, 2009) which is accompanied by SEEMP (Ship Energy Efficiency Management Plan) (IMO, 2011) is another important tool to observe effect of any change made in the operation on operational energy efficiency. MEPC (Marine Environment Protection Committee), encouraging the implementation of efficiency measures, has advanced the date of entry into force of phase 3 from 2025 to 2022 for some ship types as, containerships, general cargo ships and LNG ships. In particular, efficiency design index reduction is specified between 30% and 50% of the date from 2022, for the containerships between the size from 15,000 DWT to 200,000 DWT,

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[1] Dere C., Deniz C. 2020. Effect analysis on energy efficiency enhancement of controlled cylinder liner temperatures in marine diesel engines with model based approach, Energy Conversion and Management, Volume 220, 2020, 113015, https://doi.org/10.1016/j.enconman.2020.113015

respectively (IMO, 2019c). The containerized seaborne trade is the fastest growth trend in commercial fleet in recent years (UNCTAD, 2018) therefore, the date of the amendment is important for container shipping.

The CO_2 emission is an unavoidable combustion product in marine diesel engines which use carbon-containing conventional marine fuels as Heavy Fuel Oil (HFO) and Marine Diesel Oil (MDO). The amount of the emissions can be reduced by less consumption of fuel which can be achieved with more efficient ship designs (Talay et al., 2014), monitoring and maintenance (Bertram and Taşdemir, 2017), more efficient operations (Dere and Deniz, 2019a; Durmusoglu et al., 2009; Yang and Yeh, 2014), or with the using of alternative fuels (Deniz and Zincir, 2016; Zincir et al., 2019), both biodiesels (Manaf et al., 2019; Z. Zhang et al., 2018) and with their water emulsions (Jiaqiang et al., 2018; Z. Zhang et al., 2019) and also with fuel cell applications (Inal and Deniz, 2020). The CO_2 emission mitigation is the one of the main challenges in recent internal combustion engine studies (Cipollone et al., 2013). In the shipping industry, to cope with the issue, the efficiency level of the propulsion components must be analysed and energy loss nodes which have an improvement potential, must be well determined. The merchant ships are not operating at their design speed (slow steaming) in recent years (IMO, 2014). To meet the requirements of emission challenges, their voyage speed had been reduced. However the design of the propulsion system is optimised for design speed for instance, output power, propeller, turbocharger, cooling system are selected for design conditions. Therefore, the efficiency measures must be re-evaluated in such kind of the operating conditions which are different from designated one. One of the critical points that have to be observed, is the cylinder wall temperature and its efficiency interaction with the main engine. Since the average combustion temperatures of diesel engines are far higher than the material limitations, made of, engine components as piston, liner and cover must be cooled (Watson et al., 1983). Steel materials, used in diesel engines, have temperature limits (Kamo and Bryzik, 1983) and are not capable to operate for a long time under challenging conditions. The cooling and heat transfer phenomenon of the internal combustion engines have quite important effects on engine efficiency and in-cylinder products (Neshat and Saray, 2014). In order to enhance the efficiency of the engines, cooling phenomenon was investigated with the help of several methods. To increase the efficiency of the internal combustion engines, researches, heat transfer rate in low heat rejection (LHR) engines (Jaichandar and Tamilporai, 2003; Reddy, Domingo, and Graves, 1990) from the combustion gases to the cylinder walls and piston surface, which are also called adiabatic engine (Kamo, 1987) have been conducted. These studies aim to reduce heat loses to other mediums (Gosai and Nagarsheth, 2014) by considering their performance not only for fully insulated engine (Garud et al., 2017; Krishnamani et al., 2018) but also with partially insulation layers as cylinder wall, cylinder cover or piston surface (Taibani et al.,) (Uchihara et al., 2018), in the land vehicle diesel engines particularly. An insulation layer is provided by the method which is a thermal barrier coating. In the case of excessive heat transfer, the in-cylinder temperatures decreases as well as pressure; hence the work done by the gases decreases (Liu and Reitz, 1998). A great part of available work potential of transferred heat from working fluid to the cylinder walls is destroyed therefore, low heat rejection strategy aims to minimize heat loses from in-cylinder (Giakoumis, 2007). The investigation of low heat rejection engines had been conducted with both mathematical model (Poubeau et al., 2019; Yao et al., 2018) and experimental ways (Taymaz, 2006). As a result of LHR studies insulation layer leads the increase at the cylinder wall temperature and reduces the volumetric efficiency, but it is resulted with high exhaust temperature that generates higher boost pressure which can overcome with this problem to obtain increased engine brake power efficiency. However, the insulation modification is un-removable (fixed) application for all engine loads. Marine diesel engine operational parameters are adjusted for the optimum load of the main engine and optimum speed of the ship. Moreover, an anatomic modification inside the cylinder causes an inadmissible operation of main engines of the ship at full loads or in challenging conditions. Nonetheless, it is obvious that the improvement is required to increase operational efficiency when operating ships at lower loads. There are other methods to achieve low heat rejection strategy in internal combustion engines. Modification of the cooling system and controlling the cooling phenomenon is an alternative solution for optimising the cooling rate, typically temperature control, pressure control and cooling location control. An experimental study has been done by Walentynowicz (Walentynowicz, 2011), less heat transfer could be achieved from cylinder walls, by enabling the higher circulation temperature of the cooling fluid in four stroke 66 kW diesel engine. The higher coolant temperatures, up to 20℃, could be reached by increasing the circulation pressure of coolant fluid. Increased torque and decreased specific fuel oil consumption were observed and in addition to fuel consumption decrement, carbon monoxide and hydrocarbon emissions could be decreased with a significant level of as a result of the study. Rakopoulos et al. (Rakopoulos et al., 2004) investigated the effect of cylinder liner temperatures on combustion parameters, boost pressure, engine speed and its response for transient operations for variable liner temperatures via both experimental and model based approach for diesel engine. The liner temperatures are 400 Kelvin, time depended 400 to 500 Kelvin, varying with the load and for the limited cooling (adiabatic) case at which is 600 Kelvin. The study points out that there are favourable effects of limited cooling for the low load operations. In another study, (Rakopoulos and Giakoumis, 2005) the authors investigated the cylinder wall temperature effects in a second law analysis manner for transient load operations of diesel engine. Gharehghani et al. conducted a study (Gharehghani et al., 2013), with a turbocharged SI engine, fuelled with natural gas, was investigated under 90℃ and 110℃ cooling fluid temperatures. The study showed that the thermal efficiency increased 0.3% averagely, heat transfer to the cooling fluid decreased 2.5% and the energy in exhaust stream increased 3% with different (higher) cooling fluid temperatures. A noteworthy fuel consumption and CO_2 emission saving was achieved by Cipollone et al. (Cipollone et al., 2013) with the thermal management of engine cooling system by changing the cooling fluid temperatures, which had been carried out by modelling both the engine and the engine cooling system for a passenger car. Another experimental study was carried out by Penny et al. (Penny and Jacobs, 2017) that the low heat rejection strategy was achieved by increasing the coolant temperature from 56.5 ℃ to 100℃. To prevent to risk of boiling of the cooling water, ethylene-glycol/water mixture had been used. The heat transfer rate could be decreased with lower temperature difference with the mixture and wall. Although high coolant temperatures are advantageous in terms of efficiency and emission, they can cause serious problems when exposed to high operating temperatures without modification of the engine components. In addition to low heat rejection strategies which are carried out by applying thermal boundary coatings and changing the cooling regime the cooling optimization studies with also had been carried out by researchers with computational fluid dynamics methods Li and Chen (Y. C. Li et al., 2007) performed a simulation study for different environment temperatures and variable cooling flow rates to optimize the cooling phenomenon. Jafarabadi et al. (2013) performed a thermal analysis to predict the heat transfer rate and the wall temperatures. The study indicates that the optimum cooling reduces the thermal stress and improve cylinder wall temperatures on diesel engines. In another research, (Romanov and Khozeniuk, 2016) Romanov and Khozeniuk conducted a study to observe the heat flows to the coolant fluid by comparing the different coolant inlet locations. The study shows that a heat loss improvement can be achieved by adjusting the cooling inlet location. Gholinia et al. (Gholinia et al., 2018) investigated the effects of cooling fluid temperature, speed and pressure for jacket water cooling system. The heat transfer models and their comparison are investigated in a research for in-cylinder heat transfer model (Žák, Emrich et al., 2017). Another 3-D cooling model was proposed with a 1-D engine model to investigate thermal characteristics of engine in different cooling conditions for control concept (J. Zhang et al., 2018). Since the experimental studies are time consuming and costly, the numerical model of engines, with a high capability for the prediction of the engine parameters, is required to overcome these limitations.

Modelling of diesel engine has been a viable solution to carry out energy efficiency analysis under different operating conditions. Investigation of the effects of different operating parameters with a verified model is both time and cost effective solution instead of on board measurement applications particularly (Sigurdsson et al., 2014). Simulation of the engine operating characteristics, as generated power, in-cylinder gas properties, scavenge and exhaust pressures and temperatures with their coupled systems typically, turbocharger and cooling phenomenon may help to cope with the enhancement of efficiency for marine diesel engines. Diesel engine models have been proposed by many researchers to improve engines (Hendricks, 1989; Merker et al., 2006), with their heat transfer (Assanis and Heywood, 1986). Moreover, investigation of the large bore marine diesel engines have been carried out by mean value model (Baldi et al., 2015; Murphy et al., 2015), and zero dimensional model (Payri et al., 2011; Scappin et al., 2012), to optimize operational settings (Mavrelos and Theotokatos, 2018), to ascertain of diesel engines performance (Kharroubi and Sogut, 2019), and not only to investigate any available improvement, but also for diagnosis applications (Hountalas, 2000; Lamaris and Hountalas, 2010). Operation of a diesel engine is a quite sophisticated phenomenon that the steady state engine modelling approach is utilized to overcome the complexity. A lumped model is used to observe the effects of liner temperatures on the engine efficiency.

The cooling requirement of the marine diesel engine differs significantly with the variable main engine loads (Dere and Deniz, 2019b). Additionally, the percentage of heat losses to the exhaust gases or to the cooling fluid cannot remain at the same ratio which like in the optimum designed speed of the marine diesel engine. In other words, at the operation of low engine loads, the percentage of loses such as, friction and heat loses, increases (Gharehghani et al., 2013). Therefore, the specific fuel consumption (SFOC) which is an important parameter to evaluate the engine performance increases when operated below the optimum speeds. Since the heat loss to the cooling fluid comprises an important percentage of input energy, controlling the heat flux might be an advantage to increase useful work (Heywood, 1988). On the other hand, there is no direct way to control the cooling for marine diesel engine as a function of engine load. The cooling is only controlled by the outlet temperature of the jacket water cooling system which is mostly keeping at 85°C in modern marine diesel engines (Kat, 2011; MAN Diesel and Turbo, 2010). Cooling engine parts and keeping them in an optimum temperature range is an important part of engine operation. The power output of the engine is affected by the cooling rate, if the surrounding border of combustion products are cooled more than required, the less power is received from the shaft which affects total efficiency of the engine (Karamangil et al., 2006). The first law of thermodynamics states that all the input energy is converted to the heat energy, mechanical energy or used by another system. During the engine operations, the heat energy, in the waste heat and exhaust gas is very high in higher engine loads. Differently, in the low engine loads, the jacket cooling water heat energy gets relatively more important (Fu et al., 2013).

By considering the, reduced speed operations of marine diesel engines in recent years (MAN Diesel and Turbo, 2014; Wiesmann, 2010), the marine diesel engines are over-cooled in reduced loads. For the reason that the cooling control of the marine diesel engine is being conducted by only considering the jacket water outlet temperature, there is a need to implement a new technique which can be operable both in high and low loads for all life cycle of the engine. The liner temperature pattern shows a significant variation between full ahead operation and reduced load operations. Since the marine diesel engines are operating at the steady-state conditions for long time, they allow temperature control of the liners with their long stroke length with the help of the using water-ethylene-glycol mixture as a coolant. The cylinder liner temperatures with three-point (the cover, upper and lower part above the scavenge receiver) were observed and used to calculate the heat transfer. There is no such control application considering liner temperatures in marine diesel engines and the studies in the literature about modification of cooling regime or implementation of insulation layer has been carried out for small size engines. There is a need to study for large bore two-stroke marine diesel engines with high potential efficiency increment by taking into account variable engine loads. The aim of this study is to investigate the energy balance of low-speed marine diesel engines at lower engine loads with a fixed liner temperature as like in full load. The thermal loading is assumed to be withstood by the engine, by keeping the temperatures in the operational range, and without paying any attention for the examination of thermal stresses. By this study, the understanding of cylinder wall temperature effects on a marine diesel engine energy distribution could be improved. The controlled cooling will help to reduce over-cooled loses to the jacket water cooling system. The engine modelling techniques are used to represent the thermodynamic balance of the engine. The operational parameters, as output power, specific fuel oil consumption, heat transfer characteristics, the recovered heat by the turbocharger, heat energy variation in exhaust gases, recoverable waste heat by waste heat recovery system (WHRS), scavenge pressure and air consumption of the main engine which is a result of higher scavenge pressure were investigated with a comprehensive energy efficiency perspective. Thus, the findings are interesting due to the enhancement of the energy efficiency of marine diesel engine operation.

2.2 Modelling of Energy Balance of Engine

2.2.1 Engine model

First law of thermodynamics is used to estimate the pressure and temperature of the in-cylinder values. Zero dimensional engine model with a single zone approach is utilized by assuming cylinder contents as an ideal gas. The model is developed for 12 cylinder, two stroke, turbo-charged, large bore marine diesel engine. The specifications of the marine diesel engine are given in Table 2.1.

Table 2.1: Ship and Main Engine Specifications (Kongsberg Maritime, 2019).

Ship Specifications		
Ship Type	:	Container Ship
Length Overall (LOA)	:	295 m
TEU	:	4200 TEU
Breath	:	32 m
Draught	:	12.6 m
Dead-weight	:	55,000 ton
Design Speed	:	25 knots
Engine Specifications		
Type	:	Sulzer RTA84C
Cylinder bore	:	840 mm
Piston stroke	:	2400 mm
Connecting rod lenght	:	2830 mm
Number of cylinders	:	12
Number of turbochargers	:	3
Continuous serv. rating of ME	:	48,600 kW
Corresponding engine speed	:	102 RPM
Mean indicated pressure	:	17,9
Propeller type	:	Fixed
Number of propeller blade	:	5
Propeller pitch	:	0.9 P/D
Specific fuel oil consumption	:	171 g/kWh

With an air standard cycle of the engine, gas properties are assumed as changing with the temperature. Specific heat capacities, C_P and C_V values, of the air highly vary as a function of in-cylinder gas temperature (Payri et al., 2011). A tabulated data set (Keenan, J.H., Chao, J., Kaye, 1983) is

$$\left[\frac{dQ_{in}}{d\Theta}\right] - \left[\frac{dW}{d\Theta}\right] = \left[\frac{dU}{d\Theta}\right] \tag{2.1}$$

The term, Q_{in} expresses the energy addition in to the cylinder, W is the work done by in-cylinder gases, and the U is the internal energy of cylinder gases. By using detailed descriptions of the terms, an extended expression can be written as;

$$\left[LHV\ \eta_c\ m_{fuel}\ \frac{dX_{fr}}{d\Theta} - \frac{dQ_{cool}}{d\Theta}\right] - \left[P\frac{dV}{d\Theta}\right] = \left[m_{in_{cyl}}\ c_v\ \frac{dT}{d\Theta}\right] \tag{2.2}$$

The fuel is a heavy fuel oil (HFO) with a lower heating value (LHV) 40700 kJ/kg. The combustion efficiency "η_c" is assumed as a complete combustion, 100%. The mass of injected fuel into the cylinder in a cycle is represented as "m_{fuel}" and the term "X_{fr}"is burning fraction rate of the fuel which is described with Wiebe function (Vibe, 1970) in this study. "Q_{cool}" represents the heat transfer rate from in-cylinder gases or into the cylinder. The heat transfer rate will be expressed with Woschni's heat transfer model (Woschni, 1967) to describe the relation between wall temperature and heat transfer rate. An ideal gas behaviour is assumed in this study. The pressure and temperature of the closed cycle of in-cylinder composition is calculated with a crank degree basis approach. Mean value engine model is adopted for the auxiliary equipment modelling. Filling and emptying model technique is used with a mass and energy conservation principles, an acceptable computation step can be taken as 1° crank angle (Karamangil et al., 2006).

2.2.1.1 Rate of heat release model

The heat release model is performed to describe the combustion process in the diesel engines. Total heat release rate is subdivided in to two superimposed heat release distribution to represent premixed and diffusive combustion in diesel engines. A single vibe (Vibe, 1970) function can be utilized as a double Vibe function in the situation to eliminate the disadvantage one exponential function (Watson et al., 1980). The model is highly accepted in many studies (Lakshminarayanan and Aghav, 2010; Merker et al., 2006; Scappin et al., 2012).

$$\frac{dX_{fr\_p}}{d\Theta} = a \times \left(m_p + 1\right) \times \left(\frac{\Theta_{ig} - \Theta}{CD}\right)^{m_p} e^{-a\times\left(\frac{\Theta_{ig}-\Theta}{CD}\right)^{m_p+1}} \tag{2.3}$$

$$\frac{dX_{fr\_d}}{d\Theta} = a \times (m_d + 1) \times \left(\frac{\Theta_{ig} - \Theta}{CD}\right)^{m_d} e^{-a\times\left(\frac{\Theta_{ig}-\Theta}{CD}\right)^{m_d+1}} \tag{2.4}$$

The total heat release distribution during combustion process can be expressed by overlapping of the two premixed and diffusive heat release functions in equation 2.5;

$$\frac{dX_{fr}}{d\Theta} = \frac{dX_{fr\_p}}{d\Theta} + \frac{dX_{fr_d}}{d\Theta} \tag{2.5}$$

“CD” denotes the combustion duration in terms of crank angle (Θ), “Θ_{ig}” represents the crank angle, where the combustion initiates. The parameter “m” is the Vibe form parameter, utilized to describe the shape of HRR.

A single equation model “The Arrhenius” Eq. 2.6 is used to describe the ignition delay period in diesel engines, considering in-cylinder pressure, temperature and air-fuel ratio (Merker et al., 2006).

$$\Delta t_{ID} = A\ \frac{\lambda}{P^2} exp\left[\frac{-E}{RT}\right] \tag{2.6}$$

Where, λ is air-fuel ratio, P is in-cylinder pressure, E is the activation energy, R is the universal gas constant and T is the in-cylinder temperature. After the ignition delay period in terms of crank angle, the combustion period will be carried out with the help of the Vibe function. The combustion parameters of the function as combustion duration, premixed and diffusive combustion shape parameters, are varied at different loads of ME as a function of engine speed, ignition delay, air-fuel ratio, in-cylinder pressure and temperature at the ignition time (Woschni and Anisits, 1974). After combustion process, expansion process starts by the time exhaust valve opens. The energy of combustion products, coming from its pressure and temperature, is utilized by turbocharger to supply pressurized fresh air to the engine.

2.2.1.2 Turbocharger

A simplified turbocharger model developed to represent the behaviour of turbocharger. Turbocharger allows for having a pressurized air pushed through the air cooler and into the combustion chamber with a designated pressure. The discharge air temperature from the turbocharger to the air cooler can be calculated with the following Eq. 2.7 (Merker et al., 2006).

$$T_{comp\_dis} = T_{comp_{in}} \left[1 + \frac{\left(\frac{P_{comp_{dis}}}{P_{comp_{in}}} \right)^{\frac{n-1}{n}} - 1}{\eta_{comp}} \right] \tag{2.7}$$

According to temperature increase in the air, required power can be calculated with the enthalpy variation of the air through the compressor together with the total efficiency of the T/C at the compressor side. The required power is afforded by the heat and pressure of the exhaust gases, which comes from the exhaust manifold. The generated power by the turbine can be calculated indirectly with the following Eq. 2.8.

$$N_{TC} = 30\, \dot{\mathrm{m}}_{\mathrm{air}} \, \frac{\left(\mathrm{h}_{\mathrm{air\_dis}} - \mathrm{h}_{\mathrm{air\_in}}\right)}{\pi\, \tau_{TC}} = 30\, \dot{\mathrm{m}}_{\mathrm{exh}} \, . \frac{\left(\mathrm{h}_{\mathrm{exh_{in}}} - \mathrm{h}_{\mathrm{exh_{dis}}}\right)}{\pi\, \tau_{TC}} \tag{2.8}$$

The enthalpy variation of exhaust gases can be written as a function of exhaust temperature. The temperature decrease through the turbine is the function of inlet exhaust temperature, pressure and turbine efficiency, which helps to calculation of the generated power by turbine in the Eq. 2.9.

$$\dot{\mathrm{P}}_{\mathrm{T}} = \dot{\mathrm{m}}_{\mathrm{exh}} . \dot{\mathrm{c}}_{\mathrm{p_{exh}}} . \eta_{T} . T_{T_{in}} \left[1 - \left(\frac{P_{T_{dis}}}{P_{T_{in}}} \right)^{\frac{n-1}{n}} \right] \tag{2.9}$$

Exhaust valve closing timing controller decides the timing as a function of scavenge pressure, and the opening timing is given in the operational specifications of main engine's digital twin (Kongsberg Maritime, 2019).

The temperature at the outlet of the turbine "$T_{T_{out}}$" is described as "$\left(\frac{P_{T_{dis}}}{P_{T_{in}}}\right)^{\frac{n-1}{n}}$" in the eq.9. After the turbocharger, the exhaust gas enters the waste heat recovery system to utilize the heat to produce electricity with the help of turbo generator. The schematic diagram of WHR system is represented in Figure 2.1.

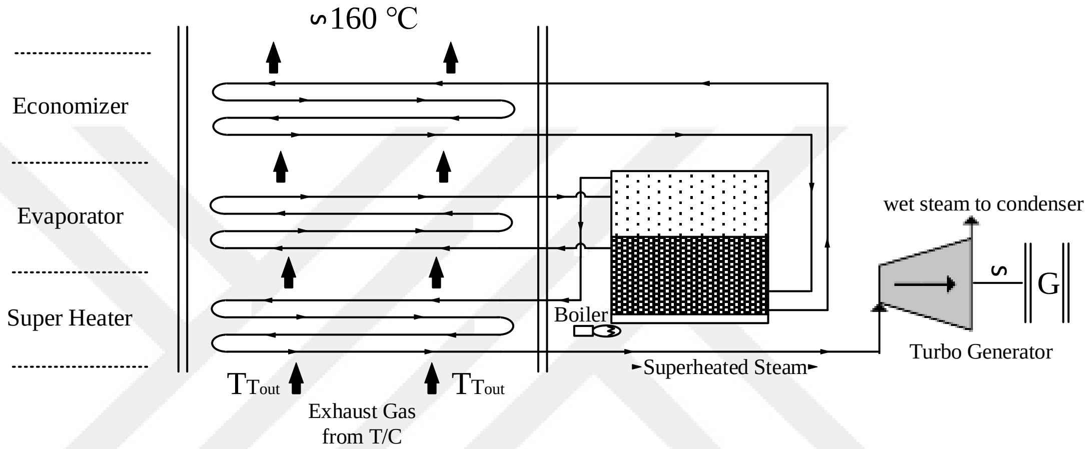

Figure 2.1: Schematic representation of waste heat recovery system.

The upper section is the economizer with relatively low exhaust temperatures. The less quality exhaust heat is recovered to heat the circulating water at this stage and the heated water returns to the pressurized tank. By this way the water can be prepared for evaporation by using waste heat. The second stage, the evaporator absorbs the heat to vaporize the high temperature water coming from the tank and after the evaporation process the saturated steam returns to tank. In the bottom section where the high temperature of exhaust gases faces with the saturated steam, the steam is heated by hot exhaust gases to send it steam turbine for electricity production. The exhaust gas enters the waste heat recovery system after T/C with high temperatures and leave around 160 ℃ to avoid sulphuric acid condensation. The calculations for potential recoverable heat amount are conducted assuming this phenomenon.

2.2.2 Heat transfer model

The present study is performed with a zero-dimensional model and does not include emission prediction of exhaust gases. Since the mean value models are not capable of predicting the in-cylinder specifications (Theotokatos et al., 2018), the first law analysis calculations were conducted with a zero-dimensional, single-zone model to simulate thermodynamic processes of the ME. The heat transfer, to the cylinder walls, which are cooled by the jacket water cooling system is modelled. Inlet and outlet temperatures of main engine jacket water are measured by two temperature sensors that are set inlet and outlet of the ME on freshwater cooling system. The system principle is represented in Figure 2.2.

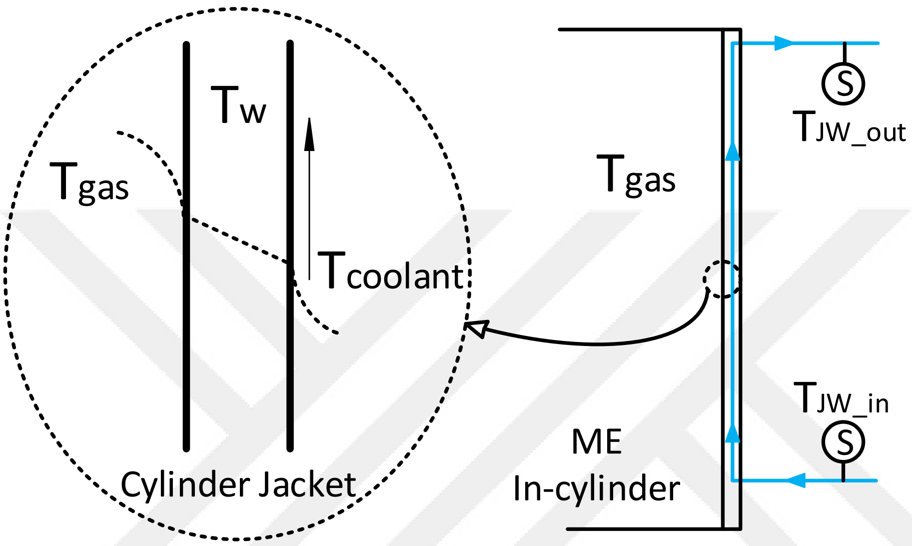

Figure 2.2: Schematic representation of cylinder jacket cooling system.

In the context of cooling modelling, a calculation process is carried out with the temperature of cylinder liners, inlet and outlet temperatures of jacket water cooling flow. The energy balance in one cycle of the engine could be expressed by the eq. 2.10.

$$\dot{Q}_{fuel} + \dot{Q}_{air} = \dot{P}_{bp} + \dot{Q}_{cool} + \dot{Q}_{exh} + \dot{Q}_{rad} + \dot{P}_{mech} \qquad (2.10)$$

$\dot{Q}_{rad}$ and $\dot{P}_{mech}$ are the loses which are difficult to measure during operating conditions are assumed as 0.3% and 6% ,respectively as indicated in the study (Morsy El Gohary and Abdou, 2011). The other terms, exhaust energy and cooling loss are seen in Eq. 2.11-2.12.

$$\dot{Q}_{exh} = [\dot{m}_{air} + \dot{m}_{fuel}] \,.\, h_{exh} \qquad (2.11)$$

$$\dot{Q}_{cool} = \dot{Q}_{jw} + \dot{Q}_{lub} \qquad (2.12)$$

$\dot{Q}_{jw}$ is the heat flux through the combustion chamber walls, removed by a cylinder jacket water in a steady state condition. The amount of the energy can be calculated with the Eq. 2.13.

$$\dot{Q}_{jw} = \dot{m}_{jw} c_{p_{jw}} \left(T_{jw_{out}} - T_{jw_{in}}\right) \tag{2.13}$$

The instantaneous heat flux in an internal combustion engine is highly dependent upon mean temperature of in-cylinder contents, in-cylinder flow conditions, liner temperatures and the surface area which is in contact with cylinder gases (Woschni, 1967). Total heat transfer can be estimated by integrating the instantaneous heat flux of each crank angle during the cycle.

Woschni (Woschni, 1967) heat transfer model is used to overcome the lack of accuracy of the most common other models which include Nusselt's approach (Annand, 1963; Hohenberg, 1979) in different phases of the cycle typically, scavenging, compression, combustion, and expansion phases. The general convective heat transfer equation is given in Eq. 2.14;

$$\dot{Q}_{cool} = hc\, A \left(T_{cyl} - T_w\right) \tag{2.14}$$

The term A is the heat transfer area, which changes by piston motion, T_{cyl} and T_w are mean temperature of in-cylinder gases and cylinder wall, respectively. The convective heat transfer coefficient hc [W/m^{2}K] is modelled using the Woschni model, in Eq. 2.15;

$$hc = C_0 D^{-0.2} w^{0.8} P^{0.8} T^{-0.53} \tag{2.15}$$

C_0 is the constant and it is determined by engine geometry, D denotes the cylinder bore diameter and the speed term w is expressed with the following Eq. 2.16.

$$w = C_1 C_m + C_2 \frac{V_d\, T_r}{V_r\, p_r} \left(p - p_0\right) \tag{2.16}$$

C_m is the mean piston speed, C_1 and C_2 are constants and depend on cylinder processes, as scavenging, compression, combustion and expansion. V_d denotes the volume of the displacement. T_r, V_r, P_r are the temperature, volume and pressure at the position of scavenge port closes, respectively. The pressure terms, p and p_0 are the pressures at the crank angle, in firing and without firing operations. Coefficients of Eq. 16 in the heat transfer sub-model, is calibrated for each type of engine. The calibration is achieved by measuring the heat transfer to the jacket water cooling system.

A is the contact surface of the cylinder gases with the cylinder wall, cover and piston surface and it can be explained by the Eq. 2.17.

$$A = \pi D^2/2 + \pi D x \tag{2.17}$$

Where the x is the instantaneous cylinder distance from the top of the combustion area. The distance is expressed by the geometric relations of components in Eq 2.18;

$$x = r[1 - \cos(\Theta)] + L_{rod}\left[1 - \cos\left(\sin^{-1}\left(\frac{\sin(\Theta)\, r}{L_{rod}}\right)\right)\right] \tag{2.18}$$

The "L_{rod}" is the connecting rod length, "r" is the radius of the crank, and Θ is crank angle with vertical direction. For each crank angle, the distance of "x" is calculated by using of equation 18, and the heat flux is calculated by utilizing Eq. 14 . The calculation of heat flux also requires the temperatures of cylinder cover and cylinder liner. To this end, the temperature data, obtained from high fidelity realistic engine model (Kongsberg Maritime, 2019), is used for cylinder cover, upper and lower cylinder liner temperatures.

In-cylinder gas temperature increases with a polytrophic function as the gases are compressed, and the burned gas temperature reduces during expansion stroke. Therefore, cylinder wall temperatures decrease as getting away from the TDC. The temperature gradient of the cylinder wall can be modelled with a polynomial function (Sigurdsson et al., 2014). The wall temperature gradient is assumed as invariable with the time for the steady main engine operations. The cooling fluid heat change is equal to the heat loss from the in-cylinder products to the cylinder wall, represented in Eq. 2.19.

$$\dot{\mathrm{Q}}_{\mathrm{cool}} - \mathrm{d}\dot{\mathrm{Q}}_{\mathrm{in-out_{jw}}} = \frac{dE_{wall}}{dt} = 0 \tag{2.19}$$

The heat loss to the jacket water cooling system see in Figure 2.1 can be calculated from the Eq. 2.20 with the temperature difference of jacket water through the main engine and water flow rate.

$$\dot{\mathrm{Q}}_{\mathrm{cool}} = \dot{\mathrm{Q}}_{\mathrm{in-out_{jw}}} = \dot{m}_{jw}\dot{C}_{p\_jw}\left(T_{in\_jw} - T_{out\_jw}\right) \tag{2.20}$$

The equations from 10 to 20 are utilized to calibration of all constants in heat flux sub-model. Steady-state heat transfer conditions are observed under different main engine loads, as 100%, 85%, 78%, 65%, 55% and 45%. In all ME loads, flow characteristics of jacket water side is assumed as same because of HTFW pumps are fixed drive to supply sufficient inlet pressure for main engine jacket water cooling system. The calculation process is conducted for each crank angle. For the simplicity, temperature gradient function calculates the liner temperature at each millimeter along with the liner. Besides, in each crank angle, heat transfer is carried out by the calculations between cylinder composition and interacted liner surface for each millimeter slides at relevant cylinder volume.

2.2.3 Cylinder wall temperature control

The heat transfer to the cooling medium is highly related with the cylinder wall temperatures as seen in Eq. 2.14. The temperature difference between gas and wall is changing with the crank angle because of the both temperature variation of in-cylinder gas content and temperature gradient of liner. The temperature pattern of liner is changing as a function of location, that distance from the TDC, and the engine load. A temperature gradient is interpolated by taking in to account three different measuring point temperatures in a vertical direction. In this way, more precise temperature values can be used to calculate heat transfer rate to the cylinder walls. The temperature gradient, along with the liner from the cylinder head to the bottom point could be modelled via second order polynomial function (Sigurdsson et al., 2014). The study in the ref. liner temperatures are 250℃ and 48℃ at the cylinder head and scavenge port locations, respectively and also they are assumed as a constant at the full engine load of the marine diesel engine. However, the engine load, engine speed, injection timing, in-cylinder composition, coolant temperature, and wall material are the main parameters of heat transfer to combustion walls while engine load and speed are the most two important ones (Heywood, 1988), as seen in Eq. 2.14. The mean temperature of the liner can also be expressed with a function which was described in ref. for homogeneous charge compression ignition engines (Neshat and Saray, 2014) for the variable engine loads with the help of the engine speed (rpm), indicated mean effective pressure (IMEP) and the mean coolant temperature.

Main purpose of the study is estimating the potential energy efficiency gain by keeping the cylinder wall temperatures of the main engine at the designated maximum permissible operation level. The temperature of cylinder cover and upper part of cylinder liner are the main parameters, to be controlled. The points which are exposed to thermal stress are the lower part of cylinder cover, and the upper of the cylinder liner (Romanov and Khozeniuk, 2016). The high temperatures of cylinder head and liner around TDC are highly depended on heat release timing and increased gas temperatures during compression process. Therefore, the temperature values of the cylinder wall do not remain as in the operation at maximum ME load at lower loads. In this study, the temperatures are allowed to reach the temperature which is a condition for upper MCR limits for all steady state operation of ME.

There are two kind of cooling systems in ships as sea water cooling system and central cooling system. The central cooling system is the most common one in recent years which uses fresh water to cool the systems and it is also cooled by sea water via heat exchangers, in order to eliminate corrosive effects seawater. As fresh water two main circuits are used with different temperature levels as low and high temperatures. The high temperature fresh water (HTFW) is used to cool down cylinder walls in marine diesel engines conventionally. A schematic diagram is of the HTFW cooling system is represented in Figure 2.3. The HTFW system uses a 3-way valve to set inlet temperature to ME. The fresh water coming from auxiliary coolers and coming from ME outlet are mixed and controlled by the 3-way valve. It is pumped to the ME engine with fixed drive HTFW pumps with nearly the same flow rate during all loads of main engine operation.

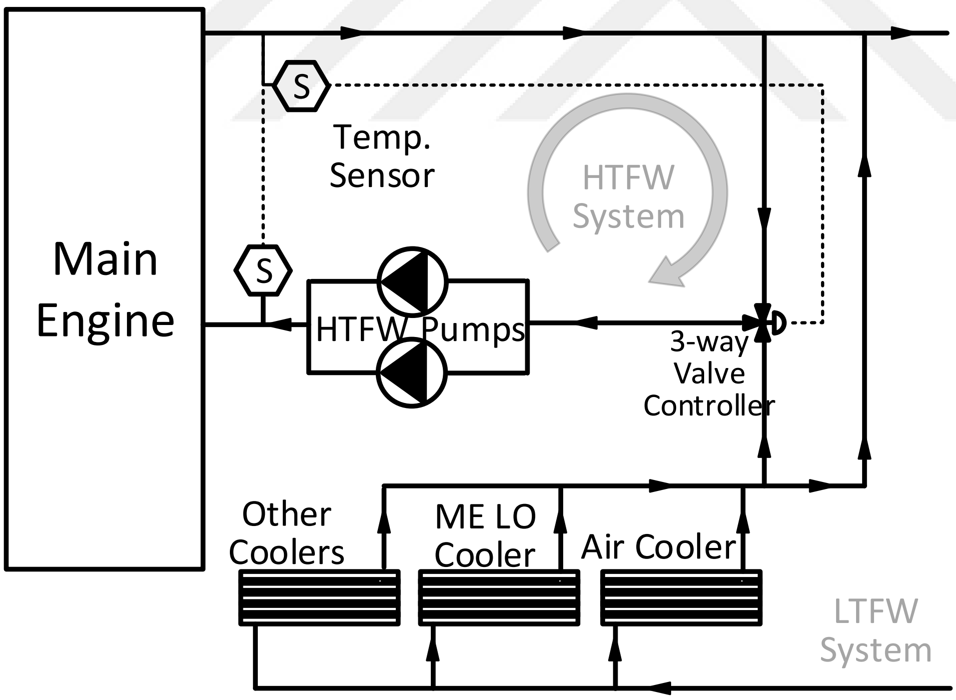

Figure 2.3: High Temperature Fresh Water Cooling system representation of ME.

In order to keep the liner temperatures in the range of operation at MCR, HTFW pumps pump the cooling water in to the cooling gallery of the liners. There are two kind of cooling phenomenon as convective heat transfer and boiling heat transfer, which is determined by vapor proportion fraction in cooling water and these two control the total heat transfer coefficient in the liners (Z. Zhang et al., 2020). The total heat transfer coefficient is highly affected by pressure and the velocity of the coolant, while the increase in flow velocity of coolant increases the convective heat transfer and reduces the boiling heat transfer under constant pressure (E. et al., 2018). However, HTFW pumps are fixed drive and they provide almost constant pressure and flow in ship cooling system. For the control of the cooling the measured temperatures as seen in Fig.4 are controlled during the operation. According to difference in desired temperature and temperature at the liners, inlet temperature of the coolant mixture is adjusted by inlet sensor and 3-way control valve in HTFW system seen in Figure 2.3. The glycol/water mixture provides extended temperature range to operate control the risk of boiling. The mixture, water with ethylene glycol is the most common mixture that used in the automobile engines to enhance thermal properties as freezing and boiling point, but comes with a lack of heat transfer capability (Sandhya et al., 2016). Since having a limited heat transfer coefficient, many nanofluid investigations had been conducted to enhance the heat transfer capability for the coolant (Al-Nimr and Alajlouni, 2018; Hatami et al., 2017; X. Li, Zou, and Qi, 2016). High potential heat transfer capabilities have been achieved by using different kind of nanofluids in the coolant fluid (Bigdeli et al.,; Peyghambarzadeh et al., 2011) which makes the coolant capable of responsive thermal changes even in small size engines (Che Sidik et al., 2017). If the liner temperatures are lower than the desired degree, the inlet temperature of the coolant is increased to decrease the heat difference between coolant and liner or vice versa. The values of inlet and outlet temperatures must be studied in different CFD research for each specific engine. In this study, it is considered that the liner temperatures were kept at the elevated temperatures as like 100% ME load in all main engine loads.

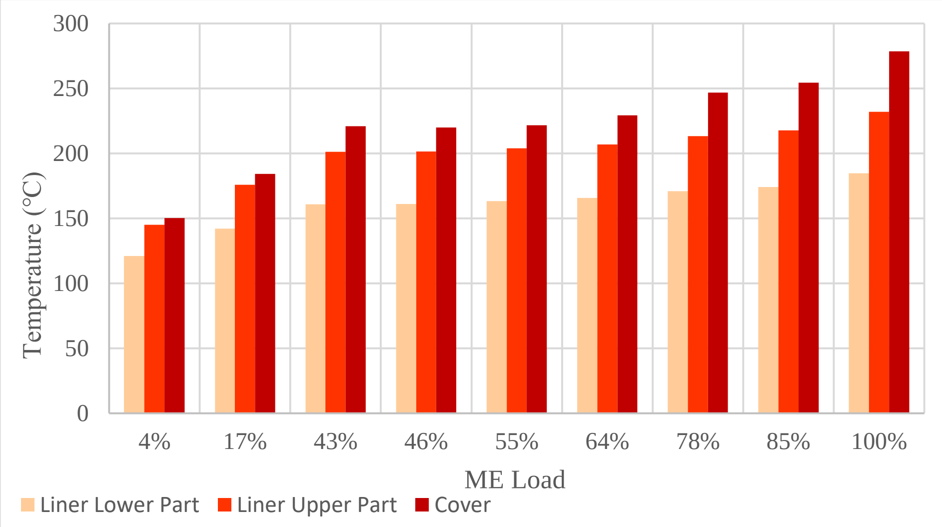

Figure 2.4: Liner and cylinder cover temperatures variation against to engine load as a result of changing heat transfer coefficient.

The cylinder liner temperatures for a given ship in this study are varying significantly with the operational load of main engine. The temperature variations are represented in the Figure 2.4.

While the cover and liner temperatures are much higher at the 100% MCR of the main engine, the temperatures decrease as the main engine load decreases. The main reason is that the differences of heat transfer characteristics of in-cylinder gases. The mean speed of engine (RPM) and the pressure variation in-cylinder change the heat convection coefficient and the less heat is transferred to the liner. By considering the pressure and speed parameters, cooling phenomenon must be adjusted for lower loads.

In addition, when main engine jacket water cooling inlet and outlet temperatures are examined, a considerable change on the temperature difference is observed at reduced loads. The Inlet and outlet temperatures of jacket water and also cooling load of jacket water system are given in Figure 2.5.

The temperature difference between inlet and outlet of ME is getting less at lower engine loads (red column). While the inlet temperature is 70°C, the outlet temperature is 85°C at full engine load and the 7.5% of fuel chemical energy is dissipated to the jacket water cooling system (seen in Fig. 5 - green line). The heat transfer to the jacket water is expressed as a percent of fuel chemical energy, in all variable loads. In the 85% ME load operation, the inlet temperature is adjusted 72°C to fix the outlet temperature 85°C. In this condition the heat dissipation has increased slightly. The most efficient operation of ME is 78% ME load that the lowest specific fuel oil consumption in this range. There is a slight increase of inlet temperature and dissipated heat to the jacket water. In 64%, 55% 46% and 43% of ME loads, while the liner temperatures are close to each other, inlet temperatures are set as 74°C, 75°C, 75°C and 76°C in these conditions.

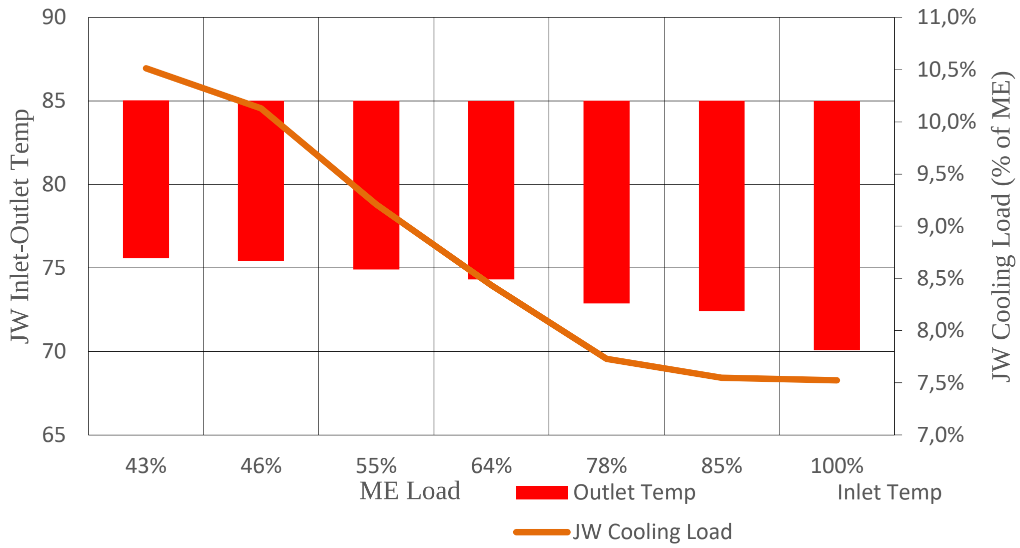

Figure 2.5: Jacket water temperature variation throughout liner and the increasing cooling load % of ME at reduced loads.

However, the heat dissipations to cooling water are 8.4%, 9.2%, 10.1% and 10.5%, respectively which make lower ME loads inefficient. Since the inlet and outlet temperatures are only controlled to keep the outlet temperature at a fixed degree (85°C) there is no consideration for the cooling requirement of specified engine load. By considering the flow rate of the HTFW almost the same rate, the first law of thermodynamics is utilized for control volume around the engine to calculate the output power, transferred energy to the cooling fluid and exhaust gases.

By applying the thermal engine model equations and engine heat transfer model, the given equations have been solved simultaneously for steady state conditions which are corresponding 85%, 78%, 65%, 55% and 45% operating loads of marine diesel engine. For the present study the maximum operating liner temperatures are used to investigate liner temperature effect on marine diesel engine efficiency by comparing with the reference temperatures as seen in Figure 2.4.

2.3 Results and Discussion

2.3.1 Validation of the model with reference data

A zero dimensional model is used to simulate two stroke marine diesel engine thermodynamic responses. The engine is 12 cylinder, cross-head type and fit with three turbochargers. The main engine specifications and the required operational data were taken from the digital-twin of engine, which represents the dynamic real engine behaviour (Kongsberg Maritime, 2019). The model was developed by using the engine geometrical data, turbocharger sub model, and injection and exhaust valve timing functions. The functions are described with the scavenge air pressure based control in the system.

For the validation of the model, 100%, 85%, 78%, 65%, 55% and 45% of the main engine maximum continuous rating loads are considered in steady state operation. The matching between experimental and predicted engine parameters show good agreement with each other for the great part of operating parameters. The percentage of the errors between the predicted and reference parameters are given in Table 2.2.

Table 2.2: The errors [Δ] between the predicted [*Pr*] and reference [Rf] parameters.

ME LOAD			100,0%	85,0%	78,0%	65,0%	55,0%	45,0%
RPM	ME speed	Rf	102,0	96,9	94,3	88,8	84,4	80,0
		Pr	*102,0*	*96,9*	*94,4*	*88,8*	*84,4*	*80,0*
		Δ	0,01%	0,02%	0,06%	-0,02%	-0,01%	0,00%
bara	ME Cylinder max pressure	Rf	137,4	133,3	132,5	124,0	116,2	107,9
		Pr	*138,9*	*132,2*	*129,8*	*121,7*	*114,2*	*107,1*
		Δ	1,10%	-0,84%	-2,07%	-1,90%	-1,71%	-0,74%
bar	T/C Compressor outlet Pres.	Rf	2,92	2,51	2,32	1,90	1,67	1,47
		Pr	*2,89*	*2,52*	*2,34*	*1,97*	*1,76*	*1,58*
		Δ	-1,03%	0,32%	1,03%	3,58%	5,39%	7,41%
ºC	T/C Turbine inlet Temperature	Rf	402,5	376,7	368,2	362,2	365,1	375,1
		Pr	*384,4*	*362,9*	*356,8*	*353,4*	*354,2*	*357,8*
		Δ	-4,50%	-3,67%	-3,10%	-2,44%	-2,99%	-4,62%
bar	T/C Turbine inlet Pressure	Rf	2,59	2,22	2,07	1,71	1,52	1,36
		Pr	*2,56*	*2,24*	*2,09*	*1,77*	*1,59*	*1,45*
		Δ	-1,02%	0,72%	0,82%	3,33%	4,61%	6,62%

The parameters predicted in this study are, main engine speed, maximum in-cylinder pressure, pressure and temperature of exhaust gas and scavenge air pressure. Additionally, jacket water cooling rate, specific fuel oil consumption and turbocharger power are also predicted to observe the effects of the cooling phenomenon. The comparison of the all variables together with their error range (according to RMS error anaysis) are represented in Figure 2.6.

It can be figured out that the model prediction represent acceptable accuracy for the operating load of the engine. Therefore, the engine model can be used for investigating the efficient engine operation.

The maximum pressure is slightly higher in 100% engine load and slightly lower in other engine loads with a maximum 2.1% variation with predicted and operational data. On the other hand, scavenge and exhaust pressures' trends are in good agreement with the collected data. The deviations may be caused by turbocharger model which is developed by a simplified modelling approach. Energy and mass conservation laws have applied while calculating the pressures in manifolds. Efficiency of turbocharger, mass flows and the specific heat variation of exhaust gas are also play an important role in the calculation of the pressures. The trend of the exhaust gas temperature is keeping the variation in 2.5% to 4.6% range in all engine loads.

At the lower engine loads, below from 60%, the exhaust manifold temperature increases due to the insufficient air supply by turbocharger which is caused by operating lower speeds. For the other parameter, the prediction of heat dissipation to the jacket water cooling, Woschni model is used by taking into consideration of engine size, engine speed, temperature and pressures. The heat dissipation rates are represented for one cylinder in 12 cylinder engine. The specific fuel oil consumption is estimated with at maximum 2.5% variation. The minimum specific fuel oil consumption is around 80% of ME load in predicted data while it is around 70-75% in reference data. A significant part of exhaust energy is recovered by turbocharger. The matching of utilized power of exhaust heat by turbocharger between predicted and reference data is also satisfactory.

The model does not contains exhaust emission properties of exhaust gases. The Vibe heat release model is used to predict heat release rates, in-cylinder pressure and temperatures. The turbocharger model has affected the prediction pressure in scavenge and exhaust manifold and the temperature of exhaust, therefore a slight increase is seen in turbocharger utilized power.

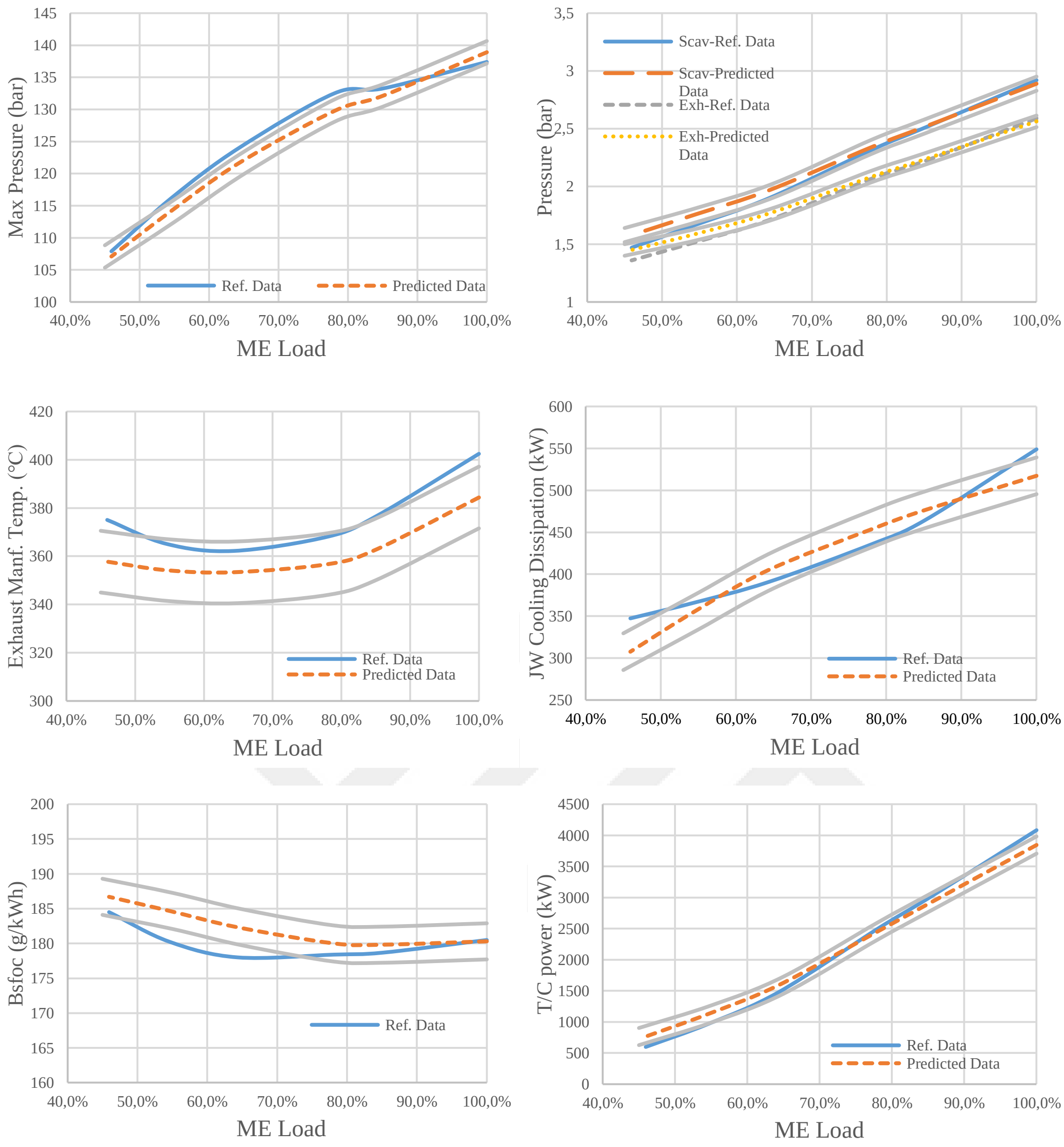

Figure 2.6: The comparison of operating parameters of the main engine and model.

As the load of the engine decreases, cylinder wall temperatures decreases because of low RPMs of ME and relatively low speed of air flow in the cylinder, result decreasing of convection constant. However keeping the cooling regime at the same rate, which means keeping the outlet of the JW cooling water at the same temperature (85°C in this engine) causes overcooling at reduced loads. The cooling effect will be investigated in the next step in terms of engine efficiency, by keeping the liner temperatures at the maximum operating degree in all engine loads.

2.3.2 Results of keeping liners at MCR operating temperatures

The matching between reference and predicted responses of the main engine was satisfactory for both engine and turbocharger variables (engine speed, and boost pressure exhaust and cooling rates) to investigate the liner temperature effects on the engine. The same operating parameters have been investigated at steady-state conditions for the same main engine loads. The temperature gradient of cylinder liner at 100% MCR is utilized for all other engine loads typically for 85%, 78%, 65%, 55% and 45%.

The new heat dissipation rates to jacket water cooling fluid are predicted with the verified engine model. In the all cases, ME loads and so engine RPMs were kept at the same rate. The other operating parameters are presented in Figure 2.7 with the maximum operating temperature gradient of the liner. There are slightly increment in the maximum in-cylinder pressures in all the engine loads. The maximum pressures were observed as 0.2 bar, 0.1 bar, 0.3 bar, 0.5 bar and 0.2 bar increment at 85%, 78%, 65%, 55% and 45% engine loads. The pressures in the scavenge and exhaust manifolds have also increased with the reduced heat dissipation strategy. The higher energy in-cylinder gases causes higher exhaust temperature and pressure which lead higher boost pressure in scavenge area via turbocharger.

It can be seen from the figure that as the load of the main engine decreased, the percentage of dissipated heat difference increased to the maximum then decreased slightly. The decrements in the heat dissipation rate to the jacket water cooling fluid are 24kW, 29.3kW, 35.3kW, 34.4kW and 30 kW for 85%, 78%, 65%, 55% and 45%, ME loads respectively. The amount of heat is given for one cylinder in the figure. By taking into consideration that the percentage of heat dissipation rates increases as ME load decreases, the heat rejected by cooling fluid are 7.7%, 8.0%, 8.5%, 8.7% and 8.8% for uncontrolled cylinder liner temperatures and the new cooling dissipation rates are 7.3%, 7.4%, 7.8% 7.9% and 8.0%. The heat flux reduction as 5.0%, 6.4% 8.7%, 9.7% and 9.6% has been achieved due to the new cylinder liner temperature. The remaining heat in the cylinder results in increase in the output power of engine and increase in exhaust gases energy. The output power increment affects the efficiency of the main engine.

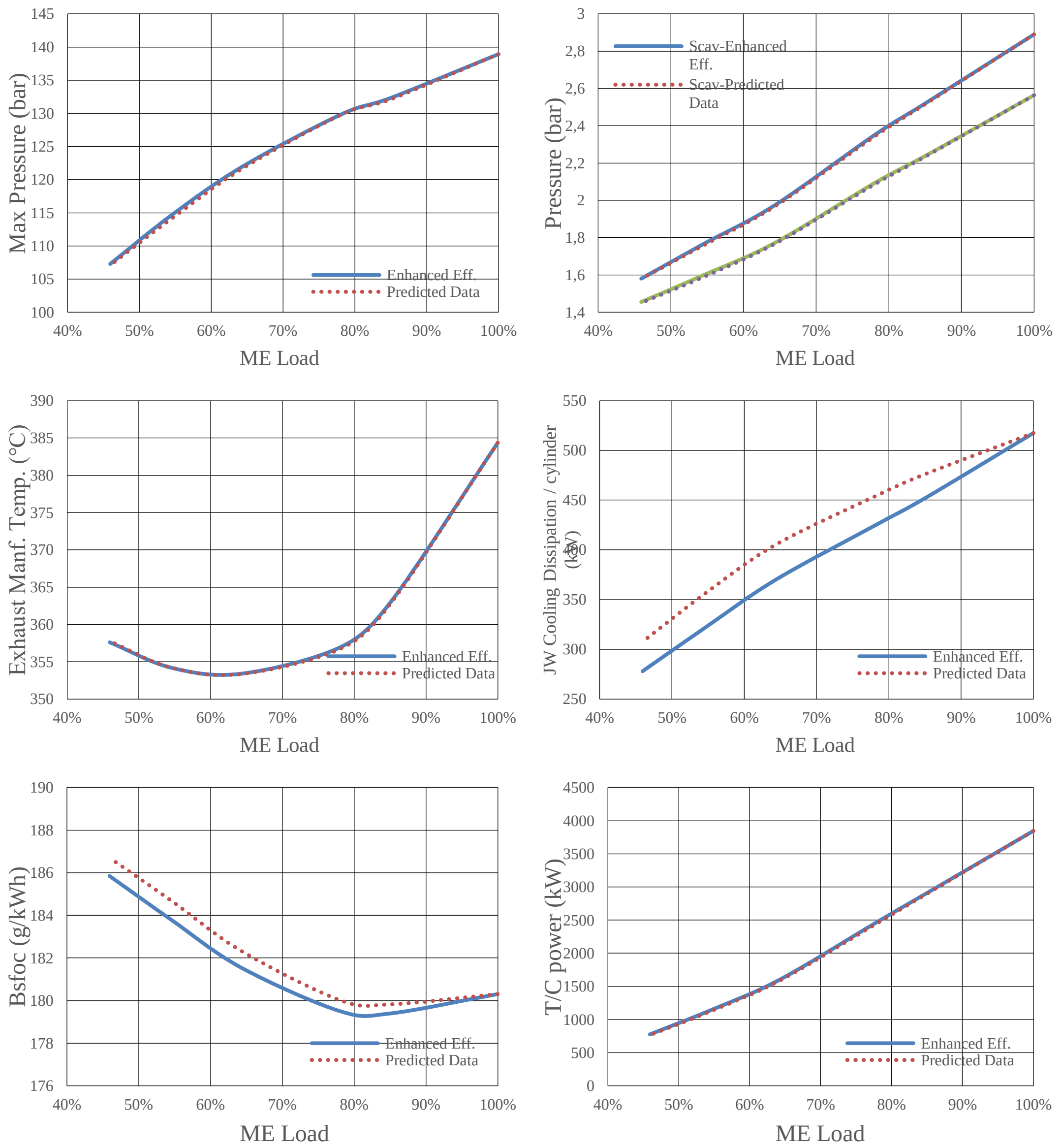

Figure 2.7: Variation of the operating parameters with the new cooling phenomenon.

There is a considerable efficiency increment in specific fuel oil consumption as up to 0.5% of nominal consumption. The maximum efficiency increment is achieved between 55% and 45% ME load and also 0.2%, 0.3% and 0.4% reduction for 85%, 78% and 65% ME loads, respectively. The other part of untransferred heat energy is utilized by turbocharger. The utilized power of exhaust gases by turbocharger is given only one turbocharger (1/3) in the Figure 2.6. There is a slight increment in the turbocharger power which results as minor increments in both scavenge and exhaust manifolds. When the less heat is transferred to the cooling fluid, the remained heat energy results as a power increment, higher turbocharger speed and higher exhaust temperatures. The distribution of the heat, utilized by ME as a power generation, in the turbochargers and in the exhaust gases are given with respect to main engine load in Figure 2.8.

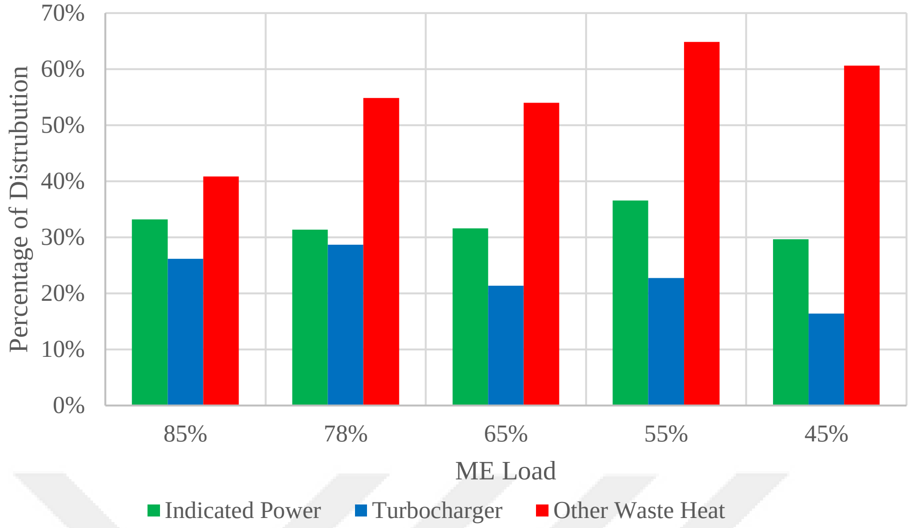

Figure 2.8: Unrejected heat impact to the other mediums.

The energy distribution of the heat is measured with the help of the model by performing the simulation with the same amount of consumed fuel in an hour. Results from the simulation point out that, in the 85% of ME, the amount of unrejected heat is 288 kW for the main engine and 26% of the energy is recovered by turbochargers. With the help of the turbocharger boost effect, 33% of this energy is utilized by in-cylinder gases to produce work, this is a combined effect of turbocharger and remained heat though. A long with the higher energy in cylinder gases, the increment in the energy of exhaust gases after turbocharger equals 41% of unrejected heat. These effects are interpenetrated to each other which is described with the effect of turbocharger in diesel engines. Even though the power increments in power output and turbocharger are higher in terms of kW in the 78% ME load, the percentages are lesser as 31% and 28%, respectively, where the heat rejection reduction corresponds 339 kW in this load. As a result of this remained energy, 55% of the heat energy passes to exhaust side. In the 65% ME load, the highest difference in heat rejection is observed as 406 kW. As a consequence of undissipated heat by the JW cooling system, 31% and 21% of this energy equivalence are calculated as the power increments in the main engine output and turbo compounds, respectively together with the 54% of correspondent energy raise in exhaust gases. The observed heat dissipation gains, are relatively higher at 55% and 45% ME load than other first two loads as 370 kW and 341 kW. Additionally, the energy, transferred to the exhaust gases have the highest proportion as 65% and 60% equivalent, when they are compared with the other engine loads. At the turbocharger side, the operational power increase 23% and 16% of unrejected heat amount and so, 36% and 30% of the energy could be used as an output power of ME. The energy balance is a nested situation in turbodiesel engines. The amount of energy recovered by low heat rejection cannot be addressed as separately. Therefore it is expressed only in terms of equivalent percentages. The turbocharger has effect on both ME power and exhaust heat. The in-cylinder gas temperature raise, which is derived from less heat dissipation, not only helps the main engine to produce more power but also helps the turbochargers to increase boost pressure, with a bilateral effect. Furthermore, some of the energy coming from turbocharger also causes to increase the waste heat energy in exhaust gases.

The study focuses on the energy efficiency that the most important parameter is specific fuel oil consumption of the main engine. The effect of low heat rejection by JW cooling system was investigated at the above for the same amount of fuel consumption. This part is performed under same ME power output conditions. Proposed cooling effect is discussed under energy efficiency perspective. Ship's operating load and speed, operating profile, specific fuel oil consumption variation and the reduction in the hourly fuel consumption of the ship is given in the Table 2.3.

The fuel saving in a year is calculated based on 280 days operation in a year. The merchant ships do not use 100% of their machine power while cruising at their design speed. The operation profile is considered starting from 85% to 45% ME load. As in the study, performed in ref. (Dere and Deniz, 2019b) the operational profile taken as 10% for both 85% and 78% of ME loads, 20% contribution at 65% load and 30% for 55% and 45% ME loads. The respective fuel savings for an hour are given for each operating load. In 85% ME load, 15.5 kg fuel saving in an hour could be achieved and 17.3kg, 21.8kg, 21.1kg, 16.8 kg fuel amount could be also saved in other loads respectively. While the maximum fuel saving is in 65% ME load, the maximum percentage of the savings are in 55% and 45% ME loads. By multiplying the operational profile of the ship, the total amount of saving is 127.8 tons fuel in a year. By considering the CO_2 emission factor of marine HFO "3.114 g/g.fuel" (IMO, 2014) 398 tons of CO_2 emission is not released to the atmosphere.

Table 2.3: Fuel consumption decrement in the operation.

ME LOAD		100,0%	85,0%	78,0%	65,0%	55,0%	45,0%
Ship Operational Profile %			10%	10%	20%	30%	30%
Ship Speed	knots	25,1	23,9	23,3	22,0	21,0	20,0
Conventional Cooling	SFOC (g/kWh)	180,31	179,82	180,01	182,32	184,68	186,70
Controlled Liner Temp	SFOC (g/kWh)	180,31	179,40	179,50	181,54	183,78	185,85
Efficiency Enhancement	Δ SFOC (g/kWh)	0,00	0,42	0,51	0,79	0,90	0,85
	%	0,0%	-0,2%	-0,3%	-0,4%	-0,5%	-0,5%
Saving in Hourly Consumption of ME	kg/h	0,0	15,5	17,3	21,8	21,1	16,8
Saving in Fuel Consumption in a Year	ton/year	0,0	104,4	116,2	146,2	141,8	113,1
Saving in Total	ton/year	127,8					

On the other hand, there is another significant energy potential in exhaust gases. The waste heat recovery system and utilization of additional waste have been also investigated in this study. There is an increment in the available energy after the turbocharger, which cannot be underestimated, that is wasted via exhaust gases. The model shows that, at least 40% of the unrejected heat remains as an exhaust gas energy. This amount equals approximately 110 kW, 185kW, 220kW, 240kW and 205kW respective to between 85% and 45% ME loads. However, the all remained heat cannot be recovered by waste heat system by the reason of total heat increment is derived from the both temperature and exhaust flow increase. The model calculates recoverable heat by considering both of the parameters and as a result of the analysis maximum 17,3% and minimum 11,1% of the remained heat could be recovered by waste heat system for the electricity production which is represented in Figure 2.9 against the ME load.

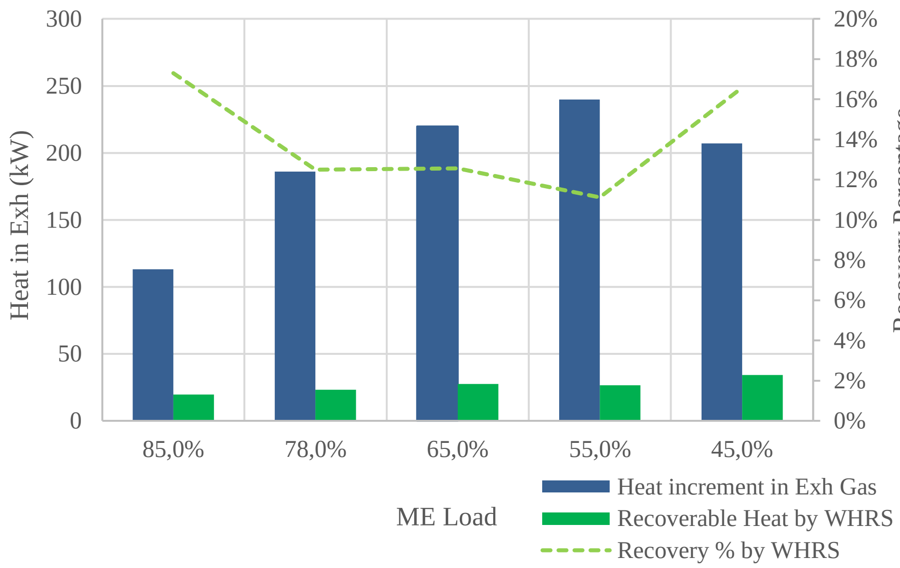

Figure 2.9: Heat energy increment in exhaust gases and available recovery.

As seen in Figure 2.9, most part of the heat increment is remained in exhaust gases after WHRS. This is because of increase both of the air flow and exhaust flow, which is a result of higher boost pressures. The electricity production of a waste heat system is dependent on exhaust gas flow and temperature nevertheless, the electricity consumption variation does not change with the ME load. The major electricity consumers are pumps in ships as; sea water pumps, fresh water pumps, lubricating oil pumps and they work as long as the ship is in operation and need energy. However, in reduced ME loads, the available heat in exhaust is limited and the waste heat system works together with oil fired boiler to produce demanded heat for turbogenerator (IMO, 2014), below the 70% ME load for the modelled ship, the heat is provided by waste heat and oil fired boiler (Dere and Deniz, 2019b). Any increment in the available heat helps to reduce oil fired boiler fuel consumption where the specific fuel oil consumption is assumed as 305 g/kWh as a constant value (IMO, 2014). Table 2.4 shows the recoverable heat amount from exhaust gases and their savings in oil fired boiler in terms of kW and hourly fuel consumption.

According to operational profile of the ship and using the mass based CO_2 emission factor, the recovered heat by WHRS and corresponding fuel oil consumption of oil fired boiler are indicated. It is not necessary to operate oil fired boiler for electricity production above 70% ME loads. Therefore, the effect of the heat energy recovered by WHRS higher than 70% loads on the fuel consumption of the oil fired boiler is not considered for fuel saving.

Table 2.4: Fuel saving in oil fired boiler operation as a result of increased heat in WHRS.

ME LOAD		100,0%	85,0%	78,0%	65,0%	55,0%	45,0%
Ship Operational Profile %		-	10%	10%	20%	30%	30%
Ship Speed	knots	25,1	23,9	23,3	22,0	21,0	20,0
Increment in Recovered Energy by WHRS	kW	0,0	19,6	23,3	27,6	26,7	34,3
Horuly Saving in Oil Fired Boiler	kg/h	0	0	0	8,4	8,1	10,5
Yearly Saving in Oil Fired Boiler	ton/year	-	-	-	56,6	54,7	70,4
		WHRS satisfies energy demand, oil fired boiler operation is not required			WHRS can not satisfy energy demand oil fired boiler operation is required		
Saving in Total	ton/year	48,8					

In the operation below this load, hourly and annual fuel savings were calculated according to the specific fuel consumption and operational profile of the ship. The corresponding savings in a year, 56.6, 54.7, 70.4 tons of fuel equal 0.2%, 0.2% and 0.3% of ME fuel consumption in 65%, 55% and 45% ME loads, respectively. These savings values are the additional savings besides the main engine fuel consumption and the fuel consumption reduction, to be saved in the oil fired boiler that operates independently from the main engine to assist the WHRS. The total fuel saving is 48.8 ton/year for oil fired boiler which is derived from increased heat potential of waste heat recovery system. The CO_2 emission is calculated according to the emission equivalent of HFO which is 3.114 g/g.fuel (IMO, 2014) and saving in CO_2 is 152 tons.

With the operational improvement, the cooling rate was regulated by the reduction in the need for ME cooling as the ME is operated away from the maximum operation range. In this regard, four groups of energy increase are observed as the main machine indicated power increase, the power increase in turbo compound, the recovered energy increase in WHRS after T/C and the heat increase in the exhaust gas coming out of the funnel. The distributions are shown in the Figure 2.10.

The percentages are given as a ratio of unrejected heat. In each operating load, the unrejected heat is different and represented as 100% regardless of kW to normalize the ratios. The chart shows the distribution of percentage of the utilization of unrejected heat in indicated power, T/C, WHRS and wasted heat via exhaust gas. For the indicated power 33%, 31%, 32%, 36% and 30% of unrejected heat could be utilized as an increment in power production.

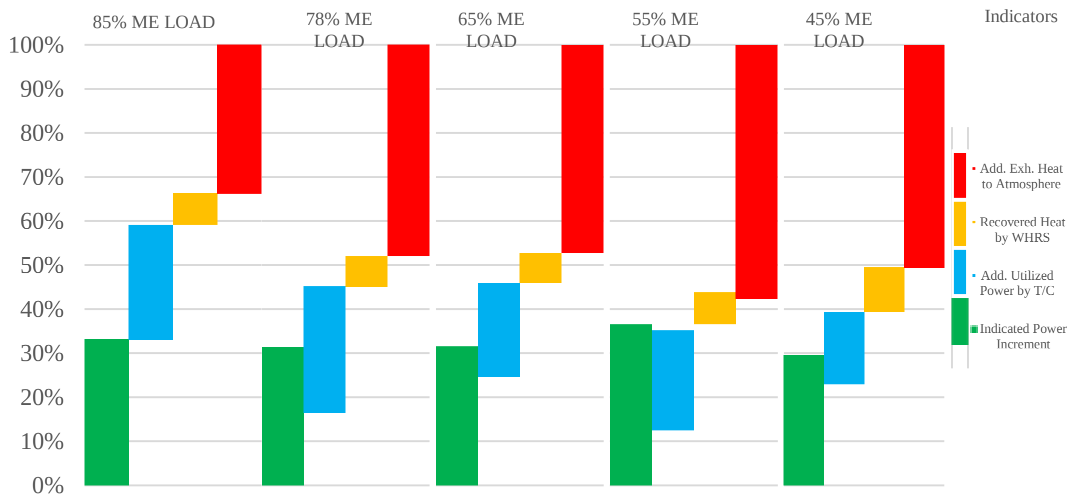

Figure 2.10: Energy distribution of unrejected heat on the indicated power increment (green), T/C power (blue), recovered heat by WHRS (yellow) and Exhaust Heat after WHRS (red).

In these power increments, it is obvious that the help of boost pressure of T/Cs, where 26%, 29%, 21%, 23%, 16% of unrejected heat energy are used in operational power of them. Because of the typical characteristics of turbocharged engines, the energy distribution of power increment in ME power and T/C must be considered as integrated value. The some part of the power increment in T/C is also used as boost factor of ME power increment. After the turbochargers 7% of unrejected heat recovered by WHRS in 85%, 78%, 65%, 55% ME loads and 10% heat is recovered in 45% ME load. The remaining heat rates which are 41%, 55%, 54% 65% and 60% of unrejected heat are wasted via exhaust gases.

2.4 Conclusion

A zero dimensional model was developed and validated to investigate cylinder liner temperature variation effect for efficiency at the wide range of operation load of a two stroke marine diesel engine. The validation has been completed to represent satisfying results of the engine for the engine model, which shows good agreement with the reference data for the combustion parameters. The analysis reveals that, during the low engine loads, heat loses to the cooling medium increases in terms of percentage of generated power of main engine and the efficiency decreases. With proposed approach, heat dissipation have been optimised by controlling liner temperatures at the permissible rate for 100% MCR. Here, the cylinder wall temperature is used as a control parameter to monitor the effect of variable temperatures on, scavenge pressure, compression pressure, maximum combustion pressure, and the temperature and pressure of exhaust manifold. Additionally, specific fuel oil consumption and waste heat recovery availability are evaluated. Keeping more energy in the cylinder at the reduced ME loads with the proposed liner temperatures causes the reduction of irreversibility in the engine, which can reduce fuel consumption with high exhaust energy availability and high combustion pressures, assisted by high scavenge pressure. Respectable part of unrejected heat energy could be converted to the work output (around 30%) for the marine diesel engine. The output power increment is assisted by turbocharger with higher boost pressures, where the some part of unrejected heat is utilized (16% - 28%). Furthermore some of the energy recovered in waste heat recovery system (7%-10%) from elevated heat of exhaust. To sum up, the specific fuel oil consumption could be decreased up to 0.5%. By this way, 127.8 ton fuel and 398 tons of CO_2 emission could be saved. In the oil fired boiler operation, the additional corresponding savings up to 0.3% of ME fuel consumption could be achieved with the increased potential of WHRS. In total 176.6 tons of fuel with 550 tons of CO_2 emission in a year can be mitigated

An application of cylinder wall temperature controlled cooling system requires additional equipment as liner temperature sensors, and wide range of temperature flexibility in the high temperature cooling system for marine diesel engines. The modification, ethylene-glycol/water mixture is required in HTFW system because, higher temperature levels cannot be achieved with only conventional fresh water circulation. Additionally, although simulation provides a good basis for optimal cooling system design it is required to mention that experimental studies are necessary to optimizing the engine cooling by measuring liner temperatures to adjust cooling strategy. Nucleate boiling in the jacket water system is the other factor affecting the heat transfer rate, computational fluid dynamics study needs to be conducted at the coolant side. And to enhance the efficiency concern the elevated temperature of the coolant mixture, provides more available heat to organic rankine cycle to recover energy from higher quality heat. Additional heat recovery system studies can also be conducted after CFD studies.

3. LOAD OPTIMIZATION OF CENTRAL COOLING SYSTEM PUMPS OF A CONTAINER SHIP FOR THE SLOW STEAMING CONDITIONS TO ENHANCE THE ENERGY EFFICIENCY[2]

3.1 Introduction

More than 90% of global trade is carried out by commercial vessels (Deniz and Zincir, 2016; Harrould-Koleib, 2008) Maritime transportation responsible for approximately 3.1% of global CO_2 emission and 2.8% of global greenhouse gases (GHG) (IMO, 2014). Maritime transportation is the most cost efficient way in transportation sector and thus, it should be environmentally efficient (Fan et al., 2018). In this regard, the International Maritime Organization (IMO) introduced Energy Efficiency Design Index (EEDI) which aims to reduce in CO_2 emissions from ships (IMO, 2012a). The regulations will get stricter in every five year periods until 2030, which are the leading motivators for improvement of ship operational efficiency as EEDI, Ship Energy Efficiency Management Plan (SEEMP) (IMO, 2011) and Energy Efficiency Operational Indicator (EEOI) (IMO, 2009). The most common efficiency gain method is the reduction of vessel speed. In order to comply with the efficiency and environmental challenges, speed reduction is a prominent way which is known as “slow steaming” in shipping parlance. This practical operation is applied to both new built and existing ships. In order to carry out energy efficiency and energy management methodologies, most frequently operating points were investigated among tankers, bulk carriers, container ships and revealed that most of ships are operating in slow steaming conditions (Tsitsilonis and Theotokatos, 2018). However, slow steaming causes delays in voyage time. The delay in delivery time can also be caused by many kinds of reasons typically, over work load or lack of capability of ports, heavy weather conditions or propulsion problems. In order to overcome to delivery problems, higher

[2] Dere C., Deniz C. 2019.Load Optimization of Central Cooling System Pumps of a Container Ship for the Slow Steaming Conditions to Enhance the Efficiency, Journal of Cleaner Production - Volume 222, 10 June 2019, Pages 206-217 https://doi.org/10.1016/j.jclepro.2019.03.030

voyage speeds also could be a solution occasionally in these specific cases (Lee et al., 2015; Notteboom and Vernimmen, 2009).

The function of fuel consumption against vessel speed has a cubic relationship (Psaraftis and Kontovas, 2014). Whilst the legislations are enacted to decrease environmental impact of maritime transportation, financial concerns are the main challenge for maritime corporations. Both financial and environmental issues are taken into consideration in slow steaming conditions. Studies were conducted as considering ship routing and scheduling (Psaraftis and Kontovas, 2014; Tai and Lin, 2013) fuel price, market balance and freight rates in global trade (Lindstad et al., 2013; Psaraftis and Kontovas, 2010; Wen et al., 2017; Wong et al., 2015). Simulation based models were developed to estimate optimum operational speed by considering these parameters (Lindstad et al., 2011; Magirou et al., 2015; Wen et al., 2016). In order to determine optimum vessel speed, regarding environmental concerns or cost benefit actions have different priorities. In view of the environmental subject, weather conditions and ship routing are important factors, while the fuel cost and freight rates are the leading parameters to achieve benefits in cost saving. Specifying optimal speed is a dynamic process that it is heavily depended on market condition, bunker prices and freight rates, therefore a great part of the studies were carried out by considering revenue generation (Chang and Wang, 2014).

Vessel fuel consumption is mainly depending on vessel's speed because of the relation between speed and ship resistance (MAN Diesel and Turbo, 2013a). The higher speed requires a higher propulsion power. As a direct consequence, ship speed significantly affects the fuel consumption, emissions and operating costs. Since container ships have usually higher voyage speeds than both tankers and bulk carriers, speed reduction is more popular practice in container shipping. In recent years, strict emission limitations and decreasing rate of global trade have driven shipping operations to be carried out at slow speed rates to reduce the operational costs and to satisfy IMO emission limits (Psaraftis and Kontovas, 2014). Slow steaming is spurred by low trade volume along with the over capacity of merchant fleet. Excess idle vessel capacity, which is derived from decreased global trade, could be diminished with the help of the slow steaming operations (Wiesmann, 2010). Slow steaming operations initiated by Maersk Line for container vessels in 2007 (Maersk, 2013). Many container ships as, tramp freighters and liners have applied slow steaming, extra slow steaming and super slow steaming, which run with 21 knots, 18 knots and 15 knots respectively for a ship who has 24 knots full speed (Liang, 2014). Container vessels' speeds could be reduced to half of their design speed with very low engine loads which are more than traditional load reduction rates without endangering operational safety. The container ships consume large part of the fuel in shipping market and they are responsible for greater part of CO_2 emissions in the world because of their higher speeds (Psaraftis and Kontovas, 2009). This is the reason of why slow steaming has been popular in container shipping. According to excess idle vessel capacity, reduced trade volume and challenging emission limits, slow steaming is expected to be a permanent solution in the future. It would be preferred by operators to reduce both CO_2 emissions and operating costs (Corbett et al., 2009). It is already being considered as an ordinary operation in new design ships (Woo and Moon, 2014). The recent studies had been conducted considering economical aspects of slow steaming as reducing fuel consumption for the propulsion, trading rates, freights or voyage durations. Optimum speed can be calculated by taking in to consideration of vessel size, engine power, voyage distance, bunker fuel prices, and voyage specifications (Woo and Moon, 2014).

The fuel cost could account for 50% to 70% of the total operating costs depending on ship type and specifications. This is the highest portion of total costs among all the ship types (Fagerholt et al., 2015; Guangrong, 2017). In the case of slow steaming operations, main engine (ME) fuel consumption can be easily cut down. It also reduces the combustion emissions from ships which have harmful effects to the environment. CO_2 emission, as combustion emission, causes climate change therefore, IMO enforces shipping companies and operators to mitigate CO_2 emissions (EEDI) from their ships (IMO, 2012b). In a speed reduction application, the study reveals that while the EEDI of a container ship with 145m length with design speed of 16 knots is 12.89 gCO_2/(ton per nautical mile) the value can be reduced to 9.87 gCO_2/(ton.nm) at 14 knots (Molland et al., 2014). Nevertheless, the same rate of fuel and emission savings cannot be achieved in every steps of speed reduction. Deceleration at higher speeds provides more benefit than lower speeds both in operating costs and emissions. Applying the method at low voyage speeds may lessen the marginal utility of the method. Although speed reduction gives a great advantage in diminishing CO_2 emissions, it is not possible to navigate at the lowest speed because of economic reasons. Woo and Moon (2013) studied about effects of speed reduction in liner shipping. They revealed that, the range between 24 to 22 knots efficiently satisfies IMO reduction target for the container ship which has high power output and has 25 knots designed voyage speed. Additionally, a 50% of CO_2 reduction was achieved at the speed of 22 knots in their study. Though slow steaming can be operated at even lower speeds. The minimum speed is specified as 13.7 knots where the most efficient speed 18.6 knots in the study for slow steaming (Woo and Moon, 2014).

Many of ship types are appropriate for cubic modelling for the estimation of power requirement regarding to a ship's speed. However, the model may not give realistic outputs at high speed large container ships at slow steaming which is caused by other electricity consumers (Psaraftis and Kontovas, 2014). Total fuel consumption of a ship is not only depended on main engine (ME) fuel consumption, auxiliary engines as generators and boilers have contribution which are taken in part on fuel consumption. Both main engine and auxiliary engines are required to supply electrical power when needed. However, apart from cutting down of power requirements for propulsion, reduction in ship auxiliary equipment's electrical energy demand is also considered prerequisite factor for fulfilment of total energy efficiency concept. During slow steaming operation, auxiliary machines have a critical role on electrical power consumption. Waste heat recovery systems as exhaust gas boiler cannot be operated with exhaust gases alone due to the reduced exhaust gas amount and its temperature. Electricity generation is supplied by turbo generator coupled with oil fired boiler or auxiliary diesel generators. Since auxiliary engine fuel consumption is taken into account in calculation of EEOI, electricity consumption is an important parameter from the view point of the regulations. Additional fuel savings could be achieved by the reducing electric energy demand which is produced by diesel generators, shaft generators or steam turbine.

Electrical power consumption of pumping equipment is account for 70% of total electrical consumption in ships (Chun-Lien Su et al., 2014). Many pumps work at a fixed rate and appropriate for utilization of variable speed pumps (VSP) to reduce energy consumption (The Hydraulic Institute, 2004). Almeida et al. (Almeida et al., 2005), Rooks and Wallace (Rooks and Wallace, 2004), Wiechmann et al. (Wiechmann et al., 2008) studied about energy saving potential and applicability of variable speed pumps. They revealed that VSPs could be used in many areas and have a great potential in terms of energy efficiency (Almeida et al., 2005; Rooks and Wallace, 2004;

Wiechmann et al., 2008). Energy efficient control of cooling systems with utilizing VSPs, in chiller systems (S. Wang and Burnett, 2001), at heat exchanger optimisations (Y. F. Wang and Chen, 2015), in buildings' air conditioning (Zhenjun Ma and Wang, 2009), in fossil fuelled power plants and nuclear power plants (Xia et al., 2015) were conducted in previous studies. The utilizing of variable speed pumps in marine power plants provides an advantage in operating cost of ship cooling system.

Tomislav Mirakovcic (Mrakovcic et al., 2004) studied on modelling of ship central cooling system optimisation. The numerical study had been conducted to reveal determination of pressure drop or flow rates estimation in a cooling system design. The model prevents improper selection of equipment, leads high flow rates and energy consumption, at the design stage (Mrakovcic et al., 2004). Chantasiriwan (2013) investigated the performance of variable speed pumps for the energy efficiency in his study (Chantasiriwan, 2013). An energy efficiency method, developed by Giannoutsos and Manias, proposes an engine room power balance method to reduce diesel generator fuel consumption via utilizing frequency control of engine room air fans and sea water pumps (Giannoutsos and Manias, 2013). The study considers, air requirement of main engine and system temperatures in order to develop power management system for a typical tanker vessel.

Chun-Lien Su (Chun-Lien Su et al., 2014) developed a method which calculates the savings in energy on sea water pumps with installing variable speed pumps to the system on ship. Piping system, marine equipment and sea water temperatures were considered and modelled in the study. The study showed that sea water temperatures variation as a result of different routes have great effect on energy savings.

Qi, Jiao and Zheng (Qi et al., 2016) investigated the potential savings on sea water flow rates. A simulation study was conducted by considering different sea water temperatures and operational load as main factors. Frequency controlled pump was utilized in the study. They compared reference flow rates with low sea water temperatures combined with three different main engine load as 70%, 80% and 100% of ME power range. 26% to 100% variation could be controlled on sea water flow rates to realize energy saving and emission reduction. (Qi et al., 2016).

Flow optimisation and efficiency analysis using variable speed pumps on sea water system for different sea water temperatures had been conducted by Durmusoglu

(Durmusoglu and Kocak, 2019). It was assumed that, the ships are operated at the range of their design speed. They found out that this approach significantly reduces the cooling flows. Theotokatos et al. (Theotokatos et al., 2017) investigated for the enhancing of cooling system operation efficiency with variable speed pumps at different ME loads and the power consumption could be decreased in study.

Variable Speed Pump (VSP) calculations were carried out by researchers for ships. As stated in the studies, reducing sea water flow rate in ship cooling system leads a great advantage in energy consumption of the pumps.

According to aforementioned studies, slow steaming has been very popular in recent years. Therefore, due to the reduced main engine loads, cooling demand reduces in fresh water system. The circulation flow of both fresh water and sea water flow could be reduced by considering the heat dissipation rates in the system. The most efficient optimisation can be possible when both of the cooling circuits are considered simultaneously.

For the reason that, many slow steaming studies were carried out by considering financial priorities, fuel and emission reduction is just coming as a result of cost reduction. To the best of authors' knowledge, the operational efficiency perspective of cooling system of ME and cooling equipment has not been studied before in slow steaming conditions. The cooling system researches in literature mainly consider sea water system pumps and sea water temperatures. This article proposes a solution for the slow steaming operation with the combination of both sea water and fresh water cooling system demand optimization for varying main engine loads. Operational characteristics of cooling system are obtained from Engine Room Simulator (ERS) of a container ship and verified with technical project guide of an engine which has similar power output. Permanent technical solutions can be realized in slow steaming operation, regardless of bunker price, freight and global economic trade, were discussed. The approach provides a cut down mechanism for operational energy demand of central cooling system. In this study, excess electrical energy consumption is diminished in the proposed method without endangering operational safety in slow steaming operations. Electricity productivity of waste heat recovery system is also considered, and available electrical power and power requirement are determined. Corresponding cost of differential electrical power for both fixed and variable speed pumps are demonstrated with using diesel generator and oil fired boiler.

3.2 Operational Characteristics of Speed Reduction

Slow steaming is intentionally reducing ship speed to comply with emission regulations and optimize operating costs for operational sustainability. Since the companies consider profitability as a primary concern, any change in freight rates, fuel price or market conditions will change slow steaming rate. Speed reduction can be applied, until cost savings are still achievable. Different rates of slow steaming, variable voyage speeds, which are operated under design speed of ship have to be investigated in a technical perspective by considering ME and auxiliary machines.

The relation between ship speed and the power required for the propulsion is nonlinear function. In this regard drag force can be expressed with the following function;

$$\text{Drag Force} = \mathrm{C_1 v^2 + C_2 v + C_3} \tag{3.1}$$

Where the C_1 C_2 and C_3 values are constant and comes from regression analysis for a specific ship. The relationship among design speed and actual speed can be used to estimate relationship among power requirements for the propulsion.

$$\mathrm{P_{actual}/P_{design}} = \left(\frac{\mathrm{v_{actual}}}{\mathrm{v_{design}}}\right)^{\alpha} \tag{3.2}$$

The value of speed coefficient α can be between 2.5 and 3 (Moreno-Gutiérrez et al., 2015). The ratio of actual power and design power , L_R (Load Ratio), is used to estimate specific fuel oil consumption (SFOC) of ME (Moreno-Gutiérrez et al., 2015).

$$\mathrm{SFOC_{actual}} = \mathrm{SFOC_{ratio}} \text{ x } \mathrm{SFOC_{base}} \tag{3.3}$$

$$\mathrm{SFOC_{ratio}} = 0.455\mathrm{L_R}^2 - 0.71\mathrm{L_R} + 1.28 \tag{3.4}$$

$SFOC_{base}$ is the specified fuel consumption at designed of a specific engine in the Equation 3.3.

Propulsion power demand reduces significantly at low engine loads, thereby fuel consumption of ME decreases. The propulsion characteristics of a ship as; total propulsion efficiency (nm/ton), specific fuel oil consumption (SFOC - g/kWh) and ME load (%) variation were observed with respect to vessel speed under slow steaming conditions. The total propulsion efficiency means that the sailing distance in nm (nautical miles) for per unit fuel consumption at the specified vessel speed.

The specifications of the ship are given in Table 3.1.

Table 3.1: Technical specifications of the ship Specifications.

Ship Specifications		
Ship Type	:	Container Ship
Length Overall (LOA)	:	295 m
TEU	:	4200 TEU
Breath	:	32 m
Draught	:	12.6 m
Dead-weight	:	55,000 ton
Design Speed	:	25 knots
Engine Specifications		
Type	:	Sulzer RTA84C
Cylinder bore	:	840 mm
Piston stroke	:	2400 mm
Connecting rod length	:	2830 mm
Number of cylinders	:	12
Number of turbochargers	:	3
Continuous serv. rating of ME	:	48,600 kW
Corresponding engine speed	:	102 RPM
Mean indicated pressure	:	17,9
Propeller type	:	Fixed
Number of propeller blade	:	5
Propeller pitch	:	0.9 P/D
Specific fuel oil consumption	:	171 g/kWh

The data of ship specifications and operational parameters were obtained by using Kongsberg realistic Full-mission Ship Engine Room Simulator (ERS) (Kongsberg Maritime, 2019). The engine model of ERS is a replication of actual ship operating data and specifications which represents realistic responses for the engineering applications (Kongsberg Maritime, 2019). The ERS is used as a data source for marine engineering studies (Chybowski et al., 2015; Dimitrios, 2012; Durmusoglu and Kocak, 2019; Durmusoglu, Kocak, Deniz, and Zincir, 2015; Kocak and Durmusoglu, 2018; Ozturk et al.,, 2016).

Design speed of the investigated ship is 25 knots. By reducing vessel speed, 25 knots to 15 knots total propulsion power demand had been reduced. Figure 1 shows that the operating parameters of investigated ship as propulsion efficiency, specific fuel oil consumption and main engine load variation as a function of speed.

Speed reduction is implemented to the ship by considering aforementioned studies for optimum speed determination and IMO report (IMO, 2014). The speed of 14 knots is stated as the minimum effective speed for the slow steaming condition and the 19.5 knots is the speed that propulsion efficiency dropping slope changes as shown in the Figure 1. The speed “19.5 knots” was determined for the slow steaming speed for the investigated ship that the speed is very close to the most cost efficient speed range (Woo and Moon, 2014). Furthermore, 19.5 knots of vessel speed corresponds 45% of

ME load for the specified ship which is an acceptable power range for container ships. The ratio of voyage speed to design speed was 75% in 2012 which was 85% in 2007 for container ships (IMO, 2014). 19.5 k3nots is the 78% of the design speed of investigated ship.

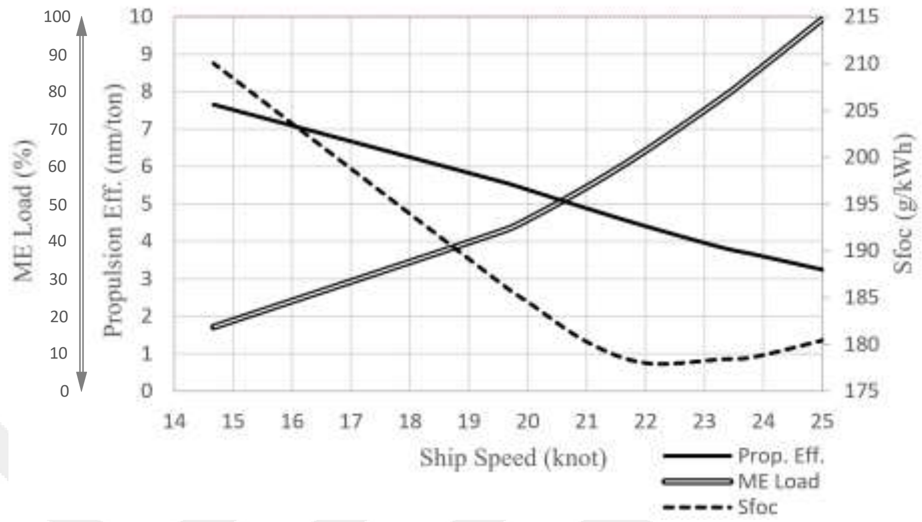

Figure 3.1: Fuel consumption, propulsion efficiency and ME Load as a function of speed.

Figure 3.1 indicates that, specific fuel oil consumption increases at the out of optimum vessel speed 22 knots. The total propulsion efficiency increases at low speeds. Drag force and main engine power demand reduce significantly with the lower vessel speeds because of the intrinsic phenomena of the ship propulsion phenomena.

With the reduction of the main engine load, many operational parameters change in the engine as fuel consumption, amount of generated emissions, dynamical stresses and particularly combustion parameters. Combustion temperatures and combustion pressures decrease with low loads. Exhaust gas energy which is utilized by turbocharger (T/C) and exhaust waste heat recovery system (WHRS) is reduced with the lower ME loads. On the other hand, power demand of auxiliary systems remains constant while propulsion demand reduces. The capacity of WHRS is designated by considering design operating loads with optimum exhaust amount rate and temperature. The WHRS coupled with Turbo Generator (TG), has a capability to meet the electrical power requirement of the investigated ship in the designated conditions. However, in the reduced ME loads, exhaust heat energy is not sufficient for the generation of required electrical power itself. The amount of utilized waste heat power to provide superheated steam to the turbo generator for the generation of electrical power by the exhaust gas boiler can be described by the Equation 3.5.

$$\dot{Q}_{exh} = \dot{m}_{exh}\, C_{p_{exh}}\, (T_{exh} - 433) \tag{3.5}$$

433 K is the temperature of minimum permissible temperature to avoid sulphuric acid condensation in the funnel (Deniz, 2015; Koroglu and Sogut, 2017) for the usage of Heavy Fuel Oil (HFO-ISO 8217) which contains 3.50% m/m sulphur content. The equation 5 indicates that, exhaust mass flow rate is crucial parameter as the exhaust temperature in terms of exhaust heat extraction. ME load reduction causes decrease in mass flow rate of air inlet and exhaust amount through the engine which also means reduction in exhaust heat, utilized in waste heat recovery system.

Reduction rates of exhaust mass flow rate and variation of exhaust temperatures are demonstrated in Figure 3.2(a) and $\dot{Q}_{exh}$ recovered heat from exhaust gas in Figure 3.2(b). $\dot{Q}_{exh}$ is calculated by using the equation 3.5. $C_{p_{exh}}$ is assumed as 1.0 kj/kgK for the combustion products (FLSmidth, 2019). The all $\dot{Q}_{exh}$ heat cannot be converted to electrical energy in turbo generator (TG). A significant amount of heat energy is dissipated to environment in the condensation process (Koroglu and Sogut, 2018). Figure 3.2(c) illustrates that the amount of the heat energy utilized by TG for electrical energy production and the total electrical power requirement for the ship systems.

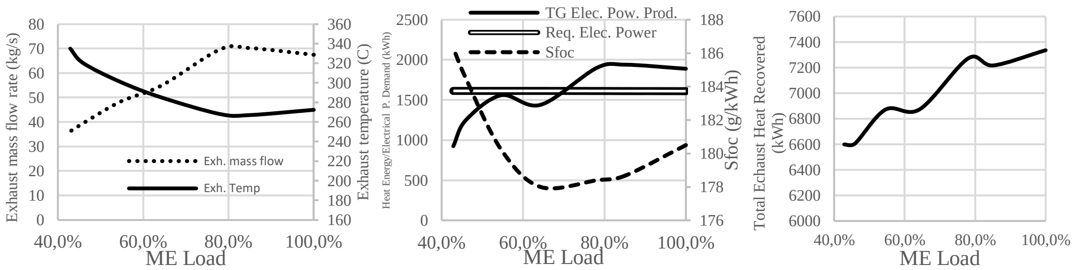

Figure 3.2: (a) Variation of exhaust flow rate and temperature, (b) Amount of converted exhaust heat energy to electrical power in TG, electric power requirement and SFOC of main engine as a function of ME Load. (c) Amount of recoverable energy from exhaust gas.

The recoverable heat from exhaust gases increases at the range of 85-80% and 65-55% of ME loads despite the decrement in total fuel consumption as a result of exhaust temperature and SFOC variation. Amount of total heat difference extracted from exhaust gas is accounted for 50% of the energy which is utilized for electricity generation by TG. The value of extractable exhaust heat degradation is 900kW for the investigated ship. At the point of 70% ME load, WHRS barely covers the electrical power demand which also includes reserve power. At the lower loads, heat difference is compensated by oil fired boiler and the other part of electrical demand is provided by diesel generators. Specific fuel oil consumptions of diesel generator and oil fired boiler are 225 g/kWh and 305 g/kWh diesel fuel respectively, which cause large amount of fuel consumption. In this regard, cutting down of auxiliary system electrical energy demand is an important subject to enhance energy efficiency on the ship. Since the pumps are main energy consumers of a ship power plant, main engine cooling system optimization is the worth-while solution to reduce energy consumption of auxiliary machinery in the case of low engine loads.

3.3 Load Optimization of Central Cooling System

Cooling of ME and auxiliary equipment is handled by central cooling system in the ship. There are two main streams in the system as sea water and fresh water flows. The ME is cooled by High Temperature Fresh Water (HTFW) system and other auxiliary coolers are cooled by Low Temperature Fresh Water (LTFW) system. The cooling fluid, fresh water, is cooled by the sea water. The rejected heat is given to environment via sea water. Fresh water cools down scavenge air, ME lubricating oil, air compressor, T/C and TG lubricating oil (other coolers) in the first step of cooling circulation. After the first step, relatively high temperature fresh water is pumped to cool ME as a Jacket Water (JW) that the system is called as HTFW system. The inlet temperature of JW is adjusted by a three-way mixing valve to control outlet temperature of ME. After the cooling process, excess heat is dissipated to sea water via fresh water coolers. Sea water and fresh water flows provided by two main pumps in parallel with a small sized auxiliary pump for each one. The representation of ME central cooling system is given in Figure 3.3.

The temperature of fresh water supplied to these systems, is adjusted by three way mixing valve. The detailed description of mixing valve and cooler is shown in Figure 3.4. The outlet temperature of fresh water is adjusted as 32 °C after coolers via mixing valve for the investigated ship. To adjust set temperature, a required amount of heat must be extracted from ME and related systems. If a required cooling demand is lower than the cooler capacity by the reason of slow steaming, fresh water bypasses the cooler and mixes with the cooled part of stream which is coming from the cooler to control temperature.

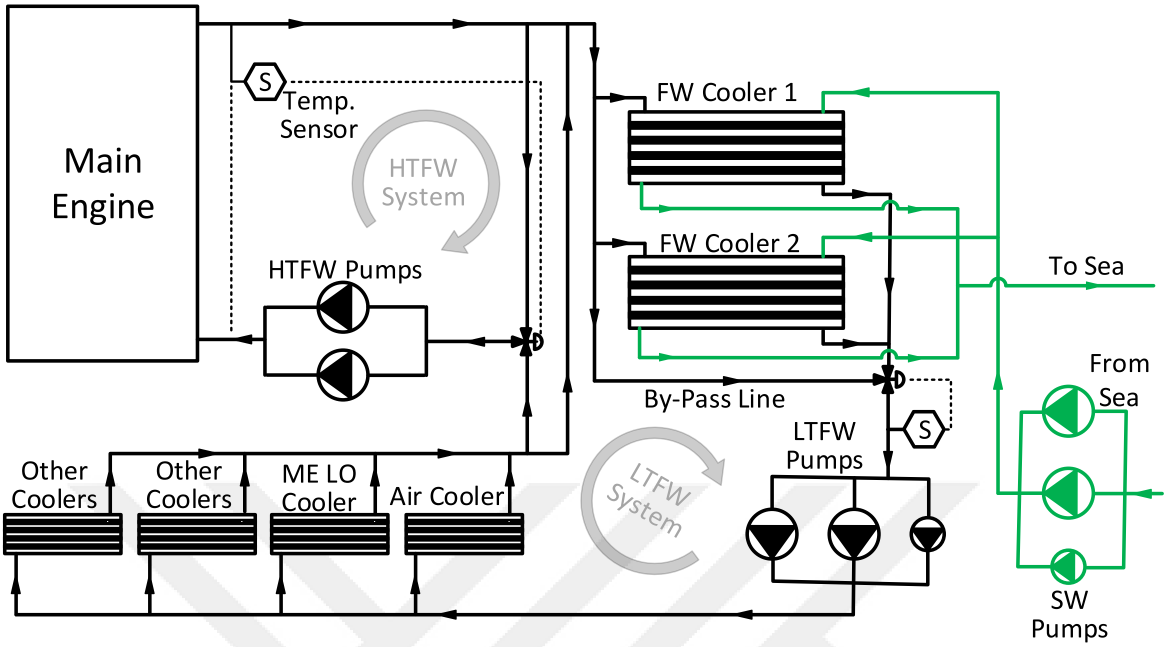

Figure 3.3: Representation of the central cooling system of marine power plant.

Since the cooling demand reduces, fresh water flow in the bypass line increases and heat transfer demand reduces in fresh water cooler which leads excess power consumption to pump excess flow rates. The excess flow rates' optimisation for both streams are the main concern must be performed in this study. Two main sub-model must be developed to achieve the optimisation as fresh water cooler model and pump model.

The cooler model is developed for the purpose of calculation of flow requirements of sea water and fresh water streams according to heat dissipation of ME and other systems. The calculations are conducted with the model where the heat balance equation and Log Mean Temperature Difference (LMTD) approach are utilized in. The input parameters are inlet temperatures of fresh water and sea water streams and mass flow rates of them. The sea water inlet temperature is assumed as constant and the other parameters are both input and output values in the model. The calculations are carried out with an iterative approach explained in the study. Any variation at the heat balance parameters changes the any other output values in the equilibrium.

The required flow rates are determined with the model by controlling by-pass flow rate at minimum level and fresh water control temperature at set value. These control parameters are limitations for the model at different ship speeds. The demanded heat transfer rate in the fresh water cooler is expressed as a function of ME load by the manufacturer equations. The calculations of the manufacturer equations and collected ship data are compared in the study.

According to determined flow rates, required speed of pumps and their power consumptions are calculated with pump model which includes the energy equations, affinity law expressions and efficiency calculations of pumps. This can be achieved by changing conventional pumps to Variable Speed Pumps (VSP) with a load optimisation of them. Power consumptions of both type of pumps tabulated according to variable ME loads.

The underlying law of fresh water cooler is heat balance relation which is expressed by the Equation 3.6.

$$\dot{Q} = \dot{m}_{FW} C_{p\_FW} \left(T_{FW\_in} - T_{FW\_out}\right) = \dot{m}_{SW} C_{p\_SW} \left(T_{SW\_out} - T_{SW\_in}\right) \quad (3.6)$$

Where the $\dot{m}$, C_p, and T denote mass flow of fluid, specific heat and temperature, respectively. Subscripts FW and SW indicate fresh water and sea water systems respectively.

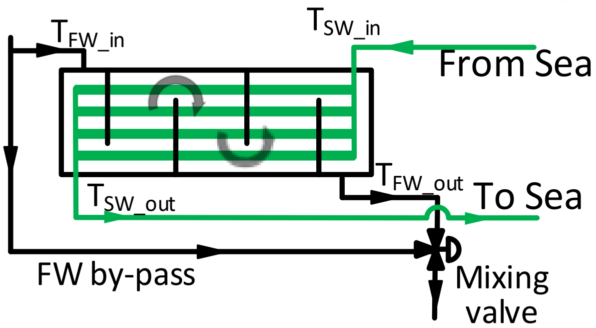

Figure 3.4: Detailed illustration of fresh water cooler.

Fresh water cooler is counter flow type heat exchanger in which the coolant fluid separated by a wall. Thermal calculations are conducted according to the following assumptions that heat transfer coefficient is constant throughout the cooler, the system is adiabatic and the temperatures and specific heats are constant in calculated cross sections. The following equations *(eq. 3.7 "a-b-c-d-e")*(Cui et al.,; Mistry and Misener, 2016) are employed to calculate outlet temperatures of fresh water and sea water.

$$Q = UA\overline{\Delta T} \tag{3.7a}$$

$$\overline{\Delta T} \equiv \Delta T_{lm} \tag{3.7b}$$

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \tag{3.7c}$$

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \tag{3.7d}$$

$$\Delta T_1 = T_{FW_{in}} - T_{SW_{out}} \tag{3.7f}$$

Q is heat transfer rate in kW, U is heat transfer coefficient and A is the transfer surface area in equations. U and A are constant for a specific cooler. They are calculated from the data obtained from ship operational parameters which are given in Table 3.2. An iterative approach is conducted to calculate two unknown outlet temperatures using equations 3.6 and 3.7 “a-b-c-d-e”. The mean effective temperature difference $\overline{\Delta T}$ is main variable which is described by log-mean temperature difference ΔT_{lm}. The method can be called as Log Mean Temperature Difference (LMTD) approach (Cartaxo and Fernandes, 2011; Cui et al., 2014; Mistry and Misener, 2016; Y. F. Wang and Chen, 2015).

The output values as sea water and fresh water outlet temperatures are used to optimize mass flow rates of the sea water and fresh water system. Required flow rates and output temperatures with corresponding pump parameters are given in Table 3.3.

At the designed speed operation of the vessel, a single fresh water cooler can cope with the cooling demand of both ME and auxiliary systems, while the other one is not on the operation. Cooling system is designed for the tropical conditions with extreme parameters as 110% MCR. In the navigation with reduced ME loads, combustion temperatures will reduce which affect cylinder wall temperatures. This effect is followed by low jacket water temperature and low lubricating oil temperature. As shown in Figure 3.2(a) that, exhaust flow rate, scavenge flow and its pressure decrease. Scavenge air cooling demand will also decrease.

Dissipated heat amount, from scavenge air, lubricating oil and cylinder liner to fresh water stream reduces significantly. The heat dissipation variation for the coolers and jacket water are expressed in MAN project guide (MAN Diesel and Turbo, 2010) with equations 8,9,10. The prediction of scavenge air cooler, jacket water cooler and lubricating oil cooler heat dissipation in terms of percentage of nominal value are given below.

Scavenge air cooler heat dissipation ratio ($Q_{air\%}$);

$$Q_{air\%} = 100\ (P_M/P_{L1})^{1.68}\ (n_M/n_{L1})^{-0.83} k_0 \qquad (3.8a)$$

$$k_0 = 1 + 0.27\ (1 - P_O/P_M) = 1 \qquad (3.8b)$$

Where the P is the Power, n is the rpm and subscript (M) is the specified MCR and (L1) is the nominal MCR.

Jacket water cooler heat dissipation ratio ($Q_{jw\%}$);

$$Q_{jw\%} = e^{(-0.0811\ \ln(n_{M\%}) + 0.8072\ \ln(P_{M\%}) + 1.2614)} \qquad (3.9a)$$

$$P_{M\%} = \frac{P_M}{P_{L1}}\ 100\% \qquad (3.9b)$$

$$n_{M\%} = \frac{n_M}{n_{L1}}\ 100\% \qquad (3.9c)$$

Lubricating oil cooler heat dissipation ratio ($Q_{lub\%}$);

$$Q_{lub\%} = 67.3009\ \ln(n_{M\%}) + 7.6304\ \ln(P_{M\%}) - 245.0714 \qquad (3.10)$$

The heat dissipation rates which are obtained by means of ERS (ERS's specifications are given in Table 3.1) and MAN project guide equations' results (3.8, 3.9, 3.10) (MAN Diesel and Turbo, 2010) are compared. The collected data and calculations from the equations show good agreement with each other. The variations of cooling loads are demonstrated as a percentage load of maximum operating conditions in Figure 3.5.

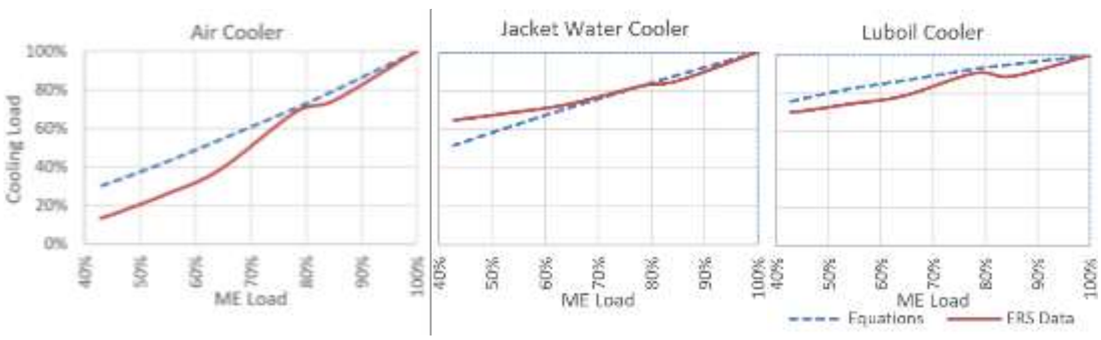

Figure 3.5: Amount of heat must be removed from coolers.

Figure 3.5 indicates that, cooling load of air cooler, and lubricating oil cooler are reduced with the low engine loads. HTFW system, including ME jacket water, and LTFW system is cooled by sea water system. The cooling load of both sides of FW and SW cooler reduces by slow steaming. In such a case, considering the pumping costs of the cooling system, optimizing of sea water and fresh water flow rates will provide an advantage in the energy consumption of the auxiliary machines

The cooling system is not operated at full capacity at the conditions which except challenging conditions. The cooling demand of the system diminishes and fresh water bypass flow rate is increased by temperature control system during the slow steaming operation. Since the pumps are the main electrical energy consumers on ships, to adjust the speed of the pumps will be a great advantage in terms of energy efficiency. The operational characteristics as power consumptions of the fixed speed fresh water and sea water pumps and flow rates during navigational speed and during slow steaming operations are presented in the Table 3.2. The investigated operating loads of ships (% ME load) have been determined by considering container vessel average speeds stated in IMO report (IMO, 2014).

Table 3.2: Pump and flow characteristics of central cooling system with fixed speed pumps.

	Parameter Name	Unit	FIXED SPEED DRIVE					
	ME LOAD **Ship Speed**	**%** **knots**	**100%** 25,1	**85%** 23,9	**78%** 23,3	**65%** 22,0	**55%** 21,0	**45%** 20,0
SEA WATER SIDE	SW Pump Flow	toh/h	2126	2126	2126	2126	2126	2126
	SW Flow to Fresh Water Coolers	toh/h	1378	1378	1378	1378	1378	1378
	SW Pump Speed	rpm	1800	1800	1800	1800	1800	1800
	SW Pump Power	kW	191,3	191,3	191,3	191,3	191,3	191,3
	SW Pump Head	bar	1,89	1,89	1,89	1,89	1,89	1,89
	SW temperature inlet SWC	degC	20	20	20	20	20	20
	SW temperature outlet SWC	degC	34,38	31,45	30,2	27,81	26,55	25,62
FRESH WATER SIDE	FW Flow to FWC line	toh/h	1378,3	1345,6	1329,85	1311,74	1299,92	1285,68
	FW Flow to FWC	toh/h	1039,82	837,98	740,67	604,95	521,81	461,04
	FW Flow to by-pass line	toh/h	338,26	507,31	579,83	703,8	771,64	832,66
	FW Pump Speed	rpm	1800	1800	1800	1800	1800	1800
	LTFW Pump Power	kW	155	155	155	155	155	155
	LTFW Pump Head	bar	2,75	2,75	2,75	2,73	2,73	2,75
	FW temperature inlet FWC	degC	46,3	43,6	42,3	40,2	38,9	38,0
	FW temperature outlet FWC	degC	27,4	25,0	24,5	22,5	21,8	21,3
	FW temperature after mixing	degC	32	32	32,06	32	32	32

Table 3.2 indicates that, sea water outlet temperatures become cold by means of sea water cooler and fresh water by-pass line flow rates increase with the ME load decrement.

Utilizing of fixed speed pumps cause excessive flow rates with a great power consumption. Sea water and fresh water pumps are consuming same power rates individually in all ME loads. The rate of power consumption is excessive in terms of energy efficiency. The central cooling system is considered in order to cut down electrical power demand at slow steaming conditions.

According to Table 3.2, more than 50% of fresh water flow by-passes the cooler due to the diminished cooling demand at the 65% and lower loads. It is applicable to use variable speed pumps to reduce the excessive flow rate. Utilizing cooling systems with frequency-controlled pumps lead reduction in electrical power demand. In the case of slow steaming operation, cooling demand of main engine with fresh water cooler which is combined with sea water cooling system operates with over capacity. Frequency controlled pumps reduce energy demand by 60%. When it is compared with the total energy consumption of a ship, the saving is very limited as 0.1-0.6%. However, this application compensates its initial investment cost in a short period (Bertram and Taşdemir, 2017). The mentioned saving rates are valid for different range of navigational speeds. In the case of slow steaming saving percentages are much more than the stated values. The savings in pumps vary depending on the operating power and efficiency of the pumps. The efficiency of a pump can be described indirectly by the equation 3.11.

$$P = \rho \text{ x } g \text{ x } \dot{m} \text{ x } H \,/\, \eta_p \tag{3.11}$$

Mass flow rate "$\dot{m}$", Pressure increase (head) "H" and power consumption "P" of pump is directly related with pump speed. η is the pump efficiency that consists of pump and electrical motor efficiency. Fixed speed pump's efficiency is heavily depended on its differential pressure between suction and discharge sides and mass flow rate. Variable Speed Pumps' (VSP) efficiency varies as a function of their rotational speed, by changing the pump speed, flow rate and head can be adjusted. Efficiency correction factor should be involved in the calculations which is described by pump speed variation from optimum operating point. By considering the variation of VSP's speed never exceeds 40%, the correction factor is employed at the calculations. The relation of the parameters as a function of pump speed is given below (equations 3.12"a-b-c"); which is known as affinity laws. For the high power centrifugal pumps (>90 kW), efficiency prediction of affinity laws is very close to the measured efficiency values (Simpson and Marchi, 2013).

$$\dot{m}_1/\dot{m}_2 = (N_1/N_2) \tag{3.12a}$$

$$H_1/H_2 = (N_1/N_2)^2 \tag{3.12b}$$

$$P_1/P_2 = (N_1/N_2)^3 \tag{3.12c}$$

When the speed of pump increases by 10%, the increments of pressure and power become 21% and 33% respectively. Using variable speed pumps instead of fixed speed pumps allows to control flow rate. In the case of using fixed speed pumps alone, flow rate adjustment is managed by throttle valves and this causes excessive usage of pumps. A sample of fixed speed and variable speed pumps' operations are demonstrated in Figure 3.6.

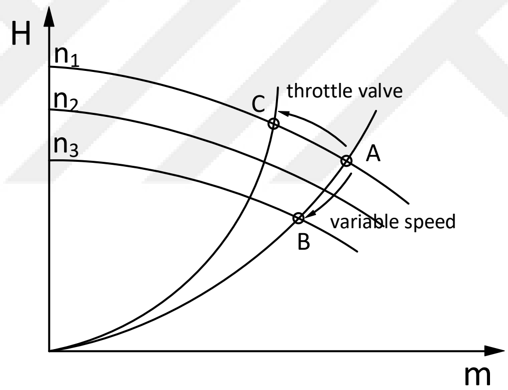

Figure 3.6: Changing of system curve in the case of fixed speed pump and variable speed pump operations.

The pump efficiency remains constant at the operation curve of A to B for variable speed pump while the system pressure and pump efficiency change from the point A to C for fixed drive.

SW pumps and FW pumps can be considered as high power demanding pumps which need 191 kW and 155 kW for the investigated ship, respectively. Affinity laws can be employed to estimate new parameters of the pumps. It can be expected that small deviations may occur at the efficiency rate. Sarbu and Borza (1998) proposed an equation for the calculation of new efficiency of the VSPs (Equation 3.13).

$$\eta_{p2} = 1 - (1 - \eta_{p1}).\left(\frac{N_1}{N_2}\right)^{0.1} \quad (3.13)$$

Efficiency deviation becomes greater in the case of large amount of speed variations. However, it is indicated that high rates of pump efficiency until to 70% keeps its efficiency deviation at minimum levels (i.e: 5%). Besides, the equation is not convenient for small sized pumps and the pumps which have more than 40% speed variations.

3.4 Results and Discussion

According to calculations carried out with thermodynamic equations and affinity laws, new flow requirements are specified for the slow steaming conditions. ME and auxiliary cooling loads are the critical parameters for the optimization. The fact that, the pump's speed variation does not exceed by more than 40% is an important consideration in the determination of pump speeds, by the reason of efficiency variation. Calculations are carried out by using the equations 6 to 13. Adjusted pump speeds and their flow rates are presented in the Table 3.3. The temperature of sea water which enters to fresh water cooler was assumed as 20 ℃ and the fresh water temperature is adjusted as 32 ℃ after mixing valve.

Table 3.3 shows that even if the main engine operates at the full load, variable speed pump adjustment can offer saving on the power of both the sea water and fresh water pumps. As the ME load decreases, the amount of both sea water and fresh water circulation flow should be decreased to minimize by-pass flow mass. As indicated before, this flow could be adjusted by variable speed pumps.

As shown in the Table 3.3, fresh water and sea water flow rates could be diminished by using variable speed pumps. The nominal speed of pumps is 1800 rpm for both systems. The two systems have different flow rates because of different operating temperature ranges and their own cooling responsibilities. Power consumptions of sea water pumps and fresh water pumps are shown in the Table 3.3. Selected variable speed pumps' power consumptions are higher than conventional ones at the nominal speed which are 241,9 kW and 196,2 kW for the sea water and fresh water pumps, respectively. However, when challenging conditions are not at stake, nominal speed flow rates are not required.

Table 3.3: Operating characteristics of central cooling system with variable speed pumps.

	Parameter Name	Unit	VARIABLE SPEED DRIVE					
	ME LOAD	**%**	**100%**	**85%**	**78%**	**65%**	**55%**	**45%**
	Ship Speed	**knots**	25,05	23,86	23,28	22,01	21,01	20,01
SEA WATER SIDE	SW Pump Flow	toh/h	1889,8	1712,6	1594,5	1476,4	1299,2	1133,9
	SW Flow to Fresh Water Coolers	toh/h	1141,8	964,6	846,5	728,4	551,2	385,9
	SW Pump Speed	rpm	1600	1450	1350	1250	1100	960
	SW Pump Power	kW	175,2	131,4	106,7	85,2	58,7	39,4
	SW Pump Head	bar	1,89	1,56	1,35	1,16	0,90	0,68
	SW temperature inlet SWC	degC	20	20	20	20	20	20
	SW temperature outler SWC	degC	37,5	36,5	36,4	35,0	36,5	40,1
FRESH WATER SIDE	FW Flow to FWC line	toh/h	1225,2	957,2	918,9	842,3	765,7	735,1
	FW Flow to FWC	toh/h	1055	670	605	485	425	460
	FW Flow to by-pass line	toh/h	170,2	287,2	313,9	357,3	340,7	275,1
	FW Pump Speed	rpm	1600	1250	1200	1100	1000	960
	LTFW Pump Power	kW	139,0	67,5	59,9	46,4	35,2	31,2
	LTFW Pump Head	bar	2,75	1,68	1,55	1,30	1,07	0,99
	FW temperature inlet FWC	degC	48,1	48,4	47,0	44,7	43,8	42,5
	FW temperature outlet FWC	degC	28,7	25,0	24,3	22,6	22,6	25,9
	FW temperature after mixing	degC	32	32	32	32	32	32

The first column of the Table 3.3 depicts flow characteristics at 100% of ME load. At the operating point of 100%, sea water pump and fresh water pump speeds can be adjusted as 1600 rpm to meet the cooling demand. According to calculations, the adjusted flow rates are lower than pumps' nominal flow rates. As a result of less power produced by ME, cooling demand becomes lower. In the other reduced ME load cases namely 85%, 78%, 65%, 55% and 45%, sea water pump operates at 1450, 1350, 1250, 1100 and 960 rpm while fresh water pump operates at 1250, 1200, 1100, 1000 and 960 rpm respectively. Accordingly, a reduction in requirement of pumping costs can be achieved. Pumping power requirement could be decreased to 39.4 kW for sea water pump and 31.2 kW for fresh water pump at 45% of ME loads. The variation of power requirements for SW and FW pumps are demonstrated at Figure 3.7.

The power consumption difference which is accounted for energy saving is calculated. The input data used for the potential fuel saving calculations has been obtained by ERS. The total saving capacity of the system is calculated by considering ME load. The pumping power reductions are 103 kW, 194.4 kW, 219.7 kW, 246.8 kW, 276 kW and 293.8 kW for the 100%, 85%, 78%, 65%, 55% and 45% ME loads. The reduced electrical power demand and the variation of electrical power generation as a function of ME load are illustrated in Figure 3.8.

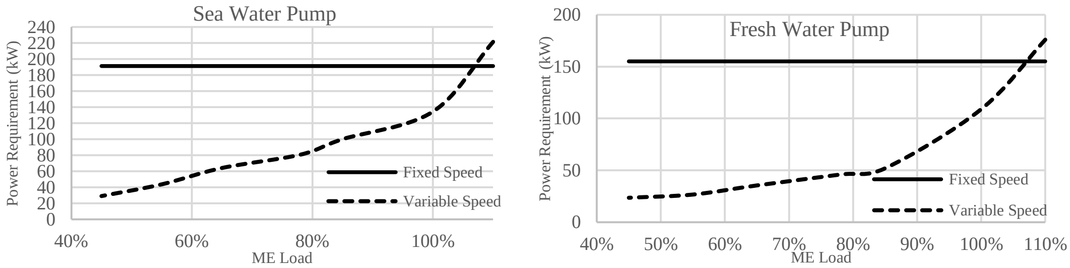

Figure 3.7: The variation of power requirements of pumps as regards to ME loads for fixed and variable speed pumps (a) SW pumps (b) FW pumps.

Electrical power reductions of pumps are 103kW/h, 194kW/h, 219kW/h, 247kW/h, 276kW/h and 294kW/h which correspond 0.3%, 0.7%, 0.8%, 1.1%, 1.4% and 1.8% of total propulsion efficiency shown in Figure 2 for the 100%, 85%, 78%, 65%, 55% and 45% engine loads respectively.

Assuming that the waste heat system is utilized to generate electrical power of the ship, WHRS barely meet the requirement of all electrical power demand of ship systems at 70% ME load when cooling system is utilized by conventional (fixed speed) pumps.

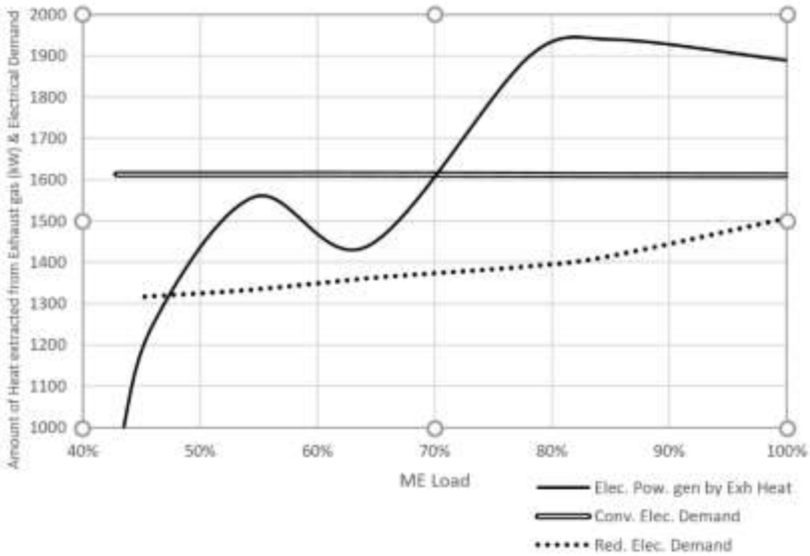

Figure 3.8: Reduced electrical power demand and the amount of electrical power generation by utilizing exhaust heat.

In the case of using variable speed pumps, WHRS with 47% of ME load could meet the electrical power demand of the ship systems because of the reduced electrical power demand as shown in Figure 3.8. This would reduce additional fuel consumption and provides a great advantage in slow steaming conditions. The saving rate increases at low engine loads because of low cooling demand. 65% and lower ME loads are considered as slow steaming operation and in the study, and 80% of the operational profile is assumed as slow steaming operation. Since all the required electricity is supplied by exhaust heat recovery system at the navigational speeds of 70% or higher ME loads as shown in Figure 8, calculating the energy saving of electrical consumption is not required in these conditions for the investigated ship with WHRS.

Electrical power requirement of the ship's auxiliary systems is 1610 kW for the ship and it includes 150 kW reserve power. By decreasing of operating power of pumps, the values could be reduced. However, exhaust heat capacity also reduces with the reduced loads. Additional electrical power demands are calculated in the slow steaming conditions given in Table 3.4. The power demand difference is compensated by oil fired boiler or diesel generator. While the diesel generator provides an advantage in efficiency (225g/kWh), the working load limitation eliminates this advantage. 15% of operating load specified as a minimum continuous operating load for 1800kW power diesel generator.

Generation of electricity via supplying oil fired boiler has an advantage in 65% to 50% ME loads where the power demand is under 180kW, because of the constraint about the minimum operating load of diesel generator. Oil fired boiler efficiency is lower than the diesel generator efficiency, thus when power requirement increases the boiler may consume more fuel. Therefore, when the power difference exceeds 180 kWh, diesel generator becomes advantageous. The calculations were carried out via using oil fired boiler alone, diesel generator alone and combination of two systems in which ship WHRS is utilized. Additional power requirements are 160 kW, 150kW, 50kW, 180kW and 430kW for the 65%, 60%, 55%, 50% and 45 % of ME loads, respectively. The amounts of fuel used to meet electrical power requirement of the system are 48.8 kg, 45.8 kg, 15.3 kg, 54.9 kg and 103.5 kg for the mentioned ME loads. In the case of unavailability of the WHRS, diesel generator operation is conducted to meet electrical power requirement. Since the all required electricity is supplied by diesel generators, power demand of variable speed pumps have to be considered in all speeds. Pumping power reduction has favourable effect on diesel generator power generation and fuel oil consumption which is indicated in Table 3.3.

Operating profile is determined as indicated in Table 3.3. Navigational operation assumed as 300 days per year and by considering operating profile assumptions, the reduction in electrical energy usage, has a great advantage in fuel consumption and CO2 emission generation.

Table 3.4: Electrical power demand variation of ship systems and compensation cost.

	ME LOAD	%	85%	75%	65%	60%	55%	50%	45%
	Ship Speed	knots	23,86	23,00	22,01	21,48	21,01	20,50	20,01
	Annual Ship Operating Profile	%	10%	10%	10%	10%	15%	15%	30%
	Elec. Power Req. with conventional pumps (with 150 kW reserve power)	kW	1610,0	1610,0	1610,0	1610,0	1610,0	1610,0	1610,0
	Elec. Power Req. with variable speed pumps (with 150 kW reserve power)	kW	1415,6	1390,3	1363,2	1350,0	1334,0	1324,8	1316,2
WITH WHRS	Electrical Power Production using Exh. Heat Recovery	kW	1940,0	1800,0	1450,0	1460,0	1560,0	1430,0	1180,0
WITH WHRS	Additional Electrical Power Requirement with conventional pumps	kW	0,0	0,0	160,0	150,0	50,0	180,0	430,0
WITH WHRS	Additional Electrical Power Requirement with variable Speed pumps	kW	0,0	0,0	0,0	0,0	0,0	0,0	137,0
WITH WHRS	Additional fuel amount to compansate electrical power demand with oil fired boiler	kg/h	0,0	0,0	48,8	45,8	15,3	54,9	131,2
WITH WHRS	Additional fuel amount to compansate electrical power demand with diesel generator	kg/h	0,0	0,0	65,0	65,0	65,0	65,0	103,5
WITH WHRS	Combination of electrical power generation system	kg/h	0,0	0,0	48,8	45,8	15,3	54,9	103,5
WITH WHRS	Marine Gas Oil consumtion cost	$/h	0,0	0,0	31,2	29,3	9,8	35,1	66,3
WITH WHRS	CO2 emission	kg/h	0,0	0,0	156,2	146,4	48,8	175,7	331,3
WITH WHRS	Fuel consumption with conv. pumps	ton/year	367,4						
WITH WHRS	Total Saving in fuel with WHRS system	ton/year	296,2						
WITH WHRS	Fuel consumption after optimisation	ton/year	71,2						
WITHOUT WHRS	Power Requirement reduction with variable speed pumps	kW	194,4	219,7	246,8	260,0	276,0	285,2	293,8
WITHOUT WHRS	Fuel saving with diesel generator due to reduced electrical demand	kg/h	43,7	49,4	55,5	58,5	62,1	64,2	66,1
WITHOUT WHRS	Marine Gas Oil consumtion cost	$/h	28,0	31,6	35,5	37,4	39,7	41,1	42,3
WITHOUT WHRS	CO2 emission	kg/h	140,0	158,2	177,7	187,2	198,7	205,3	211,5
WITHOUT WHRS	Fuel consumption with conv. pumps	ton/year	2608,2						
WITHOUT WHRS	Total Saving in fuel without WHRS sys	ton/year	428,3						
WITHOUT WHRS	Fuel consumption after optimisation	ton/year	2179,9						

The fuel consumption is calculated by considering diesel generator specific fuel oil consumption as 225 g/kWh. The sfoc variation is implemented via equations 3.3 and

3.4 as a function ME load. The following total fuel saving with WHRS is 296.2 tons per year and 948 tons of CO2 emission reduction which is calculated by using 3.206 t_{CO2}/t_{MDO} emission factor (IMO, 2012b). The total cost saving is $207,300 annually. The saving rates are much more for the system which is not utilized by WHRS as 428.35 tons of fuel saving 1,336.4 tons of CO_2 emission and $299,600 cost saving.

3.5 Conclusion

Ship's cruising speeds have been reduced in order to satisfy IMO regulations in the recent years. The paper points to the conclusion that implementation of additional technical solutions for energy efficiency gain in importance at efficient speed operations particularly. The first law of thermodynamics based calculation of central cooling system is carried out under slow steaming conditions. The necessity of optimization of auxiliary systems has emerged to enhance energy efficiency. Because the systems are designed to cope with the heavy operating conditions as tropical climate and the conditions which require maximum engine load of ship main and auxiliary engines. The cooling loads of ME with jacket water, scavenge air, and lubricating oil which constitute large part of total cooling demand are analysed in different ship speeds. The cooling demand was calculated with using ERS ship data and verified with MAN two-stroke marine diesel engine project guide assumptions. The total cooling demand decreases by 55%. According to the obtained data and calculations, sea water and fresh water flow rates are adjusted by considering system operating temperatures with using variable speed pumps. Calculations were carried out with using affinity laws and given equations for the variable speed pumps in the literature. All the required electricity is supplied by exhaust heat recovery system at the navigational speeds of 70% or higher for the investigated ship. In the case of slow steaming electrical energy deficit is compensated by oil fired boiler and diesel generators. The power reduction saved by pumps, decreases a certain amount of Marine Diesel Oil (MDO) used by these systems. An assessment of the annual energy consumption saving of the system was presented. 80% of the operating time of the ship assumed as a slow steaming operation. A noteworthy amount of power saving and accordingly, cut down of fuel consumption and emission release could be achieved. 296.2 tons of fuel is saved by utilizing variable speed pumps in slow steaming operation which also means that the $207,300 saving and 924 tons of CO_2 emission reduction. The amount of saving increases in ships which are not equipped with WHRS by the reason that all electricity is supplied by diesel generators. The paper has proved that additional savings can be achieved if such equipment utilized. The application can be applied not only to container ships, which was subjected here, but to other ship types as well.

Reduction of heat radiation and air consumption of ME in slow steaming conditions results in a reduction in the amount of air to be pumped into the engine room. The engine room fans speed variation or partial operation of the fans increase the amount of savings which can be considered as an additional complementary application for slow steaming conditions.

4. ENERGY EFFICIENCY BASED OPERATION OF COMPRESSED AIR SYSTEM ON SHIPS TO REDUCE FUEL CONSUMPTION AND CO_2 EMISSION[3]

4.1 Introduction

Energy efficiency has been very popular subject for many years in maritime transportation as well as in industrial areas. The energy efficiency policy aims to both reduction of energy costs in whole operation and decrement of hazardous emissions emitted to atmosphere. Restricted emission limits in maritime sector, governed by International Maritime Organization (IMO) and Marine Environment Protection Committee (MEPC), prompted operators to pay attention on this subject. Fuel consumption in the operation is evaluated by Energy Efficiency Operational Indicator (EEOI) (IMO, 2009). This issue starts with the beginning of design stage of ships that Energy Efficiency Design Index (EEDI) (IMO, 2012b) became a mandatory regulation for new ships together with the Ship Energy Efficiency Management Plan (SEEMP) (IMO, 2012c). The primary objective of these amendments is mitigation of emissions by reducing fuel consumption (IMO, 2011, 2019b). High level of fossil fuel usage to generate electrical energy or propulsion power in ships, makes efficiency consideration more vital phenomena for the both economic and environmental aspects. Each action which can be done while considering fuel saving, has a reaction on emission reduction. It would both give great advantages on mitigation of carbon emissions and preserving money accompanying with the fuel reduction. CO_2 as a greenhouse gas (GHG) has noxious impact, as climate change, on environment. It is shown in studies carbon finance will gain in importance in future (Chang, 2016). For this reason, there are some implementations of novel technologies to reduce emissions as usage of alternative

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[3] Dere C., Deniz C. 2019. Energy Efficiency Based Operation of Compressed Air System on Ships to Reduce Fuel Consumption and CO2 Emission Transactions RINA, Vol 161, Part A2, International Journal Maritime Engineering, Apr - Jun 2019 DOI No: 10.3940/rina.ijme.2019.a2.532

fuels (Deniz and Zincir, 2016) or renewable energy usage (Nuttall et al., 2014; Pflanz, 2000; US6610193, 2003) While occasional concept designs enable alternative and renewable energy usage on board, there are very limited samples in practice usage of renewable and alternative energy to generate electricity on ships.

Starting with the new and efficient design of ships, many other practices have been applied operations of aged ships to minimize fuel consumption such as, slow steaming, route optimization, on board efficiency managements, adaptation of novel technologies (Seddiek, 2015) and maintenance planning which cover overall metrics. All of these plans must extend main energy flow lode to capillaries. All possible and feasible efforts have to be undertaken at the whole ship equipment. Utilization of energy in an efficient way is the cheapest and rapid solution for efficiency issue. These solutions comprise, increasing efficiency of processes, production, conversion and conservation together with diminishing excessive consumption (Bilgen and Sarıkaya, 2016). In this context, all energy consumer equipment must be evaluated in term of efficiency concept. Since, the most economical energy is the energy you never waste, consumption and conservation of energy in an efficient manner are two main objectives which must be focused on in efficiency considerations (Bilgen and Sarıkaya, 2018). One of the most inefficient equipment is compressed air system in engine room by virtue of having natural ineffective process; nonetheless, starting air system is indispensable equipment for an engine room of a ship which is propelled by a large marine diesel engine. By increasing the pressure of the air, the starting air bottle is qualified to start to the reciprocating motion of large bore diesel engine by means of pneumatic way. Compressed air is a kind of energy storage method which uses potential energy of the gas. By both reason of the free charge of air before compression process and no contamination or penal sanction in the case of any leakage, a particular attention is not paid on this matter. However, after compression process the air is qualified with a pressure and gets a valuable expense. It must be emphasized that, pneumatic output power of compressed air only 5-10% of consumed electrical power (Brodyansky et al., 1994; Taheri and Gadow, 2017). Therefore, compressed air is a high cost energy source for all applications. Independent of the types of compressors not only efficiency level, but also control type, storage strategy and leakage volume of the system are also highly related with the total outcome of compressed air system efficiency. After all, misconception about cost of the air must be cleared up and efficiency measures must be taken into consideration in production, storage and utilization stages.

Many more studies carried out to reveal the compressed air costs in land industrial areas. This cost is highly depended on electrical costs of the company 0,15-0,25$ per kW (Taheri and Gadow, 2017). But in commercial ships the most convenient energy source is fuel oil, accordingly cost of generating electric power via diesel engines in ships much more expensive than land industries because of limited efficiency of internal combustion engines. In industrial areas, compressed air is cached as an underground energy storage, by the reasons of large scale, long life-time, low operational and maintenance costs, thereby it can be considered cheapest energy storage technology in terms of unit cost (Chen et al., 2009). Despite the industries have a great advantage of cost of electricity and feasible energy production aid as renewable energy usage that, several studies had been carried out to improve compressed air systems energy efficiency (Benedetti et al., 2017; Kaya et al., 2002; Luo et al., 2016). These systems are operating under desired profit widely to diminish risk potential of the system (Neale and Kamp, 2009). By taking into account these studies, it is essential to realize why this subject need more attention in maritime sector too. As a results of the studies, system efficiency is highly dependent on the isentropic efficiency, operating pressure and cooling performance of system (Luo et al., 2016). Operating pressure must be well-adjusted within optimum range for the efficient use of compressor. It directly affects isentropic efficiency. Furthermore, leakage rate is another factor for overall efficiency of the system.

This paper analyses impact of operating pressure range variation and leakage rate factors that have major impact on efficiency of the system together with being identifiable and adjustable by marine engineers, on power consumption of compressor and its equivalent fuel consumption with carbon dioxide emissions.

4.2 Analyzing and Modelling of Compressed Air System

The compressed air system enables to start main engine, auxiliary engines, emergency generator, and also enables to perform hydraulic and pneumatic tools as exhaust valves motion, cleaning filters or sea chest, back flash filter, cargo operation, horn, fire pumps or a control air, etc. Furthermore, many more technologies can be utilized by compressed air system, powered by service air. A simplified schematic representation of a compressed air system is given in Figure 4.1.

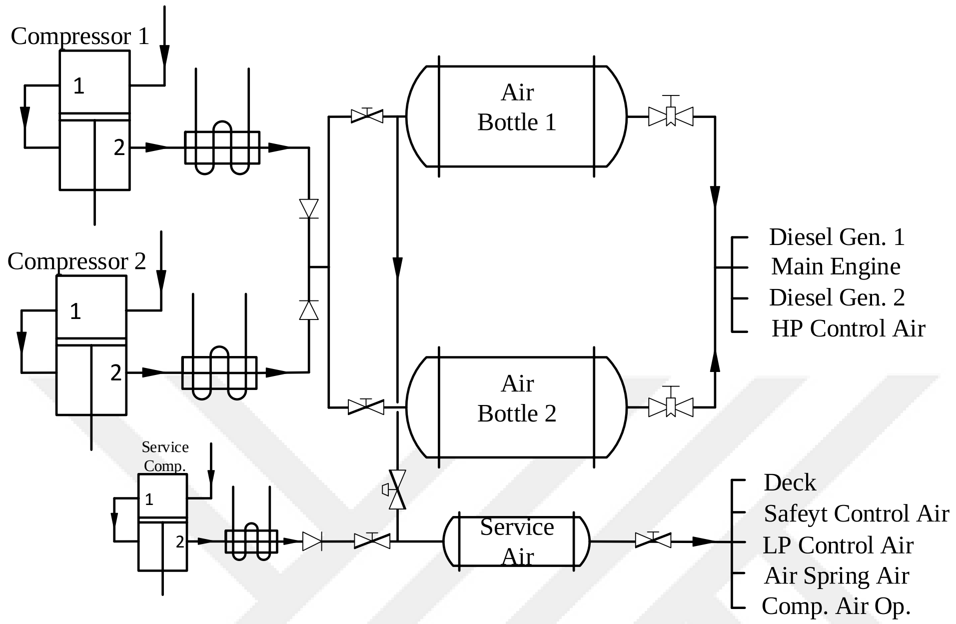

Figure 4.1: Schematic diagram of compressed air system in ships.

These abilities increase air consumption on board. Compressed air is consistently in use in any commercial ship, qualified and available at any time during the navigation. By the reason that, operating of appropriate systems by pneumatic way provides low initial cost and operating cost for them. Even in this case, operational energy cost is much more than initial and maintenance costs of compressed air system (Mousavi et al., 2014). The proposed evaluation method in this study realizes why attention must be paid to compressed air system optimization.

The compressor is operated on a ship to meet the air demand used by whole ship systems. The air is supplied by air bottles stored in by air compressors. Starting equipment (air bottles) rules have been specified by classification societies. Germanischer Lloyd, as a technical regulatory and supervisory authority, declared minimum requirements for starting air system, for instance, appropriate air bottle capacity, accepted minimum and maximum operating pressures (9-30 bars) and maximum demanded time to fill up air bottles at atmospheric pressure to maximum operating pressure (30 bars) (Lloyd Germanischer, 2015). Air bottles have safety pressure at 31 bars which has critical mission. Compressor operating condition is set at 30 bars. Whenever pressure of air bottle reaches 30 bars, the compressor is automatically shut down by controller, even a case, if automatically shut down control disabled safety valves opens at specified safety pressure. The system contains minimum two compressors and two air bottles except service air bottles and reduction valves. There is an uncomplicated system configuration for compressed air, and not many more system equipment for it. There is no peremptory rule to increase the efficiency of compressed air system, just an advice by MEPC, pay attention to minimize fuel consumption during operation (IMO, 2016). Keeping compressed air system in a good condition and operating at optimum range contribute energy efficiency matter on ships. The compressor efficiency is heavily depended on compressor operating load and decreases with the higher loads of compressor. The load changes with the air discharge pressure, which must be higher than tube pressure that, can be set to a specific value as a peak pressure to control. Operating range adjusted by control parameters and control method of the system. Naturally, control of the system affects the efficiency of the operation directly. Many type of control systems are in use in the compressor operation, according to air demand, minimum pressure limitation, and complexity of the system. Types of control systems are start/stop, load/unload, modulating inlet, variable displacement and variable speed controls. Most of these systems change the load on the compressor by considering discharge air pressure only except start/stop control type. Start/Stop control type is widely service in small-scale compressors. This type of control is the most energy efficient control type for these compressors, by the reason of start/stop control shuts off the compressor automatically when the pressure reaches the desired value. The conventional and the most convenient control system is this the type of control system in ships too, because of uncomplicated system requirement and controlled by operating ranges of pressure.

A virtual sample of any system can be simulated via computer program to demonstrate outputs and responses of the system, which must be close to real values. For a prosperous model, theoretical knowledge has to be combined with a mathematical model and experiences (Sapietova et al., 2017). In this study, compressor is modelled and operational process is simulated to reveal potential savings in the compressed air system operation. The process simulation includes air leakages, artificial demand and cooling process of compressed air. Furthermore, the modelled compressor parameters, comprises energy consumption, air flow, pressure and temperature values and these are also validated from the data obtained from ships. The results which are considering these issues and obtained from the model are demonstrated in this study.

4.2.1 Modelling of system dynamics

The unit energy cost of compressed air is depended on many factors as, storage pressure, compressor efficiency, cooler efficiency and the source of power generator which can be a shaft generator or diesel generator for a ship.

Modelled compressor algorithm is applied on a ship operation to observe the energy consumption of the system. Efficiency and any other outputs of the system could vary depending on compressor type, capacity, condition of the system, system requirements and operational control type. To calculate energy efficiency of the compressor, measurement of energy consumption and pressure trace of the air bottles are considered. With the relation of air bottle capacity and the pressure trace allow to calculate, how much air stored in air bottle. By considering trend of pressure increase, mass of air pressurized to air bottles can be calculated. Modelled system uses these relations and the power signatures to simulate the operational efficiency of the system. The manufacturer instruction manuals can be used for any type of compressor which will be used in ship for this assumption tool.

Ideal gas assumption is applied to compressed air systems to facilitate the calculation mass of the compressed air in the air bottle with a following fundamental equation.

$$P.V = m.R.T \tag{4.1}$$

Where the units are; P, pressure [kPa], V, volume [m3], m mass [kg], R individual gas constant of air 0,2871 [kJ/kg.K] which value obtained by dividing universal gas constant to molecular weight of air (〖R_u/M〗 _air) T, temperature [Kelvin].

Volume of a single air bottle can be 5.5 m^3 to 15 m^3 in existing commercial ships and varies ship to ship. Furthermore, as specified in Germanischer Lloyd (2015), maximum permissible operating pressure is 30 bars. In the study, these information was taken in to account for modelled system. All consumers are supplied from this system as service air, or engine starting air. Except from consumption another, a small amount of air consumed via leakage by piping, junction or holes. Air leakage is modelled as a function of pressure inside the air bottles or in system pipes. Leakage is an unavoidable issue in such systems (Cerci et al., 1995). Allowable leakage can be approximately 2-15% as a base leakage according to system capacity where in the system that maintained properly. However, in the case of poor maintenance, leakage, the undesired amount of air consumption, can reach 40-50% of system consumption capacity (Dindorf, 2012; Seslija et al., 2012) (Seslija et al., 2012, Dindorf, 2012). The basic amount of leakage can be accepted as a tolerable and unavoidable leakage. More than this value can be expressed as wasted air and depends on number of leakage holes and dimeter of holes, expressed by Compressed Challenge Inc., leakage is totally based on pressure and orifices areas (Compressed Air Challenge, 2004). For a considered system, the leakage rate is a function of a system pressure, written in Equation 4.2.

$$\dot{m}_{air\_leak} = f\left(P_{system}\right) \tag{4.2}$$

An estimated cost of air leakage can be calculated via this equation, as well as mass after detection of leakage points. In case of well-known of initial system parameters, as basic consumption and operating pressures, leakage trend can be observed clearly. Additionally, cost of leakages in the system in the specified time intervals can be projected from deviation of specified initial air consumption.

Primary consumers of compressed air were mentioned above. In the operation, the air is provided as a service air from service air bottle or from start air bottle through reduction valve. The pressure of the system controlled around 7-8 bars therefore, the consumption of the air is not highly dependent on start air pressure. However, in some cases, consumption and leakage volume increase in higher pressures. The increased demand will not exist in the case of the absence of the high pressures. If the system can operate its functions with lesser amount of air, the excessive usage of air and excessive leakage amount is called as artificial demand in the system. In a commercial ship, consumption amount of the air by ship can be around 40kg/h to 100kg/h according to system types and consumption volumes. Total operating consumption is expressed by following Equation 4.3;

$$\frac{m_{air\_out}}{dt} = \dot{m}_{air\_cons} + \dot{m}_{air\_leak} \tag{4.3}$$

Where m ̇_(air_cons) and m ̇_(air_leak) are the total consumption by systems and leakages from system respectively. To meet this demand and keeping system in a steady pressure, air compressor feeds up the air bottles at the range high pressures. After a while, compressor stops and runs again to compensate consumption. In such a way, fluctuation in the system can be minimized. Depending on system capacity, total consumption, number and size of the compressors, control type of compressors varies. Main control types are inlet modulation, load/unload, variable displacement, variable speed, start/stop control, listed least efficient to most efficient respectively.

To perform an efficient operation of compressed air system, control type is an important factor but, for small size of compressors start/stop control, in use on ships, is the most efficient method for controlling pressure in the air bottles. The aim of this part to switch compressor control ranges to more efficient range. To achieve this objective operating principles of compressors must be well-known. Applying the knowledge to a mathematical model with a data taken up from ships allows simulate different pressure or leakage conditions. In this regard, effects of different parameters, mentioned above, on power consumption of air compressor or cost of air per unit for ships and also an evaluation can be easily done in terms of emission generation for the production of compressed air. The model allows this kind of calculations and comparing them with each other.

In the modelling of these system compression process is assumed as polytrophic process. The increase of pressure in the compressor described by polytrophic equations, that temperature change in the air can be expressed as a formula below which is assumed as a polytrophic compression as equation 4.4;

$$\left(\frac{P_{c\_out}}{P_{c\_in}}\right)^{\left(\frac{n-1}{n}\right)} = \frac{T_{c\_out}}{T_{c\_in}} \tag{4.4}$$

Where n is the polytrophic superscript constant used in the compression processes as the value range 1 to 1.4. The value has been assumed as 1.2 or 1.3 (Festo, 2016; Taheri and Gadow, 2017; Yu et al., 2014) in the studies.

Thus, the total work of the compressor to air can be calculated with the converting the general formula in Equation 4.5 to Equation 4.6;

$$Q = m.C.\Delta T \tag{4.5}$$

Where Q is the total heat work, $\dot{W}_c$, expressed in equation 6, done by compressor to the air.

$$\dot{W}_c = \dot{m}_{air\_in}\,\Delta h_{out-in} = \dot{m}_{air\_in}\int_{T_{c\_in}}^{T_{c\_out}} C_p dT \tag{4.6}$$

The temperature difference can be calculated with the *equation 4,* and *equation 6* can be transformed to Equation 4.7 to obtain electrical consumption of compressor $\dot{W}_{elec}$, which must include compressor efficiency varies by the forced air flow through to air bottle.

$$\dot{W}_{elec} = \dot{m}_{air\_in}.C_p.T_{in}\left(\frac{\left(\frac{P_{c\_out}}{P_{c\_in}}\right)^{\frac{(n-1)}{n}} - 1}{\mu_{comp}}\right) \tag{4.7}$$

μ_{comp} is the efficiency of the compressor which is calculated by adopting the relation between compressor output capacity, discharge pressure and electrical power consumption of compressor while in operation besides, this nomenclature includes both electric motor efficiency and compressor efficiency. By relating them, the efficiency expressed as a function of discharge pressure or can be called as adverse pressure of air bottle. However, measuring of compressor output flow requires flow sensors, implementing the measurement system to any ship is impractical. To simplify the matter, air flow can be calculated by using relationship of the increasing air pressure trend in the tube with the time via using mass-pressure balance equilibrium. In this method both compressor working characteristic and air consumption by other systems can be calculated.

To represent μ_{comp}, energy balance equilibrium can be written as in Equation 4.8;

$$\dot{W}_{elec} - \dot{E}_{loss} = \dot{H}_{air_{out}} - \dot{H}_{air_{in}} \tag{4.8}$$

Where total electric consumption of compressor changes enthalpy of air passes through compressor. $\dot{E}_{loss}$ is the losses which represents, mechanical loses as friction by 10% (Festo, 2016), pressure drop, piping, storing, filter etc. Considering losses, efficiency can be defined by ratio of internal energy change in air and electrical power consumption by compressor. Enthalpy change also can be written depends on temperature difference of air.

$$\mu_{comp} = \frac{\Delta H_{air}}{\dot{W}_{elec}} \tag{4.9}$$

In compressor catalogues efficiency calibrated as minimum and maximum efficiency level of compressors. Maximum efficiency is accepted as a 100% efficiency for compressors. In this study equations 8 and 9 are used to calculate efficiency which includes all losses as motor losses and compression losses. Compressor model was adjusted via real compressor data and dynamic responses of the system have been changed in the model. If any enhancement of efficiency measures, that the model has capability to adopt to a new system model. The present study was performed to investigate potential energy savings in compressors which are operating in ships. The compressors fitted to ships to meet the required air demand by consumers.

Ships can have different air demands as a result of size of ships or different air consumer systems fed by. The operating data and compressor specifications are demonstrated in Table 4.1. The power consumed by compressors, measured current and voltage values, air bottle capacities (for each one), actual operating range of the compressors and loaded time (operating time of compressor in a specified time interval for instance; 12 min. in an hour corresponds 20% loaded time) are indicated for four different ships.

Table 4.1: Four Different compressed air system specifications.

Ship Type	Ship A Bulk Carrier	Ship B Container Ship	Ship C Container Ship	Ship D Container Ship
Power	55.7 kW	44,1 kW	57.5 kW	160 kW
Electric Voltage	440 V	440 V	440 V	440 V
Electric Current	85 A	65 A	85 A	267 A
Air Bottle Capacity	9.5 m^3	5.5 m^3	12 m^3	15 m^3
Operating Range	24 - 28 bar	25 - 29 bar	27 - 29 bar	28 - 30 bar
Loaded Time	10.5 %	12,50%	17,00%	18,00%

4.3 Results and Discussions

According to the data obtained from ship, air bottle capacities, electric current, operating range, operating load and operation time demonstrated in the Table 4.1. Additionally, by using the Equation 4.9 efficiency is calculated in terms of operating pressure denoted as a function in Figure 4.2.

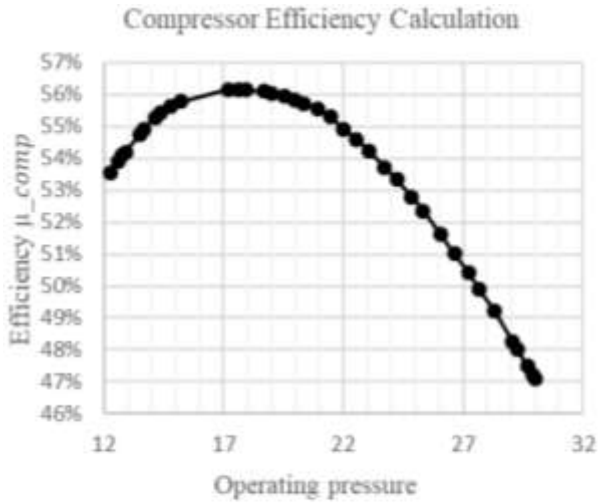

Figure 4.2: Efficiency of compressor as a function of pressure.

The formula which represents compressor efficiency was embedded in the model. The mass of air ($\dot{m}_{air}$) in to the air bottles varies with the pressure by the virtue of intrinsic phenomenon of polytrophic compression process expressed in equation 4. According to the equation, it can be seen that, initial temperature of the air is directly affects efficiency of compressor. Since the variation of air temperature is unavoidable during voyage, effect of inlet temperature of air was not added to model. However, as indicated by Dindorf, reducing inlet air temperature costs less 1% power for per three degrees drop (Dindorf, 2012). Compressor working load is part loaded during operation which is indicated in table 1 between 10-20% time that, additional effort could not be paid to adjust air temperature around compressor all the time in the engine room conditions. That is why temperature effect is not investigated in this study. The simulation gives results according to activities that can be done without the initial investment cost of the selected compressors while considering safety issues or together with least expense solutions.

In the modelled system the simulation was performed according to consumption of air. Air utilization in considered ship systems is 90-95 kg/h and supplied by the compressor which is introduced in the table 1 specified as Ship D. Since the system pressure range is very close to maximum operating pressure which means compressor is not performing at optimum ranges and together with considering electrical consumption and loaded time for the system, these criteria makes Ship D best selection to work on it. Although pressure reduction has unfavourable effect on exergy stored for one cubic meter (Budt et al., 2016), high volume of air bottles in Ship D compensates this in navigational operations. To reveal operational savings and additional variables, pressure range scenario was applied to compressor model in ranges which are 29-30, 28-30, 28-29, 28-24, 28-23, 27-24, 26-23, 25-20, 24-19, 22-18, 22-16, 18-14 bars. As can be seen in the table 1 at other ship types minimum operating pressure is 24 bar but this pressure is higher than minimum required operating value in 9 bars given in DNVGL and sufficient airline pressure 8 bars for system. Minimum operating pressure adjusted to 14 bars in the model. Simulation results gave information with regard to pressure range in specified time interval; on power consumption, air consumption, unit compressed air cost in kW, leakage and consumption rate reduction which is called as artificial demand, fuel and money savings, and CO_2 emission reduction.

4.3.1 Power consumption

As can be seen at Figure 4.3, in all cases power consumption of compressor is decreasing with the lower pressure ranges. These low consumption values depend on both increasing of compressor efficiency and decreasing of air usage in the systems. For the continuous operation (24 hours) in a day, day-wise power consumption is demonstrated. While operating pressure between 29-30 bars, the power consumption of compressor 518.9 kW/day, in the case of reduction in operating pressure, power requirement decreases. At minimum operating pressure range, power consumption can be reduced to 273.2 kW/day at the same conditions.

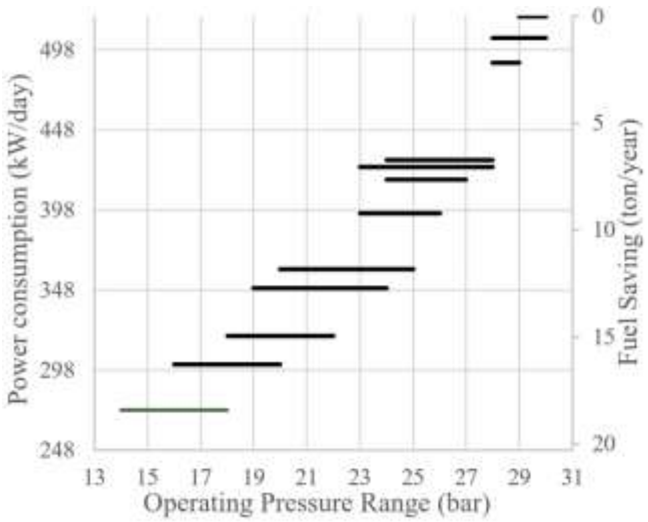

Figure 4.3: Power Consumption variation as a function of pressure in a day.

The increased efficiency of the compressor is effective in the required power. The fficiency increases 9.0% approximately near the operating pressures 17 bars.

4.3.2 Air consumption

Total air consumption which comprises both system leakages and air utilization, is demonstrated in Figure 4.4. The amount of air utilization is slightly affected by operating pressure range. The total consumption of air is decreased from 94 kg/h to 87 kg/h in specified case for an hour. The air consumption reduction is about 8.0% for maximum and minimum operating pressures. Despite the slight decrease of the air consumption, the reduction in power requirement is 47%. It can be said that a small part of power saving comes from saving in air utilization. The phenomenon is called as artificial demand that higher operating pressures cause higher air consumptions.

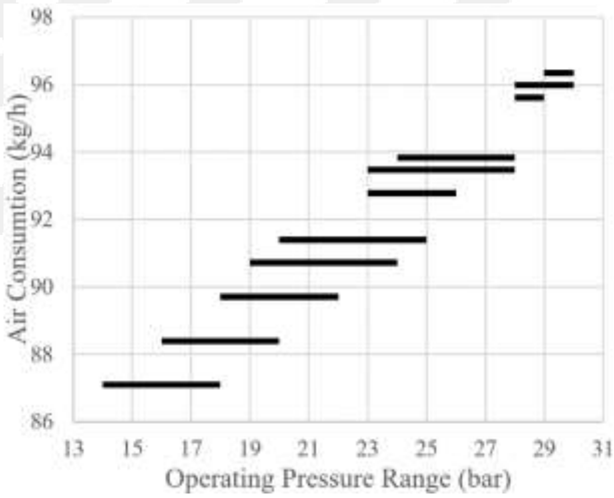

Figure 4.4: Air Consumption variation as a function of pressure.

Since air consumption is resultant of consumption of air leakage and air utilization, variation in both parts are investigated separately in another case.

4.3.3 Reduction in air utilization and leakage

The air leakage measurement was carried out during without any air consumption by systems. The pressure reduction in the air bottle is assumed as leakage rate. The amount of leakage rate is much smaller than the amount of air utilization when compared. Typically, 2.3kg/h leakage amount measured when total air consumption is 94kg/h approximately at specified operating pressure range. With the decreased operating pressure, the leakage rate decreases. The reduction in both air utilization and leakage rate demonstrated in terms of percentage of initial usages. It is assumed as 100% both usage of air and leakage in maximum operating pressures. With the drop in operating pressures, both of the incorporated consumptions could be performed with lower amount of air. On the basis of this reduction is reducing the artificial demand for the system.

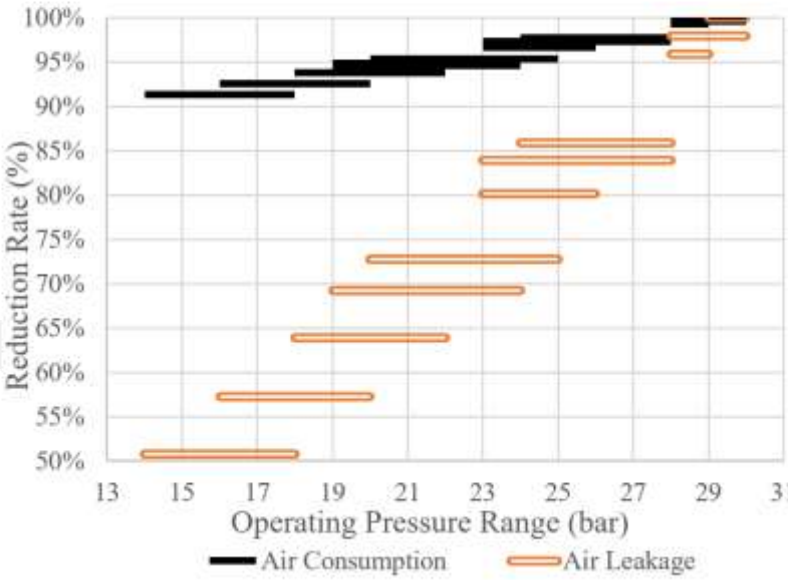

Figure 4.5: Reduction of artificial demands as a function of pressure.

It is indicated in Figure 4.5, reduction ratio in the leakage rate is much more than reduction in air utilization. While consumption rate of leakage is 50.8% of initial leakage amount, the air utilization is 91.3% of initial air consumption at minimum operating pressures. This representation shows air leakage depends on operating pressures more than air utilization for systems. The figure indicates that, ship system's air usage requirement is about 91.3% of total air usage at maximum operational pressures. The consumption amount can be accepted as a baseline. 8.7% of utilized air reduction is expressed as artificial demand and also can be eliminated by operating pressure reduction. Reducing pressures also gives a great advantage to reduce leakages. Halved leakage rate could be achieved with the reduced operating pressures, However the amount of the lowered leakage is relatively low when compared with the air utilization reduction. Additionally, variations of leakage conditions were investigated separately in next section.

4.3.4 Compressed air cost

It is important to specify evaluation factor as "compressed air cost" from the point of view in kW/kg to make a neutral assessment. By the normalization of all factors as leakage rates and consumed amount of air, required power to produce per unit (kg) of air in terms of kW becomes a reliable indicator for observers. In the evaluated system, demonstrated in figure 6, compressed air cost 0.22 kW/kg at maximum operating pressures 29-30 bars. The value of compressed air cost could be reduced to 0.13kW/kg that provided in the case of operating pressure drop at 14-18 bars. Unit cost is independent of the air amount consumed in the system, it is an indicator of cost of qualified air which is capable of performing same systems.

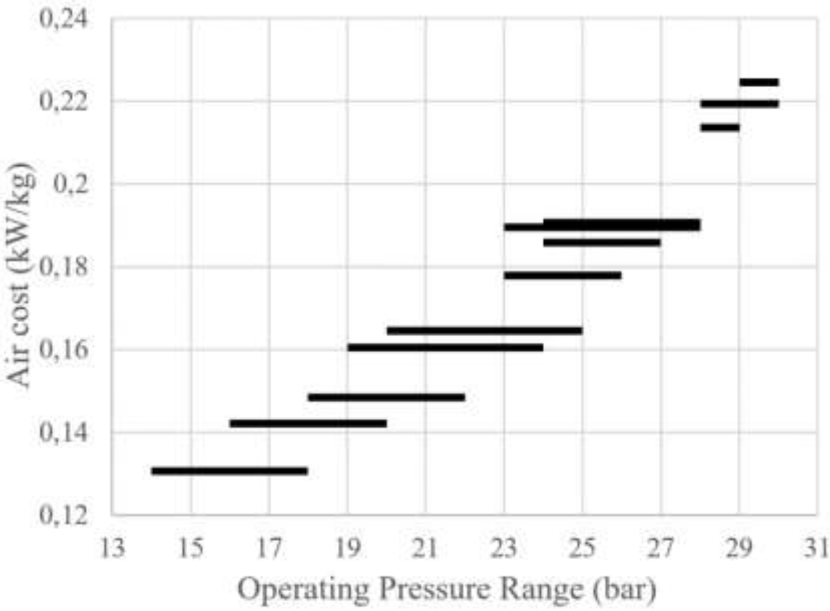

Figure 4.6: Reduction of Compressed air cost as a function of pressure.

The minimized air cost is 60% of the initial cost. However total required power is 47% of the initial power requirement for the same operating pressure ranges. The favourable difference is caused by lowered leakage and air consumption rates.

4.3.5 Savings in fuel and fund

Total possible saving in fuel is demonstrated in figure 3. The corresponding saving in dollars can be calculated from price of fuel. The savings calculated from simulation model and assumed that 300 voyage days' navigational operation for a ship in a year. Savings in fuel and fund assumed as 0% at maximum operating pressures. By the reduction of pressure steps, fuel and money savings could reach

18.4 tons of fuel and 7600 USD respectively for the fuel oil type with 380 cst and \$413.5/ton price (Ship and Bunker, 2018). The values can be seen as small amount of savings when compared with annual operating cost of a ship. The results should be considered under the perspective that the study carried out without the initial investment cost. Such a perspective will enhance applicable area of these savings.

4.3.6 CO_2 Emission reduction

Energy efficiency applications and saving fuel have positive effect on emission reduction. Since ships have only one energy source as fossil fuels, operational energy reduction directly saves both fuel and money together with taking care environment. Reduced emission value reaches 57.4 tons of CO_2 in a year in optimized operation of compressed air. Taking into account that global merchant fleet in the world exceeds 50k ships, adaptation of saving such methods in this perspective, without extra expenditure, an effortless attention can be paid for environmental aspect.

4.3.7 Leakage analyse

Leakage is crucial matter in compressed air systems in all facilities. This problem becomes more critical issue that ship has a moving and stretching structure. Because of the dynamical forces in sea conditions, compressed air system which includes numerous of pipes and junctions, prone to leak compressed air to outside. The acceptable value of leakage rate can be 2%-10% of the total air usage. In the ships leakage rates can exceed the acceptable values because of mentioned reasons. Air leakage does not leave any trace such as oil leak or water leak. It is very difficult to be noticed. Higher leakage rates cause the compressor to operate more and consume more power than desired. In the study the leakage rate is specified as 2.4% of utilized air which is assumed basic leakage from measured values and also used in previous section. The variation of leakage rates 5% – 10% – 25% – 35% – 45% were simulated in the model. The results as power consumption, leakage rate variation amount, air consumption, leakage flow and compressor load factor are investigated in terms of both variable operating pressure and increased leakage rates. Results are demonstrated in Figure 4.7- 4.8- 4.9.

The variation of leakage rates is given in Figure 7. The figure indicates that leakage amount decreases with the operating pressure drop in the all leakage rate cases. The maximum rate of the leakage is 45% could be lessened to 24.7% in low pressure operating conditions of compressed air system. The eliminated leakage amount is considerable for high initial leakage rates.

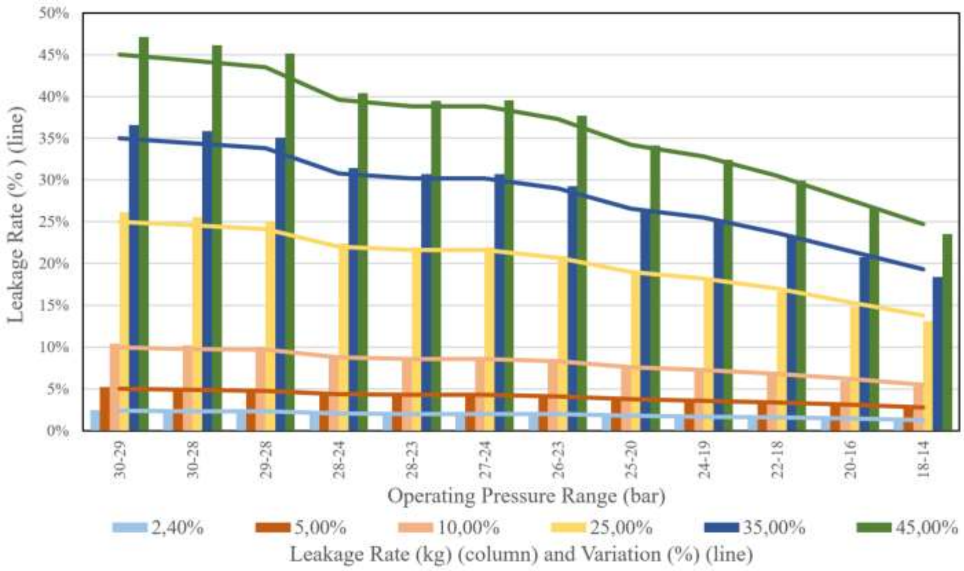

Figure 4.7: Leakage rate variation against operating pressure.

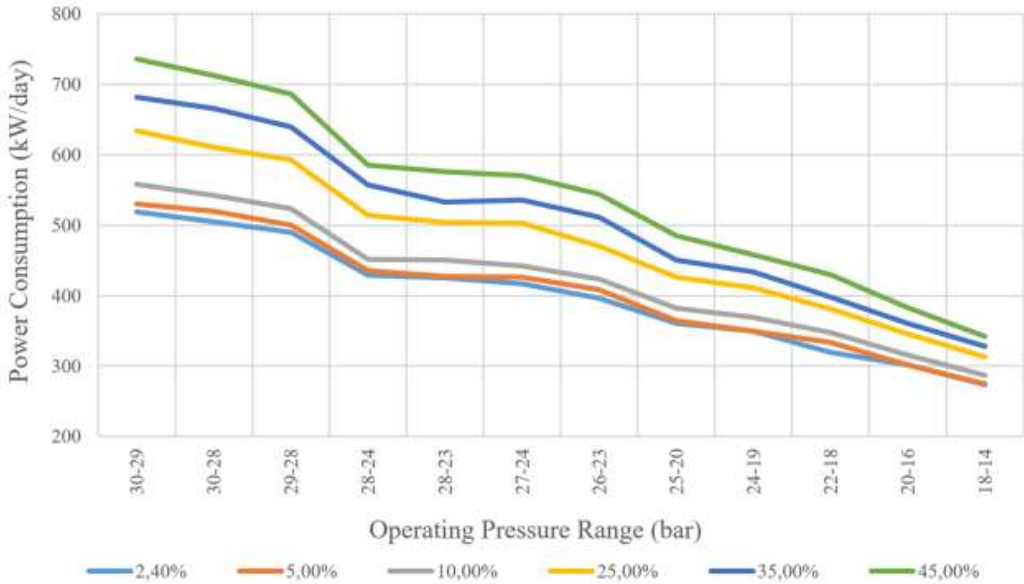

Figure 4.8: Daily power consumption with respect to pressure ranges and leakage rates.

Power consumption values increased at high leakage rates, given in Figure 4.8. At the maximum operating pressures daily power consumption of compressed air system rises from 518.9 to 735.8 kW with an increase of 41.7%. The percentage indicator shows effect of the operational pressure at different leakage rates. While in the case of operating pressure 28-24 bars the ratio is 36.2% and for the minimum operating pressures, the power need rises 273.2 to 341.8 kW with a ratio of 25% which is the comparison of minimum and maximum operating pressures. If it is considered that the 45% leakage rate, the power requirement drops 735.8 kW to 341.8 kW, which also means lesser than half power. The values reveal that the pressure effect gains in importance at high level of leakages. While additional power requirement is 216.8 kW in the maximum operating pressure, the power need 155kW, 110kW and 68kW for the 28-24, 22-18 and minimum operating pressure ranges, respectively.

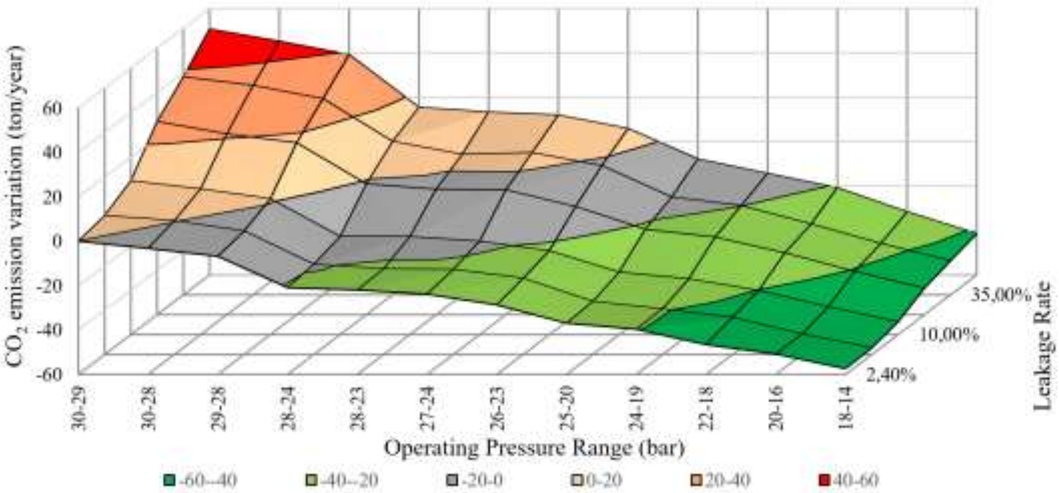

Figure 4.9: Amount of CO_2 emission change with the variation of Leakage rate and operating pressure.

The air consumption is not affected by leakage volumes. Hourly leakage amount which is given in Figure 4.7, increased 2.2 kg to 42.4 kg for an hour at maximum operating pressures. By the decreasing of operating pressures leakage amount drops. The leakage reaches 21.2 kg at minimum operating pressure with a maximum leakage rate. Leakage can be cut in half with operating pressure drop which also leads 216kW power reduction.

CO_2 emission generation demonstrated in Figure 4.9 which changes by variation of pressure and leakage parameters. In the figure emission generation is compared with the initial condition as 2.4% leakage 29-3,0 bars operating pressure.

By the increase of leakage 2.4% to 45% CO_2 level climbs 50.7 tons more, demonstrated “red (+)”. Reduction of operating pressure has favourable effect on emission. Operating pressure around 25 bars neutralize 45% excessive leakage which corresponds 29-30 operating pressure at minimum leakage rate. At minimum operating pressure condition, emission reaches maximum reduction level 57.4 ton, demonstrated as "green (-)" in the *figure 9* which was mentioned previous section and this favourable effect was smoothed over by increased leak rate about 16 tons. If considered case 45% leakage rate, operating pressure has a great effect on emission which differs 92 tons of CO_2. At the other hand, at minimum operating pressure range leakage effect is 16 tons while at maximum range 50.7 tons that another favourable effect of operating pressure at higher leakage conditions.

Compressor load is an important indicator of exceptional cases. In the case of any leakage, excessive consumption or inefficiency of compressed system, it can be foreseen any defectiveness existence. Load factor which means ratio of operating time to elapsed time includes both operating and idle time.

$$Load\ Factor = \frac{Time_{loaded}}{Time_{loaded} + Time_{idle}} \tag{4.10}$$

Time interval of compressor switch on period is another indicator of load factor. For instance, while in a proper running system, load factor can be 10 minutes, in a problematic system it can drops even further which means increase of load factor. Load factor variation is demonstrated in Figure 10. By comparing of time intervals of activation of compressor operational efficiency can be realized.

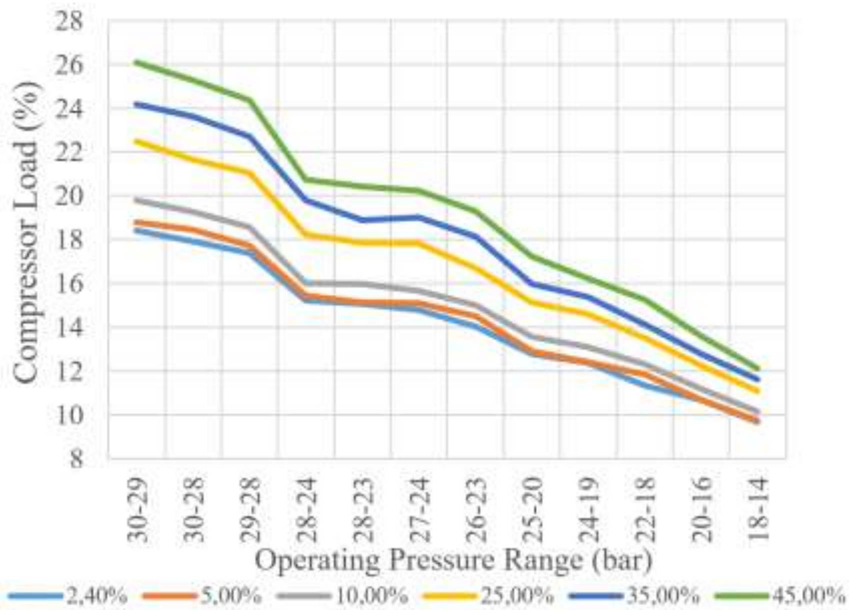

Figure 4.10: Compressor load variation with the leakage rates.

Increase of the load factor from 18.4 to 26 responds as a power consumption increment from 518.9kW to 735.8kW per day, in maximum operating pressures as mentioned in previous section. At the other operating pressure ranges, load factor decreasing with pressure drop and increasing with leakage increment. Reduction in operational pressure has more advantage at higher leakage rates both on diminishing the amount of and keeping compressor at similar loads. It can be clearly seen that changing in load factor is a response of the any imperfection in the compressed air system. An estimation can be accomplished by considering and comparing load factor with base conditions.

4.4 Conclusion

To accomplish energy efficiency plan in an extensive manner, all energy consumers must be analysed in a deeply conception. Energy efficient operation of compressed air is investigated in this study. According to analysed system, a low-cost method, reduction in operational pressures, carried out in system model to reveal possible operational savings. System air consumption and leakage rate data were embedded in to model. Except manoeuvre and departure from port operations, excessive pressures are not required for compressed air system. By considering of operating priorities and safety issues, different pressure ranges performed on the system. Results of the simulation indicate power need of compressor, leakages and compressor load were diminished while air consumption slight decrement which can be ignored at lower operating pressures. Leakage is un avoidable matter in compressed air systems. Rate of the leakage in acceptable range up to 10% and excessive leakage up to 45% were performed in the model. Leak amount, consumption change, and compressor operational load were observed. Power requirement and compressor load are increasing, while air utilization has stable value at higher leakage rates. It seems compressor load is an essential indicator of imperfection of the system. To avoid excessive leakages, leaks must be detected. To identify location of leaks, it is easy to use mobile ultrasonic acoustic devices, that can detect hissing noises and have capability of diagnosing high frequency noise sources.

Operating of compressor in high time-loads has unfavourable effect on compressor components as valves etc. and lessen efficiency of the operating duration of these components. As an additional outcome, economic life of the equipment and components must be considered in such conditions. This operation not only comes out with an inefficient operation and wasting energy together with emission generation but also reduces of economic life time of the components. Efficient operation of compressed air systems has non-energy benefits on life time of components need more research to reveal (Nehler, 2018).

The results, taken from the model, may seem inconsiderable when compared with the total operating costs of ship however, the higher saving values can be attained by the combination of solutions without initial investment cost and such applications with low expenses. To achieve desired energy saving rates in total, energy consumption routines of people should be modernized (Bilgen and Sarıkaya, 2018). If global fleet capacity is considered, implementation of these type of actions has world-wide consequences on substantial amount of fuel saving and emission reduction. Implementation of such applications in maritime industry will enhance efficiency policy in both cases of design and operation (SEEMP&EEDI&EEOI). By figuring out of these applications by MEPC as an advisory opinion or as an enforcement, considerable amount of reduction can be achieved at first step just with operational tricks without initial investment or with low expenses. Since the modelling, which used in this study, enables growth that capabilities of model can be enhanced with embedding additional data in it. The results of such simulations allow mariners to be aware of potential savings at operations thus, the results are useful for training and workshops and encourage engineers pay more attention to, measure efficiency of systems, minimization of loses and monitoring.

5. CONCLUSION

5.1 Conclusion

In the last decade, carbon emissions could be significantly reduced comparing to 2008 levels. The regulations, legislated by IMO, have encouraged shipping companies to conduct more efficient operations over the past years. There are many methods, being implemented, to reduce carbon emissions as in ship management level, ship design and construction level, using alternative or renewable fuels, and improvement in ship power plant. Except using alternative and renewable fuels, the other measures focus on reducing required power/fuel for ship propulsion. One of the common practices to decrease fuel consumption and carbon emissions is reducing operational speed of the ships.

Cruising speed of vessels had slowed down around 6.0% for bulk carriers, 15.0% for container ships and 11.0% for the tankers from the 2007 to 2012 given in IMO report (IMO, 2014). The speed reduction allows required main engine power reduction for propulsion as 19%, 37% and 30% for bulk carriers, container ships and tankers respectively. These ships covers the 65% of the shipping market. Significant amount of fuel saving could be achieved by this method and the ships have continued to slow steaming operation without no more reduction in speed. A significant part of desired efficiency levels had been completed between the years 2008 and 2016 (IMO, 2020b) then, after 2016, there is a slight variation in operating speeds of ships, which shows it became a permanent solution for energy efficiency applications. Still, in short-term strategy of IMO, speed reduction is declared as a solution for carbon emission reduction. Additionally, the conversion of the energy source from diesel oil to the alternative fuels seems will take a process because of diesel dependency of the market. Therefore, there is a need to decrease fuel consumption of ship while boosting the efficiency in slow speed conditions.

On the other hand, IMO updated the EEOI term with a new term as AER and new terms are in use EEXI (Energy Efficinecy Existing Ship Index) and CII (Carbon

Intensity Indicator), will be valid in year 2023. The ships, which cannot meet the Carbon Intensity Indicator (CII) requirement, should revise their SEEMP plans by defining their efficiency measures, evaluation methods and improvement strategies. With the help of Data Collecting System, IMO can monitor the efficiency trend and makes estimations more precisely. Thereby, IMO will be able to put more aggressive pressure for ships in the following years.

In this regard, improving energy efficiency of the ships are going to be more challenging issue in the following years while the marginal utility of the slow steaming, which was a great contributor to energy efficiency in recent years, has decreased. The study has focused on the improving efficiency of the ship by analysing the operational parameters in ship power plant in order to increase the main engine efficiency and to reduce the required power of auxiliaries. In order to calculate the effect of the proposed solutions, model-based approach was conducted, by developing models of ship's main engine and some equipment, to identify excessive energy usage. The ship's operating profile (speed or main engine load) is the main parameter for the calculations since the main engine operates at inefficiently. Therefore, the practices should be proposed to increase efficiency at reduced loads. Additionally, auxiliary systems' operation should have been re-evaluated with low energy demand perspective because of scaled down waste heat source.

The modelling solutions gaining importance in recent years because of both being simple, and time-cost efficient with a ease of demonstration of ship energy tracking with the help of capability to evaluate realistic performances of ship power plant under different operating conditions and with variable control parameters. In this context, there is a broad literature application, which improve energy efficiency in ship power plant that the measures applied for reduced load of main engines.

The development of model enables to evaluate the effect of operational variables on system energy consumption afterwards fuel consumption equivalence. The model provides beneficial understanding about operational efficiency, fuel consumption and energy balance of the system with their interaction. By this way, the optimisation methods could be proposed with accurate estimations in saving and efficiency. The integration of new developed energy saving technologies or solutions to the model together with implementing realistic operational profiles enables to explicit estimation of the results of efficiency-oriented solutions, which helps to improve efficiency level of both existing and new ships.

The model of main engine had been developed to simulate the behaviour of two-stroke marine diesel engine at various loads, therefore the energy efficiency level deterioration at reduced loads could be investigated. The dissipation of heat energy proportion to the cooling medium increases at reduced load, which results in higher specific fuel oil consumption for marine diesel engines. The one of the main parameter is the relatively high ratio of the cooling loss against to main engine generated power. When a cooling phenomenon of the two-stroke diesel engine is investigated, it is seen that the difference between inlet and outlet temperature of cylinder jacket water decreases at reduced loads. By considering, the outlet temperature of JW is fixed (or controlled) with the same amount of coolant flow, the inlet temperature is increased, hence it can be easily estimated that the heat transfer between jacket water and cylinder liners decreases. Although, the proportion of cooling loss is high at reduced loads, the high loss does not mean that the combustion gases transfers higher amount of heat to the liners. Relatively low temperatures at cylinder liners leads higher temperature difference between in-cylinder gases and cylinder liner, which causes over cooling. The temperatures of the liners were determined by the safe operation of the main engine for high loads, therefore it could also be operated with the same temperatures at reduced loads, safely. The elevated temperatures resulted in less temperature difference between gas-wall, with reduced heat flux, less cooling loss, increment in power, increment in exhaust temperatures, turbocharging and waste heat potential. For all these components, the cooling loss effect is investigated into the mechanical domain with shaft power increment and turbocharger power, heat domain with exhaust heat. Additionally the increment in waste heat recovery potential, the fuel requirement for the auxiliary generator and boiler is calculated to observe the effect on fuel as well as, less consumption of main engine to generate the same amount of power.

Ship power plant is comprehensive and complex system that includes many equipment with interaction each other. The cooling of main engine liners are carried out by high temperature fresh water system together with the scavenge air cooler and lubricating oil coolers’ low temperature fresh water flows. The flow and pressure to the lines, then to the heat exchangers supplied by fresh water pumps. The other side of heat exchanger is seawater that, together with freshwater pump seawater pump operates 24 hours in cruising. However, in reduced load the required cooling amount decreases, In the study of central cooling optimization, the need for cooling at reduced loads used as an input for the model. With the help of heat exchanger model, the flow rates and input temperature of fresh and seawater are used to estimate output temperatures, in order to conduct safe and efficient operation of cooling. As an output of the calculation, pump requirements were calculated, with respect to the operational speeds and associated efficiency. Because of system limitations, instead of considering only one side of the flows, for instance seawater or fresh waterside separately, constraints and efficiency decrement do not allow enough for main engine cooling decrement. The parallel optimizing both side of the system allows significant reduction of power demands of the pumps.

The other modelling study is the compressed air system that is an electrical consumer equipment was selected because of intrinsic efficiency of the system, The compressed air study highlighted the inefficiencies that generally had not been paid attention to, in the share of electrical power consumption in the ship power plant. However, in addition the existing functions of compressed air system, the selective catalytic reduction system and soot blowing systems have started to use compressed air continuously while in operation. The pressure depended system dynamics were measured. The consumption of air, compressor-running schedule, pressure trend of air bottles both as a function of time and as function of operating pressure ranges. According to air consumption of the systems and the electrical consumption of the compressor, the cost of the air, the efficiency of the compressor and the total electrical demand of the system were analysed with the help of the modelling method. Application of different pressure ranges, realize the saving capability of the system, due to the reduced load electrical demand of the whole system in 24h period. The study could reduce the intrinsic inefficiency in the production side of the system, additionally pin point the slight artificial excessive usage of compressed air systems at higher-pressure ranges, which is in the demand side.

To sum up, in order to increase energy the efficiency in ship operations, the more accurate estimations are needed. Modelling methods provide a great advantage to calculate how much advantage can be obtained before applying the efficiency the measure. Additionally, The modelling method disables the disadvantage of on-board measurement systems as lack of accuracy and variation of the data during the measurement process. For instance, unrealistic data due to lack of accuracy of measurement tools could be observed during the data acquisition with the high variation of data gathered via sensors. Modelling tool enables to estimate more accurate effect of any potential increment in the operation and easy to observe the all the benefits about enhancement in the system.

5.2 Practical Applications and Further Studies

Model-based studies can be extended to a wide variety of power generation facilities and equipment, no matter what type of fuel or energy is used in ship power systems. Considering the irreversibility of the systems discussed in the studies, creating an end-to-end (from energy source to consumer) flow estimation system, gives a valuable insight in tracking of energy production and consumption in ships. In this thesis, the main steps were discussed depend on the physical domains. Interaction of the systems with each other, ship power plant performance improvement is achievable by investigation of a ship with more precise estimation methods.

Additionally, the model-based solutions could support the ship managers to determine the investment of application for efficiency solutions on their ships in order of low cost/high payback advantage it cannot be done regardless of expenses.

While, energy efficiency applications vary in a wide range of operational measures in literature, this study focused on the heart of the ship, main engine and auxiliary equipment having a big share on electricity consumption, and the most inefficient equipment for the existing ship.

A comprehensive calculation method must be carried out with the help of simulation/models, in order to fully evaluate the systems interaction. The interaction between sub-systems through different domains should be well represented by mathematical infrastructure of the model. The variation of operational conditions affect the output of the model hence the following domains specifications. The model-based studies will be improved from component level to system level to achieve a holistic approach for whole ship energy system. Synchronisation of system model of the ship with a data driven approach, the attained efficiency level can be evaluated with a communication system between them. Monitoring and managing the system is achieved by paralleling real and desired data with a combination of central control of the systems.

Extending the system borders or adding new models cooperated with the existing main engine and auxiliary systems model; more extended prediction capability can be achieved with their interaction and responses. More detailed sub-models can be switched with the limited sub-models such as representing spray quality, injection pressure. Injection model can change the shape of combustion parameters which also affects forming of combustion in the cylinder. For instance compression quality can be represented by adding compression sub-model Piston rings, or exhaust valve clearances. The detail level of the system is determined by the level of sufficiency to describe exchange of energy for a given scenario, and the calculation capability is determined by the modeller to see the effects of variables with a reliable results. In order to minimize first investment cost, additional equipment solutions could be modelled in accordance with the other specifications of the ship by considering economically attractive solutions to increase energy efficiency. The model should provide extendable programming architecture.

Models may differ for each ship, as well as being exclusive for each operation. The most appropriate energy efficiency management method should be applied according to respective conditions. Weather, currents, engine condition, load condition, operation profile, freight rates are considerable inputs that cause significant changes for energy efficiency management; they may require some adaptations to the plan.

The roadmap for improving ship energy efficiency should be given in SEEMP documents. The energy management policies of the ships should be prepared in a unique way for each individual ship, taking into account that the operation modes and operating conditions of even the same-sized (sister) ships.

Ship operating companies aim to take their SEEMP plans one-step further with the planning phase, implementation, observation and feedback within the scope of their environmental management policies. Following these steps one after to another and repeating them in a cycle aims to achieve a sustainable development. It is of utmost importance not to risk ship operations, and the feasibility of the operations to be calculated must be well prepared, as ship operators will also consider company profitability. In this regard, evaluating the operation scenarios of ships with an appropriate model is a valuable tool to make accurate predictions. With the help of the simulation tool, pre-self-evaluation method is possible to fix any kind of unexpected results before putting implementation. This allows both not to waste of time and money. This method helps to understand more thing about the operation of the ship or equipment focussed with the understanding cause and effect performance. Preparing SEEMP plans for ships and creating a model of ships helps to address much more predictable results against emission emissions, fuel consumption, and various scenarios.

In terms of energy efficiency management, the importance of SEEMP must be well-understood by ship operators. Increasing the awareness of masters and engineers about energy efficiency measures is the paramount issue for shipping sector. Training of operators is an important practice that should be included in the energy efficiency policy of companies. The tool must not make any over workload to marine engineers.

Apart form the CO_2 emissions, there are significant emission reduction techniques which are called as Exhaust Gas Cleaning (EGC) systems in use in the ships to meet the NO_x and SO_x requirements of IMO. EGC systems can be installed as alternative emission abatement technologies, in spite of increase the fuel consumption while reducing emissions. Therefore, improvement of efficiency level of main engine and increasing exhaust gas temperature help to utilize exhaust gas cleaning systems. However, comparing with the world fleet, very limited number of ships (less than 750) have applied exhaust gas cleaning systems to mitigate emissions up to 2018. Although lack of collected data exist about EGC systems, it is expected that up to 2% increment in fuel oil consumption. On the other hand, increasing the scavenge air pressure can lead more air, be taken, in cylinder, with higher compression ratio and with the same amount of fuel peak combustion temperatures can be decreased, which can also leads reduction in NOx emissions. Additionally, increasing mean indicated pressure for per bar can cause reduction in specific fuel oil consumption about 0,20-0,25 g/kWh for the main engine. The technologies are using venturi tube, scrubber or Exhaust gas recirculation systems. The systems have negative effect on fuel consumption. It is possible to increase the fuel consumption around 2-3% for a ship (Ushakov et al., 2020).

For the future studies, it must be emphasized that holistic approaches will constitute the main frame of the energy efficiency studies. Investigation of waste heat energy potential can be carried out with a model-based approach for different environmental and main engine conditions. The maximum potential of the system can be calculated with the method according to realistic operating profile. New design for the waste heat recovery system can be adopted by optimizing the demand together with maximizing the heat source, by using additional feasible heat sources. The operation time of auxiliary generator and oil fired boiler can be diminished with the optimisation, which leads reduction in fuel and emissions. As an additional heat sources potential can be calculated with modelling method of elevated temperature of jacket water. With the elevated of temperature cooling circuit, Organic Rankine Cycles can be utilized more efficiently in order to harvest the waste heat in HT fluid. Additionally, at the ships operated with LNG, the tank keeps the LNG at cryogenic temperatures, therefore the heating of the LNG is required before the main engine, this heating process can be carried out by this high coolant temperatures.

In the light of energy usage of the systems, their operational priority can be ordered for energy demand shifting method. In the case of any excessive energy generation, the storage of energy techniques can be used for further use, or in the lack of energy for a short period, some equipment energy consumption can be diminished with the help of model based decision support systems.

Investigation of the new technologies and application of them the systems will result as a prominent solution only when all the related systems and sub-systems are considered with an enhanced approach. The solutions will provide sustainable growth in efficiency only by expanding the limits taken in to account. The concept of the thesis is combining the technical and operational approach in order to reduce carbon emissions. The investigations were carried out for the specific operational profile of a container ship with its specifications of main engine and components. However, different ship types with their individual specifications based on specific operational profiles can be studied with the same methodology used in these papers. In the studies to be applied to other ships, the modelling method should be focused on the ships that are the majority in the world fleet and have high emission shares.

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