Contribution to advanced waste heat recovery techniques on ship's propulsion plants

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EXPONEN:

Que han dirigido la Tesis Doctoral de D. Steven Iglesias García titulada:

Contribution to advanced waste heat recovery techniques on ship's propulsion plants

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Consideran que la investigación llevada a cabo por parte del doctorando aporta resultados útiles e inéditos. Asimismo, consideran que la Tesis está en condiciones de ser presentada y defendida ante el Tribunal designado a tal efecto. Por todo ello,

SOLICITAN:

Que la mencionada Tesis Doctoral sea aceptada para su depósito y proceder a su defensa y calificación.

A Coruña, Junio de 2020

Dr. Ramón Ferreiro García Dr. José Carbia Carril

Acknowledgment

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Finally, I would like to thank all of my family and my partner who have had faith in me and supported me during this testing time. My two lifelines (parents) have always supported me financially so that I only pay attention to the studies and achieving my objective without any obstacle on the way.

Steven Iglesias Garcia

Abstract

This thesis explores the potential of low and medium grade heat in different thermodynamic cycles used to convert wasted heat into mechanical work. The aim of this thesis is to study the state of the art of the thermodynamic cycles used to recover low-grade heat and suggest improvements.

The relevance of researching low grade heat or waste heat applications is that a vast amount of heat energy is available at negligible cost within the range of medium and low temperatures, with the drawback that existing thermal cycles cannot make efficient use of such available low temperature heat due to their low efficiency.

The present thesis offers a different approach, analyses low-grade heat recovery from a thermodynamic point of view, and compares their thermal efficiency. ORC and Rankine cycles reviewed show similar low efficiencies. In contrast, closed-based, non-condensing cycles, that exchange heat at constant volume and convert thermal energy into mechanical work adiabatically, have a configuration, which allows efficient exploitation of low-grade heat. Heat recovery strategies are also analysed and several non-conventional systems are recommended to exploit low grade heat in the proposed machine.

Two relevant factors exert a strong influence over the criterion to decide the structure of marine propulsion plants:

- The reduction of the fuel consumption which undergoes increasing the thermal efficiency of the associated thermal engines.
- The emission of carbon dioxide due to fossil fuels consumption

On the one hand the emissions of carbon dioxide from the combustion of fossil fuels used in marine propulsion plants and its associated power plants are a concern and it is due to the environmental impacts originated by greenhouse gas emissions, which contributes to modifying the composition of the atmosphere in an exaggerated negative sense. Such harmful methods can be replaced by alternative design strategies or at least could be mitigated by means of the reduction of carbon dioxide emissions, which could be achieved by changing strategies of technology design and operational methodologies. The available heat recovery

technology that can be used on Waste Heat Recovery Systems (WHRS) of medium or low-grade is one such a means that is technologically capable of decreasing carbon dioxide emissions as well as to reduce the fuel composition significantly. This is due to the fact that it is possible a significant improvement on the marine propulsion systems efficiency. Since it contributes to the reduction of fossil fuels consumption of the marine transport sector in general. The mentioned improvement can be carried out by harnessing the waste heat energy exhausted and / or rejected by the propulsion system and its associated auxiliary engines. The conventional technology so far opted for the harnessing of the exhaust flue gases, since lower temperature heat could not be used.

Conventionally the state-of-the-art technology in this field uses advanced Organic Rankine Cycle (ORC) under several combinational structures as one among some different WHRS technologies that are being used for the harnessing of the medium and / or low-grade heat wasted by the propulsion system that cannot be utilized by the propulsion engines by themselves.

On the other hand, the reduction of the fuel consumption which means increasing the thermal efficiency undergoing the reduction of carbon dioxide emissions, must be improved as much as possible based in that this criterion is sufficiently relevant to undertake the development of technologies aimed at improving the thermal efficiency of the thermal engines used so far.

Nevertheless, in this thesis a new paradigm of WHRS is investigated aiming to increasing thermal efficiency which undergoes reducing fossil fuel consumption and carbon dioxide emissions.

These improvements are associated with the following contributions proposed and carried out in this thesis:

- Low temperature heat recovery by means of designing a recovery structure able to recover residual heat of medium and low temperatures rejected by now, coming from exhaust gases, scavenge or air cooler, jacket cooler, and lube oil cooler.
- More efficient thermal engines by means of designing, analysing and conducting a proof of concept of thermal cycles based on closed processes operating by adding and extracting heat undergoing an innovative heat-work interaction mode.

Furthermore, the proposed thermal engines are characterised by a thermal cycle physically possible to be made composed of two closed isochoric processes consisting of adding and extracting heat to / from the Working Fluid (WF) and two closed adiabatic processes which undergoes useful mechanical work, consisting respectively of closed adiabatic expansion and closed adiabatic contraction-based compression of the WF.

The main results are:

- A power architecture capable of converting low-grade heat into mechanical work with a heat utilisation factor higher than any available or theoretical technology.
- A thermal cycle surpassing any other cycle for low-grade heat in terms of efficiency and heat utilisation factor.

Resumen

Esta tesis explora el potencial del calor de grado bajo y medio en diferentes ciclos termodinámicos utilizados para convertir el calor residual en trabajo mecánico. El objetivo de esta tesis es estudiar el estado del arte de los ciclos termodinámicos utilizados para recuperar calor de bajo grado y proponer mejoras.

La relevancia de investigar aplicaciones de calor de baja calidad o calor residual es que hay disponible una gran cantidad de energía térmica a un costo insignificante dentro del rango de temperaturas medias y bajas, con el inconveniente de que los ciclos térmicos existentes no pueden hacer un uso eficiente de dicho calor disponible a baja temperatura, debido a su baja eficiencia.

La presente tesis ofrece un enfoque diferente y analiza la recuperación de calor de bajo grado desde un punto de vista termodinámico, comparando su eficiencia de conversión. Los Ciclos Rankine Orgánico (ORCs) revisados muestran eficiencias similares. Por el contrario, los ciclos presentados en esta tesis, cerrados, sin condensación, que intercambian calor a volumen constante y convierten la energía térmica en trabajo mecánico adiabáticamente, tienen una configuración que permite la explotación eficiente del calor de bajo grado. También se analizan las estrategias de recuperación de calor y se proponen varios sistemas no convencionales para explotar el calor de bajo grado en la máquina propuesta.

Dos factores relevantes ejercen una fuerte influencia en el criterio para decidir la estructura de las plantas de propulsión marina:

- La reducción del consumo de combustible que experimenta al aumentar la eficiencia térmica de los motores térmicos asociados.
- La emisión de dióxido de carbono debido al consumo de combustibles fósiles

Por un lado, las emisiones de dióxido de carbono de la combustión de combustibles fósiles utilizados en plantas de propulsión marina y sus plantas de potencia asociadas son una preocupación debido a los impactos ambientales originados por tales emisiones de gases de efecto invernadero, lo que contribuye a modificar la composición de la atmósfera en un sentido muy negativo. Por lo tanto, estos sistemas nocivos pueden reemplazarse por

estrategias de diseño alternativas, o al menos podrían mitigarse mediante la reducción de las emisiones de dióxido de carbono, lo que podría lograrse cambiando las estrategias de diseño tecnológico y metodologías operativas. La tecnología de recuperación de calor que se puede utilizar en sistemas de recuperación de calor residual (WHRS) de grado medio o bajo es uno de los medios capaces de disminuir las emisiones de dióxido de carbono y de reducir significativamente el consumo de combustible. Esto obedece al hecho de que es posible una mejora significativa en la eficiencia de los sistemas de propulsión marina, ya que contribuye a la reducción del consumo de combustibles fósiles del sector del transporte marítimo en general. La mejora mencionada se puede lograr aprovechando la energía calorífica residual agotada y / o rechazada por el sistema de propulsión y sus motores auxiliares asociados. La tecnología convencional hasta ahora optó por el aprovechamiento parcial de los gases de escape, ya que no se pudo utilizar calor a temperaturas más bajas.

Convencionalmente, la tecnología de vanguardia en este campo utiliza ORCs avanzados bajo varias estructuras de ciclo combinado como una entre algunos de los sistemas de recuperación de calor residual (WHRS) diferentes que se están utilizando para el aprovechamiento del calor de grado medio y / o bajo que se desperdicia por el sistema de propulsión que no puede ser utilizado por los motores de propulsión.

Por otro lado, la reducción del consumo de combustible, lo que significa aumentar la eficiencia térmica, provoca la reducción de las emisiones de dióxido de carbono. Esto debe mejorarse tanto como sea posible, ya que este criterio es lo suficientemente relevante como para emprender el desarrollo de tecnologías destinadas a mejorar la eficiencia térmica de los motores térmicos utilizados hasta ahora.

Sin embargo, en esta tesis se investiga un nuevo paradigma de WHRS con el objetivo de aumentar la eficiencia térmica lo cual hace que se reduzca el consumo de combustibles fósiles y las emisiones de dióxido de carbono.

Estas mejoras están asociadas con las siguientes contribuciones propuestas y realizadas en esta tesis:

- Recuperación de calor de más baja temperatura mediante el diseño de una estructura de recuperación capaz de recuperar el calor residual de las temperaturas medias y bajas rechazadas hasta ahora, provenientes de gases de escape, enfriadores de aire de admisión, enfriadores de agua de camisas y enfriadores de aceite lubricante.

- Motores térmicos más eficientes mediante el diseño de análisis y realización de una prueba de concepto de ciclos térmicos basados en procesos cerrados que funcionan con interacciones que expanden y contraen el fluido de trabajo generando trabajo útil.

Además, los motores térmicos propuestos se caracterizan por un ciclo térmico físicamente realizable compuesto por dos procesos isocóricos cerrados que consisten en agregar y extraer calor hacia / desde el fluido de trabajo y dos procesos adiabáticos cerrados que convierten la energía térmica en trabajo mecánico útil, que consiste respectivamente en expansión adiabática cerrada y contracción adiabática cerrada.

Los principales resultados son:

- Un sistema energético capaz de convertir calor de bajo grado en trabajo mecánico con un factor de utilización más alto que cualquier tecnología disponible tanto comercialmente como en el campo teórico o académico.
- Un ciclo térmico que supera cualquier otro ciclo usado para calor de bajo grado en términos de eficiencia y factor de utilización.

Resumo

Esta tese explora o potencial de calor de baixo e medio grao en diferentes ciclos termodinámicos empregados para converter a calor residual en traballo mecánico. O obxectivo desta tese é estudar o estado da técnica dos ciclos termodinámicos empregados para recuperar calor de baixo grao e propoñer melloras.

A relevancia de investigar aplicacións de calor de baixa calidade ou de calor residual é que unha enorme cantidade de enerxía térmica está dispoñible a un custo insignificante no intervalo de temperaturas medias e baixas, co inconveniente de que os ciclos térmicos existentes non poden facer un uso eficiente de tal calor dispoñible a baixa temperatura, debido á súa baixa eficiencia.

A presente tese ofrece un enfoque diferente, analiza a recuperación de calor de baixo grao desde o punto de vista termodinámico e compara a súa eficiencia térmica. Os Ciclos Rankine Organico (ORCs) revisados mostran eficiencias baixas similares. En contraste, os ciclos sen condensación baseados en pechados, que intercambian calor a volume constante e converten a enerxía térmica en traballo mecánico adiabaticamente, teñen unha configuración, que permite unha explotación eficiente de calor de baixo grao. Tamén se analizan as estratexias de recuperación de calor e proponse varios sistemas non convencionais para explotar calor de baixo grao na máquina proposta.

Dous factores relevantes exercen unha forte influencia no criterio para decidir a estrutura das plantas de propulsión mariña:

- A redución do consumo de combustible que sofre ao aumentar a eficiencia térmica dos motores térmicos asociados.
- A emisión de dióxido de carbono debido ao consumo de combustibles fósiles.

Por unha banda, as emisións de dióxido de carbono procedentes da combustión de combustibles fósiles empregados nas plantas de propulsión mariña e as súas plantas de potencia asociadas son unha preocupación e débese aos impactos ambientais orixinados por estas emisións de gases de efecto invernadoiro, o que contribúe a modificar a composición da atmosfera nun sentido moi negativo. Así, estes sistemas nocivos pódense substituír por

estratexias de deseño alternativo ou polo menos poderíanse paliar mediante a redución das emisións de dióxido de carbono, que se poderían conseguir cambiando as estratexias de deseño de tecnoloxía e metodoloxías operativas. A tecnoloxía de recuperación de calor dispoñible que se pode empregar para sistemas de recuperación de calor residual (WHRS) de grao medio ou baixo é un tal medio que é tecnoloxicamente capaz de diminuír as emisións de dióxido de carbono, así como de reducir significativamente o consumo de combustible. Isto obedece a que é posible unha mellora significativa da eficiencia dos sistemas de propulsión mariña xa que contribúe á redución do consumo de combustibles fósiles do sector do transporte marítimo en xeral.

A mellora mencionada pódese aproveitar usando a calor residual esgotada e / ou rexeitada polo sistema de propulsión e os seus motores auxiliares asociados. A tecnoloxía convencional ata o de agora optaba polo aproveitamento dos gases de combustión do escape, xa que non se podía usar calor a temperatura máis baixa.

Convencionalmente, a tecnoloxía de punta neste campo usa ORC avanzados en varias estruturas combinativas como unha das diferentes tecnoloxías WHRS que se están empregando para o aproveitamento de calor de medio e / ou baixo grao residual polo sistema de propulsión que non pode ser empregado polos propulsores.

Por outra banda, a redución do consumo de combustible, que significa aumentar a eficiencia térmica sometida á redución das emisións de dióxido de carbono, debe mellorarse na medida do posible, baseado en que este criterio é o suficientemente relevante como para emprender o desenvolvemento de tecnoloxías dirixidas a mellorando a eficiencia térmica dos motores térmicos empregados ata o de agora.

Non obstante, nesta tese investígase un novo paradigma de WHR co obxectivo de aumentar a eficiencia térmica que se reduce o consumo de combustibles fósiles e as emisións de dióxido de carbono.

Estas melloras están asociadas ás seguintes contribucións propostas e realizadas nesta tese:

- Recuperación de calor de baixa temperatura mediante o deseño dunha estrutura de recuperación capaz de recuperar a calor residual de temperaturas medias e baixas rexeitadas ata o momento, procedentes de gases de escape, refrixerador de aire de admisión, refrixerador de auga de camisas, e refrixerador de aceite de lubricación.

- Motores térmicos máis eficientes mediante o deseño de análise e realización dunha proba de concepto de ciclos térmicos baseados en procesos pechados que accionan engadindo e extraendo calor sometido a un modo de interacción calor-traballo innovador.

Ademais, os motores térmicos propostos caracterízanse por un ciclo térmico realizable fisicamente composto por dous procesos isocóricos pechados consistentes na adición e extracción de calor cara / dende o fluído de traballo e dous procesos adiabáticos pechados que provocan un traballo mecánico útil, consistente respectivamente por unha expansión adiabática pechada e compresión adiabática pechada a base de contracción do fluído de traballo.

Os principais resultados son:

- Unha arquitectura de potencia capaz de converter a calor de baixo grao en traballo mecánico cun factor de utilización superior a calquera tecnoloxía dispoñible ou teórica.
- Un ciclo térmico que supera calquera outro ciclo para calor de baixo grao en termos de eficiencia e factor de utilización.

Résumé

Cette thèse explore le potentiel de la chaleur de faible et moyenne qualité dans différents cycles thermodynamiques utilisés pour convertir la chaleur perdue en travail mécanique. L'objectif de cette thèse est d'étudier l'état de l'art des cycles thermodynamiques utilisés pour récupérer la chaleur de bas grade et proposer des améliorations.

La pertinence de la recherche d'applications de chaleur à faible température ou de chaleur résiduelle est qu'une grande quantité d'énergie thermique est disponible à un coût négligeable dans la plage de températures moyennes et basses, avec l'inconvénient que les cycles thermiques existants ne peuvent pas utiliser efficacement cette chaleur à basse température disponible en raison de leur faible efficacité.

La présente thèse propose une approche différente, analyse la récupération de chaleur à faible grade d'un point de vue thermique thermodynamique et compare leur efficacité thermique. Les cycles ORC et Rankine examinés montrent des efficacités faibles similaires. En revanche, les cycles fermés, sans condensation, qui échangent de la chaleur à volume constant et convertissent l'énergie thermique en travail mécanique de manière adiabatique, ont une configuration qui permet une exploitation efficace de la chaleur de faible qualité. Des stratégies de récupération de chaleur thermique sont également analysées et plusieurs systèmes non conventionnels sont proposés pour exploiter la chaleur de faible qualité dans la machine proposée.

Deux facteurs pertinents exercent une forte influence sur le critère pour décider de la structure des installations de propulsion marines :

• La réduction de la consommation de carburant qui subit l'augmentation de l'efficacité thermique des moteurs thermiques associés.

• L'émission de dioxyde de carbone due à la consommation de combustibles fossiles

D'une part, les émissions de dioxyde de carbone provenant de la combustion de combustibles fossiles utilisés dans les centrales de propulsion marines et les centrales électriques associées sont une préoccupation pour les populations du monde entier et cela est dû aux impacts

environnementaux engendrés par ces émissions de gaz à effet de serre, ce qui contribue à modifier la composition de l'atmosphère dans un sens négatif exagéré. Ainsi, les méthodes nocives peuvent être remplacées par des stratégies de conception alternatives ou au moins pourraient être atténuées par la réduction des émissions de dioxyde de carbone, ce qui pourrait être réalisé en modifiant les stratégies de conception technologique et les méthodologies opérationnelles. La technologie de récupération de chaleur disponible qui peut être utilisée sur les systèmes de récupération de chaleur résiduelle (WHRS) de qualité moyenne ou faible est un de ces moyens qui est technologiquement capable de réduire les émissions de dioxyde de carbone ainsi que de réduire considérablement la consommation du combustible. Ceci obéit au fait qu'il est possible une amélioration significative de l'efficacité des systèmes de propulsion marine puisqu'elle contribue à la réduction de la consommation de combustibles fossiles du secteur du transport maritime en général. L'amélioration mentionnée peut être réalisée en exploitant l'énergie thermique perdue épuisée et / ou rejetée par le système de propulsion et ses moteurs auxiliaires associés. La technologie conventionnelle a jusqu'à présent opté pour la récupération des gaz de combustion, car la chaleur à basse température ne pouvait pas être utilisée.

Conventionnellement, la technologie de pointe dans ce domaine utilise le cycle organique avancé de Rankine (ORC) sous plusieurs structures combinatoires comme l'une des différentes technologies WHRS qui sont utilisées pour exploiter les pertes de chaleur de moyenne et / ou de faible qualité par le système de propulsion qui ne peut pas être utilisé par les moteurs de propulsion par lui-même.

En revanche, la réduction de la consommation de carburant, c'est-à-dire l'augmentation de l'efficacité thermique subissant la réduction des émissions de dioxyde de carbone, doit être améliorée autant que possible en se fondant sur le fait que ce critère est suffisamment pertinent pour entreprendre le développement de technologies visant à améliorer l’efficacité thermique des moteurs thermiques utilisés jusqu'à présent.

Néanmoins, dans cette thèse, il est étudié un nouveau paradigme de WHRS visant à augmenter l'efficacité thermique qui subit une réduction de la consommation de combustibles fossiles et des émissions de dioxyde de carbone.

Ces améliorations sont associées aux contributions suivantes proposées et réalisées dans cette thèse :

• Récupération de chaleur à basse température grâce à la conception d'une structure de récupération capable de récupérer la chaleur résiduelle des températures moyennes et basses rejetées jusqu'à présent, provenant des gaz d'échappement, du refroidisseur d'air de combustion, du refroidisseur d’eau du moteur et du refroidisseur d'huile de lubrification.

• Moteurs thermiques plus efficaces grâce à la conception, l'analyse et la réalisation d'une validation de principe des cycles thermiques basés sur des processus fermés fonctionnant par ajout et extraction de chaleur subissant un mode innovant d'interaction chaleur-travail.

De plus, les moteurs thermiques proposés sont caractérisés par un cycle thermique physiquement réalisable composé de deux processus isochoriques fermés consistant à ajouter et extraire de la chaleur vers / depuis le fluide de travail et deux processus adiabatiques fermés qui subissent un travail mécanique utile, consistant respectivement en une expansion adiabatique fermée et compression adiabatique fermée basée sur la contraction du fluide de travail.

Les principaux résultats sont :

• Une architecture de puissance capable de convertir la chaleur de faible qualité en travail mécanique avec un facteur d'utilisation supérieur à toute technologie disponible ou théorique.
• Un cycle thermique surpassant tout autre cycle de chaleur de faible qualité en termes d'efficacité et de facteur d'utilisation.

Related publications:

Steven Iglesias Garcia, Ramon Ferreiro Garcia, Jose Carbia Carril, Denis Iglesias Garcia, Critical review of the thermal efficiency in different power combined cycle architectures, Energy Conversion and Management, Volume 148, 2017, Pages 844-859, ISSN 0196-8904, https://doi.org/10.1016/j.enconman.2017.06.037.

(http://www.sciencedirect.com/science/article/pii/S0196890417305794)

Keywords: Combined cycle; Low grade heat; Power plant

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A review of thermodynamic cycles used in low temperature recovery systems over the last two years, Renewable and Sustainable Energy Reviews, Volume 81, Part 1, 2018, Pages 760-767, ISSN 1364-0321, https://doi.org/10.1016/j.rser.2017.08.049.

(http://www.sciencedirect.com/science/article/pii/S136403211731198X)

Keywords: Cascade ORC; Organic flash cycle; Low-grade heat; Residual heat; ORC; Waste heat

Patents with prior examination:

P [1] Conversor de movimiento alternativo discontinuo a rotativo continuo

Page bookmark ES2642812 (A1) - Conversor de movimiento alternativo discontinuo a rotativo continuo

Inventor(s): FERREIRO GARCIA RAMÓN [ES]; CARBIA CARRIL JOSÉ [ES]; IGLESIAS GARCIA STEVEN [ES] +

Applicant(s): UNIV CORUNA [ES] +

Classification:

- international: F16H19/02; F16H25/08; F16H47/02

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P [2] Conversor de fuerza alternativa discontinua a par rotativo continuo y procedimiento de operación del mismo

Page bookmark ES2641908 (A1) - Conversor de fuerza alternativa discontinua a par rotativo continuo y procedimiento de operación del mismo

Inventor(s): FERREIRO GARCIA RAMÓN [ES]; CARBIA CARRIL JOSÉ [ES]; IGLESIAS GARCIA STEVEN [ES] +

Applicant(s): UNIV CORUNA [ES] +

Classification:

- international: F02G1/04; F15B15/06

- cooperative: F02G1/04; F15B15/06 more

Application number: ES20160000386 20160511

Priority number(s): ES20160000386 20160511

Also published as: ES2641908 (B2)

P [3] Ciclo combinado de motor de combustión interna y máquina alternativa de doble efecto, procesos cerrados y movimiento continuo

Page bookmark ES2639589 (A1) - Ciclo combinado de motor de combustión interna y máquina alternativa de doble efecto, procesos cerrados y movimiento continuo

Inventor(s): FERREIRO GARCIA RAMÓN [ES]; CARBIA CARRIL JOSÉ [ES]; IGLESIAS GARCIA STEVEN [ES] +

Applicant(s): UNIV CORUNA [ES] +

Classification:

- international: F02G5/02; F02G5/04

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Acronyms and Nomenclature

Acronyms

C3L	Closed processes-based Trilateral
C4L	Closed processes-based Quadrilateral
CORC	Cascade Organic Rankine Cycle
EU	European Union
F-T	Transfer Compressor
G	Generator
HP	High pressure side
HUF	Heat utilisation factor
HWF	Heat Work Flow
HX	Heat Exchanger
ICE	Internal Combustion Engine
LGWH	Low-grade waste heat
LNG	Liquefied Natural Gas
LP	Low pressure
ME	Main Engines
N	Cannot Deliver work using heat
ORC	Organic Rankine Cycle
PE	Propulsion Engine
PSP	Propulsion Shaft Power
PTG	Power Turbine Generators
PU	Power Unit
PM	Particulate Matter
RH	Heat Recovered
RC	Steam Rankine Cycle
RORC	Regenerative Organic Rankine Cycle
SOFC	Simple Organic Flash Cycle
SORC	Simple Organic Rankine Cycle
ST-PT	Steam turbine, power turbine and generator

STG	Steam Turbine and Generator
TF	Thermal Fluid
V	Valve
WHRS	Waste Heat Recovery System
WE	Waste Heat
Y	Delivers work using heat

Nomenclature

η_f	Fan Efficiency
η_L	Lenoir efficiency (%)
η_{th}	Thermal efficiency (%)
η_{net}	Net efficiency (%)
γ	Polytropic index
$\dot{Air}_{SC}$	Scavenge Air Flow (kg/s)
C_P	Specific heat capacity at constant pressure (kJ/kg·K)
C_V	Specific heat capacity at constant volume (kJ/kg·K)
C_F	Carnot Factor
E_{HF}	Final high temp energy (kJ)
E_{HI}	Initial high temp energy (kJ)
E_i	Inlet energy (kJ)
E_{LF}	Final low temp energy (kJ)
E_{LI}	Initial low temp energy (kJ)
E_o	Outlet energy (kJ)
$\dot{Gas}_{Exh}$	Exhaust Gas Flow (kg/s)
h	Enthalpy (kJ/kg)
Δh	Enthalpy difference (kJ/kg)
$\dot{m}$	Mass flow rate (kg/s)
$\dot{m}_{oil}$	Oil flow (kg/s)
$\dot{m}_{water}$	Water flow (kg/s)
p	Pressure (kPa)
p_i	Initial Pressure (K)
p_R	Pressure Ratio
p_{sy}	Pressure system (kPa)
p_{su}	Pressure surroundings (kPa)
$\dot{P}_{th}$	Thermal power (W)
Q	Heat (kJ)
Q_{UN}	Unrecovered Heat (kJ)
Q_{RES}	Residual Heat (kJ)
Q_S	Supplied Heat (kJ)
Q_R	Recovered Heat (kJ)
q	Specific heat (kJ/kg)
q_o	Specific heat output (kJ/kg)
q_i	Specific heat input (kJ/kg)
s	Specific entropy (kJ/kg·K)
s_i	Initial specific entropy (kJ/kg·K)
S	Entropy (kJ/K)
S_{final}	Final system entropy (kJ/K)
S_{init}	Initial system entropy (kJ/K)
S_{sys}	System entropy (kJ/K)
S_{gen}	Generated entropy (kJ/K)
S_{trans}	Transferred entropy (kJ/K)
S_U	Entropy isolated system (kJ/K)
ΔS	Entropy difference (kJ/K)
T	Temperature (K)
T_i	Initial Temperature (K)
T_{exh}	Exhaust Temperature (K)
T_{SC}	Scavenge Temperature (K)
T_H	High cycle temperature
T_L	Low cycle temperature
T_{water}	Top Temperature of a cycle (K)
T_{oil}	Oil Temperature (K)
T_i	Initial Temperature (K)
ΔT	Temperature difference (K)
u	Specific internal energy (kJ/kg)

ΔU	Internal energy difference (kJ)
u_i	Initial specific internal energy (kJ/kg)
Δu	Specific internal energy difference (kJ/kg)
V	Volume (m^3)
ΔV	Volume difference (kJ/kg)
v	Specific volume (m^3/kg)
v_i	Initial specific volume (m^3/kg)
w	Specific work (kJ/kg)
W	Work (kJ/kg)
W_R	Work converted from heat recovered from a thermal plant (kJ)
w_R	Work out ratio
W_N	Net Work (kJ)
w_i	Specific inlet work (kJ/kg)
w_n	Net specific work (kJ/kg)
w_o	Specific outlet work (kJ/kg)
w_i	Specific inlet work (kJ/kg)
w_{exp}	Specific expansion work (kJ)
w_{cont}	Specific contraction work (kJ/kg)

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1 Introduction

1.1 Aim

Traditionally, for the harvesting of wasted low-grade heat, heat recovery techniques based on organic Rankine cycles among other useful options have been used, which are characterized by their installation and maintenance costs as well as an inherent low performance, all of which constituting the necessary stimulus for undertaking the search for more efficient solutions.

Thus, with the research work that gives rise to this thesis it is aiming at improving the performance of ship's propulsion plants including propulsion engines efficiency and overall heat utilisation factor, for which the tasks of designing developing and implementing advanced waste heat (WH) recovery techniques suitable are to be applied on ship's propulsion plants. Such new advanced WH recovery techniques concern to innovative closed-processes based thermal cycles characterized by doing useful mechanical work due to extracting heat to a heat sink, instead of rejecting heat as occurs in conventional thermal cycles (cycles that perform work by adding and rejecting heat).

Motivation:

The motivation is due to the evident fact related to the observation of the environmental deterioration derived from the excessive use of fossil fuels. Furthermore, not only does such abuse result in the global depletion and pollution responsible for the general health of the earth, but it also leads to an economic cost associated with the shortage of energy resources. Therefore, the endless list of concepts that motivate research in lines that clearly contribute to mitigating such a colossal deterioration of nature is summarized in the following concepts:

- The evidence of environment degradation
- Non-renewable fossil fuels resource depletion
- The Impact of Carbon dioxide
- The impact of contaminant emissions
- The global warming and its consequences

- The impact of marine transport emissions on the global environment: atmosphere and oceans

Strategy:

Consequently, to undertake a research and development process that brings together the skills to solve the problems posed requires a paradigm shift of a scientific and technological nature related to thermal engines intended for the conversion of thermal energy into mechanical energy featured by higher performance than existing technologies.

Such a paradigm shift concerns the exploitation of the concept of heat-to-work conversion based on the transfer of heat from an existing power source to a sink through heat extraction at no cost. In addition to the heat extraction strategy, the heat-to-work conversion processes must be closed.

Consequently, based on the established strategy, a series of development tasks are proposed that lead to the following chapters:

1.2 Chapter description

Chapter 1 – Introduction

The study carried out for the realization of this thesis is organized in such a way that starting from the motivation and strategy that justifies a review of the state of the art, it focuses on the definition of objectives and establishment of search strategies for alternative solutions to conventional, which is developed in chapter 1.

Chapter 2 - Review of the existing technology related to WH recovery techniques on ship's propulsion plants

Chapter 2 summarises the state-of-the-art of the Low-Grade WH (LGWH). It also reviews, from a thermodynamic point of view, the technology of commercial and published systems.

Chapter 3 – Suggested solutions

Chapter 3 describes the thermal cycle used in this thesis. It analyses them from a thermodynamic perspective, starting with the analysis of the Lenoir power cycle. In addition, it justifies the high efficiencies reached by closed, non-condensing cycles that perform work adding and rejecting heat.

Chapter 4 – 4 Proposed Alternative Engine Structures and Implementation

Chapter 4 lists and describes the patents gotten during the research and development made when looking into non-condensing, closed processes expansion-contraction based thermal cycles. There are three patents: a combined cycle and two mechanical devices. These inventions are designed to put into practice the thermal cycles included in this thesis.

Chapter 5 – Case studies: The implementation of WHRS on ship's propulsion plants

Chapter 5 Analyses sources of marine WH, involves a system based in power units, which manage to perform work following the thermodynamic paths set. Efficiencies for a specific case are calculated with different configurations using Air, Hydrogen and Helium as WF. Moreover, their thermal efficiency is analysed and compared when using the recommended power units. After that, results obtained are used to compare the heat utilisation factor obtained for an industrial engine.

Chapter 6 – General conclusions

Chapter 6 discusses the work done with the technology brought in, starting from chapter 1 and comes to a conclusion about the research and development performed in this thesis. This section also suggests further investigations.

2 Review of the existing technology related to WH recovery techniques on ship's propulsion plants

The marine industry wastes an important amount of heat at different temperatures. The heat of the scavenge air and exhaust of marine engines are attractive on account of its temperature. Iglesias et al. [1] observed that Organic Rankine Cycles (ORC) are a common solution to convert low-grade heat due to the inherent features of their WFs. According to them the main limitation to convert low-grade heat is the poor efficiency of the cycles used for this specific purpose. Iglesias et al. [2] found that ORCs coupled to Internal Combustion Engines or Steam Rankine Cycle (RC) are a frequent solution to maximise the amount of energy converted into mechanical work. However, for feasibility reasons, expanding directly the fluids of WH or using a simple RC are also common solutions. Both manufacturers reviewed use a direct expansion (Brayton) and steam turbines (Single or dual pressure Rankine).

2.1 Low-Grade WH (LGWH)

Forman et al. [3] estimated that 72 % of the primary energy consumption is dissipated. The main characteristic of most of the wasted heat is its low temperature. Therefore, even if the amount of wasted energy is high, the thermodynamic efficiency of the transformation of this energy is low due to thermodynamic constrains. There are differences regarding the temperature of the heat wasted. For instance, the electricity sector dissipates heat at the lowest temperatures, the industrial and the transport sectors waste a significant amount of heat at temperatures higher than 573 K. If the wasted heat is used, less primary fuel will be needed and air pollution will decrease.

Lecompte et al. [4] concluded that, in most of the cases, WH has a low temperature (<350 C) and it is carried as a fluid. In some applications, WH cannot be fully exploited. For example, the dew point of exhaust gases in combustion engines is an upper limit for the heat recovery. According to Ebara et al. [5], if the temperature is below the dew point the gases will condense and will potentially create corrosive dissolutions in the exhaust system. Sarkar, J. [6] classified low-grade heat as: geothermal heat, heat from combustion engines, heat from industrial processes, solar energy and alternative sources of low-grade heat. Forman and co-workers [3] studied the amount of wasted heat in the most relevant industrial activities. They found that 52 % of the world's energy is misused through exhausts or effluents, and they also found that the industrial and transport sectors waste an important quantity of energy at more than 573 K. In contrast, they determined that the electricity sector loses most of its heat at temperatures below 373 K.

Industrial marine power plants lead to the 3.3 % of annual CO_2 emissions according to Suarez S. et al. [7]. Baldi F et al. [8] found that depending on the system used to recover heat in marine engines, between an additional 5 % and 15 % of fuel can be recovered or used when recovering its WH. Bouazzaoui S. et al. [9] found that low-temperature energy can be used for cooling using this energy to run refrigeration equipment. They traced an important improvement when improving the cooling capacity of a ship.

Cullen et al. [10] classified the potential of energy according to its economic perspectives due to market forces, its technical potential, which takes into account its practicality and the theoretical potential that considers thermodynamic factors. The temperature will be used to quantify the thermodynamic potential of the WH energy.

Consequently, from a thermodynamic scope and assuming a constant low temperature, the temperature is proportional to its thermodynamic efficiency. According to Forman [3], the WH from the commercial, industrial and transport sectors have the highest temperatures. Thus, these sectors have the highest thermodynamic potential in their wasted energy.

Tab. 2.1 shows the vast amount of heat wasted at considerably high temperature from a thermodynamic point of view. For instance, the G45 – 11.12 MW [11] engine wastes roughly 11.1 MW. This shows clearly the potential to use WH Recovery Systems (WHRS). Furthermore, the improvement of the fuel efficiency is not lineal. If a ship becomes more efficient, it will need less fuel to do the same trip. Therefore, the weight of the ship will decrease and less primary energy will be required to do a specific task. Thus, fuel savings for marine engines can have interesting payback ratios.

Tab. 2.1 Energy wasted in different MAN® engines.

Model-Power at full load	LO (kW)	FW (kW)	SC (kW)	EXH (kW)
S90 – 69.72 MW [11]	5560	8950	28070	30800
G95 – 82.44 MW [11]	6450	10750	30320	30709
G80 – 42.39 MW [11]	3350	5580	16200	16957
G70 – 29.12 MW [11]	2390	3900	10650	11811
G60 – 22.70 MW [11]	1770	3010	8800	9212
G45 – 11.12 MW [11]	930	1520	4080	4574

Pollution influences human health negatively. Di Natale [12] settled that the maritime transport industry produces massive emissions of pollutants, such as: Particulate Matter, Nitrogen Oxides and sulphur dioxide. As claimed by Fotourehchi [13], a sensible amount of PM can determine life expectancy. Thus, the use of WH energy can directly reduce human diseases related to pollution. On the report of the European Union (EU) there is potential to produce 20 TWh from wasted heat, which is equivalent to 4.8 % of the EU electricity consumption [14].

2.2 Power thermal cycles used to recover WH

2.2.1 Organic Rankine Cycle (ORC) architectures

ORC are Rankine-based systems that use an organic WF. Lecompte et al. [4] consider the ORC to be a validated way to exploit low-grade heat. The following applications and architectures are discussed. Kölsch B. et al. [15] explored ways to use ORC to exploit low-grade heat from diesel engines. They discovered that Methanol offers the best performance. In their analysis, they took into consideration risks and size constraints. Song, J. et al. [16] explored the feasibility of such systems, and they concluded that if the ORC is optimized, it is a realistic option to recover heat in industrial diesel engines. Suarez. et al. [7] proved that the ORC provides with better performances than RC.

2.2.2 Simple ORC (SORC)

It is the basic architecture of a RC. Fig. 2.1 illustrates its architecture and Fig. 2.2 shows its thermal cycle in a temperature-entropy diagram. This architecture wastes a large proportion of heat in the condenser which reduces the efficiency of the cycle. It has been used to recover WH in the cement industry, where around 40 % of the produced heat is lost

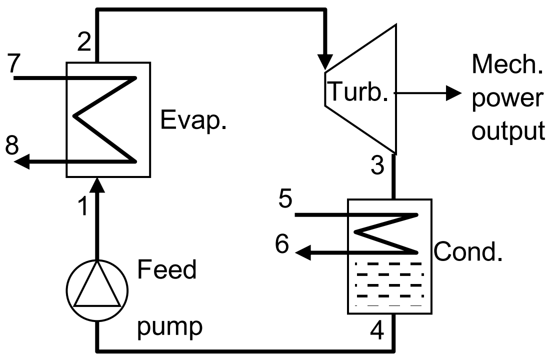

Fig. 2.1. Schematic diagram of a SORC.

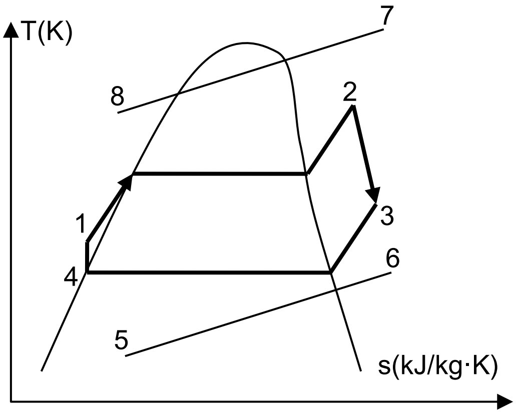

Fig. 2.2. T-s diagram of a SORC.

Deethayat et al. [17] analysed a commercial SORC system and obtained cycle efficiencies between 10 % and 7 % for different conditions. They also found that there is no significant difference between the theoretical cycle and the real cycle for different tested conditions. Tab. 2.2 shows relevant data of the empirical test. For instance, there is a 10 % efficiency for Condition 1. Tab. 2.2 shows how the hot water inlet temperature in the vaporizer and the thermal efficiency of the cycle.

Tab. 2.2. Empirical results of a SORC obtained by Deethayat et al. [17]. Condenser Cold water inlet (K) is considered 301.15 for the three conditions.

Empirical results of SORC [10]	Condition 1	Condition 2	Condition 3
$T_{water\ inlet}$ (K)	389.15	380.95	370.15
η_{cycle} (%)	9.40	8.81	7.37

Pu, W. and co-workers [18] analysed a real scale SORC using low grade thermal energy and compared the performances of R245fa and HFE7100 as WFs. The maximum power output was 1979 W with a mass flow of 0.2069 kg/s at 366 K using R245fa as a WF. Under these conditions the overall efficiency achieved was 4.01 %. Deethayat et al. [17] got considerably higher efficiencies, probably because the tests were conducted under different pressures.

Usman and co-workers [19] studied the effect of the application of a SORC in a light duty vehicle setting that such a system is feasible. The efficiency of the system increases at high speed yet is poor at low speed. The authors claimed to have improved the 130 kW Internal Combustion Engine (ICE) between 10.16 % and 15.95 %. However, so that efficiency can be realistic, it is necessary to bear in mind factors, such as additional back pressure and additional weight. Thus, the real maximum efficiency obtained, taking these factors into account, is 5.82 %. In a similar way, Shi and co-workers [20] developed a numerical simulation of a SORC applied to a vehicle. Their design has a significant difference compared to other proposed SORCs, as it has a controlled exhaust gas recirculation loop. The heat through the vaporizer is controlled and, as a result the cycle efficiency, output power and overall energy efficiency are increased.

Tian H. et al. [21] studied SORC with different fluids. They observed that depending on the vaporizer pressure, ORC running on a particular diesel engine exhaust energy (Exhaust at 792 K) can have an efficiency between 2 % and 17 % depending on the WF. that R141b, R123 and R245fa offered the best performances ranging from 16.60 % to 13.30 %. Working Fluid (WF) optimisation depends directly on the top temperature of the cycle. For instance, Wang E. et al. [22] concluded that for a low temperature WHRS (below 523 K) R152a offer best performances, reaching up to 12 % from 473 K. However, R600 or R601 can slightly outperform it in a small range of temperatures.

2.2.3 Regenerative ORC (RORC)

The Regenerative ORC (RORC) is a SORC with a regenerator. This configuration allows the recovery of a quantity of heat from the outlet of the expander, as shown in Fig. 2.3. In addition, it reduces the requirements of the condenser. In contrast, it can be counterproductive to recover the maximum heat possible in the regenerator. Fig. 2.4 illustrates the thermal process of a RORC in a temperature-entropy diagram. The heat is recovered between points 3 and 4 in the regenerator. If acid gases are cooled below their dew point, they will condense and potentially produce corrosion in the regenerator. Consequently, temperatures in heat exchangers have to be studied. Yang [23] examined these temperatures accurately using a genetic algorithm. The RORC has been thought for diverse WH sources. Wang [24] suggested a RORC running on solar energy. Idrus Alhamid, M. [25] reviewed a RORC using geothermal heat to power a RORC.

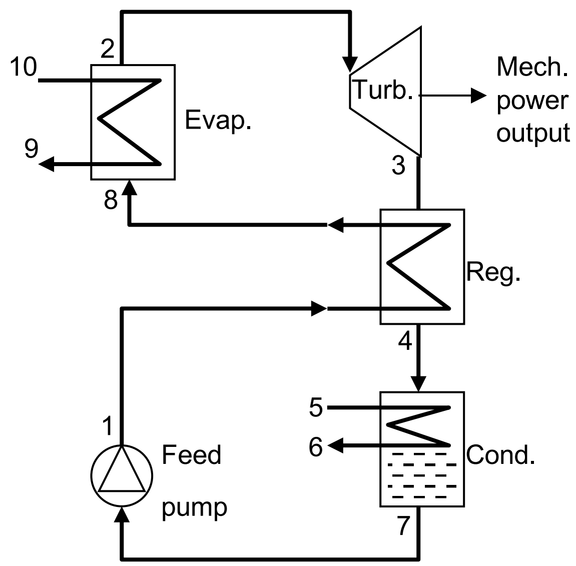

Fig. 2.3. Regenerative ORC (RORC).

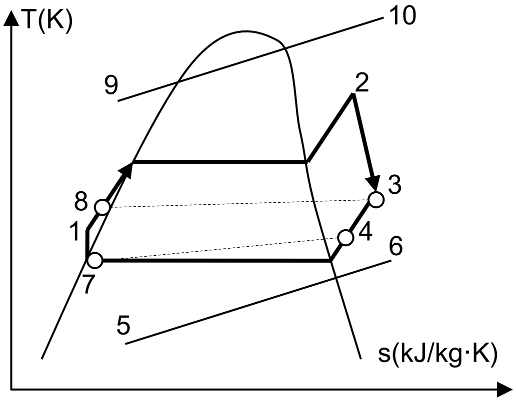

Fig. 2.4. T-s diagram of a RORC.

Molés et al. [26] evaluated empirically a RORC using HCFO-1233zd-E and HFC-245fa as WFs. Tab. 2.3 shows the results obtained for both fluids. In their opinion, the HCFO-1233zd-E produces 20 % less mass flow rate and, for this reason, it reflects a poor general performance compared to the other fluid. Results shown in Tab. 2.3 are maximum values of the conducted tests. Net electrical efficiencies vary, according to the conditions, between 5 % and 9.7 % for both WFs.

Tab. 2.3. Maximum values of input and output of the 160 empirical RORC tests conducted by Molés et al. [26].

RORC fluids [17]	$\dot{P_{in}}$ (W)	$\dot{P_{out}}$ (W)
HCFO-1233zd-E	9900	960
HFC-245fa	12000	1090

Desideri et al. [27] compared empirically two different fluids in a RORC. The facility used was not designed specifically for the tested fluids and the efficiency was around 8 % for both fluids. The WFs are R245fa and SES36. They obtained an efficiency of 4.01 % with a SORC. This shows the important efficiency improvement produced by the regenerator in the ORC. The authors concluded that, even if both fluids have the same thermal efficiency, the output is 17 % higher using R245fa.

Min Kim and his team [28] proposed three different configurations of RORC to recuperate WH from a car. Isentropic efficiencies range from 8.2 % to 10.8 %. The first configuration, which is a RORC with a preheater, this configuration allows recovery of part of the coolant heat. The second configuration vaporises the WF of the cycle in the preheater, barely improving the isentropic efficiency. The third configuration includes two regenerators and a preheater. Therefore, the use of WH from the engine is maximised.

The application of RORCs in ships to recuperate WH has also been studied. Mondejar [29] simulated the application of a RORC in a passenger vessel. They claim that a RORC running on Benzene with a preheater, a vaporizer and a super-heater can deliver 22 % of the power demand with an average thermal efficiency of 22 %. Nevertheless, the so-called RORC only uses exhaust gases as a heat source. In line with them, the exhaust heat accounts for 32.5 % of the fuel energy and the required power demand of the ship accounts for 13.74 % of the fuel energy. Thus, they can extract 22 % of the 13.74 %. This is equal to 3 % of the initial fuel energy. As a result, the net efficiency of the cycle proposed is 9.23 %. This value is similar to the values obtained by [27] and Min Kim et al. [28].

2.2.4 Cascade ORC (CORC)

The simplest version of a cascade ORC (CORC), shown in Fig. 2.5 has at least two separate stages, being the condenser of the first stage the vaporizer of the next one. These thermodynamic cycles use a different WF in each stage. Fig. 2.6 illustrates the diagram of the CORC presented in Fig. 2.5 in a temperature-entropy diagram. Between processes 2-3 wasted heat from the first cycle is transferred to the second cycle through the processes 8-5. Having different WFs allows extraction of an efficient thermal energy. In contrast, as shown in Fig. 2.5, CORCs also use a SORC, in which the vaporizer of the added SORC is the condenser of the upper cycle in a Temperature-entropy diagram.

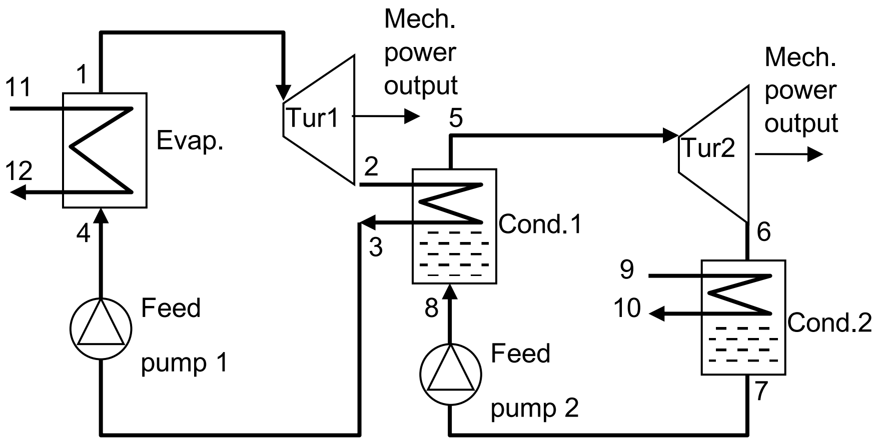

Fig. 2.5. Cascade ORC (CORC).

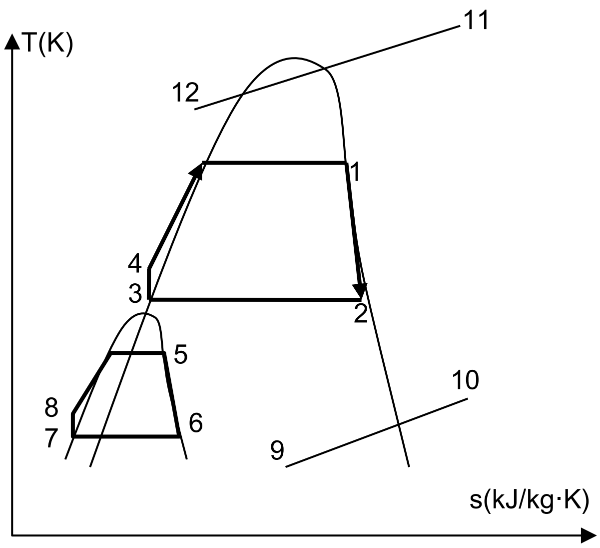

Fig. 2.6. T-s diagram of a Cascade ORC (CORC).

Li et al. [30] designed a novel architecture based on a CORC combining heat from the sun and cold from Liquefied Natural Gas (LNG). The WFs selected were isopentane/R125 and the proposed architecture had three stages. The maximum equivalent efficiency obtained was 5.99 %. The LNG cold energy is a recognised way to exploit low-grade heat sources, as stated by Gomez et al. [31,32].

Panesar et al. [33] applied a two stage CORC to a Heavy Duty Diesel Engine Truck using water and R245fa as a WF. The CORC proposed was optimised for this precise application not forgetting factors such as weight and size. The overall thermal efficiency of the engine was improved by 2.2 % applying the CORC. The heat source of the CORC came from the engine coolant and the exhaust gases. They concluded that the proposed CORC shows an average 20 % improvement in system power. They claim that the principal reason for this important improvement is the dual-pressure nature of the system.

2.2.5 Other Rankine Configurations

Sung et al. [34] published a dual-loop ORC composed of a SORC and a RORC. This system exploits the cold of the LNG and the WH from a dual-fuel marine engine. The high temperature cooling water is used in the preheater of the RORC and in the vaporizer of the SORC. On the contrary, the LNG cold is only included in the SORC and the heat from the exhaust gases only in the RORC. N-pentane and R125 were used as WFs. They managed to extract 906.4 kW with this novel dual-loop ORC, which is 5.17 % of the original power of the dual-fuel marine engine. The system proposed by Mondejar [29] produced 22 % of the power demand applying only a RORC recuperating WH from the exhaust gases of a marine main engine and auxiliary engine. However, Sung et al. [34] used the heat from the cooling water and exhaust gases from the main engine and the cold from the LNG. There is an important difference between both works, the temperature of the exhaust gases is significantly lower in the engine proposed and they do not consider using auxiliary engines.

Ferreiro et al. [35] introduced an unconventional power plant based on RC. The suggested cycle is basically a (CORC) which uses organic and non-organic WFs. In addition, it uses LNG cold as a heat sink. They propose ambient air, water or low-grade heat as a heat source. Thus, it is a valid way to exploit low-grade residual heat. The Cascade cycle proposed has two methane stages and an argon stage.

2.2.6 Simple Organic Flash Cycles (SOFC)

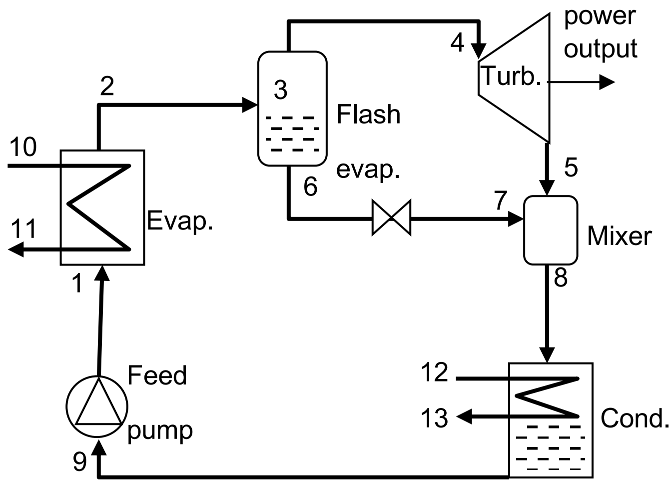

Fig. 2.7. Simple Organic Flash Cycle (SOFC).

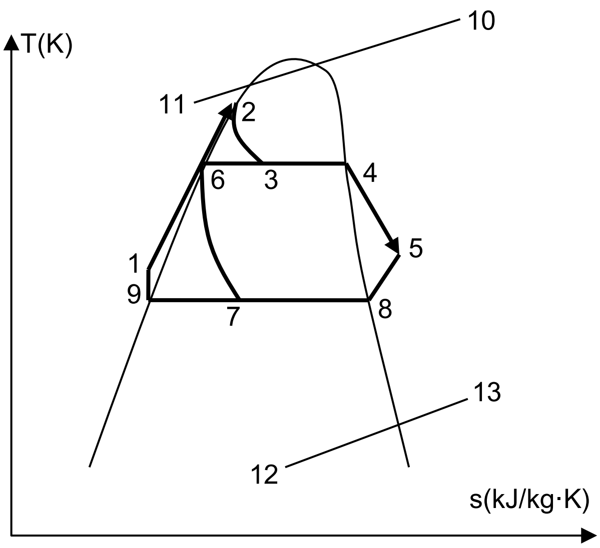

Fig. 2.8. T-s diagram of a SOFC.

The Simple Organic Flash Cycle (SOFC) is similar to a SORC. In contrast, the WF of the ORC is flash evaporated in the Mixer, where two phases of the fluid are present. The flash vaporizer is illustrated in Fig. 2.7 It mixes heat from states 3 and 6 and its output is state 4. The gas phase of the fluid, in state 4, is expanded converting its energy into mechanical work. This process characterises this thermal cycle and its thermal evolutions can be observed in Fig. 2.8. SOFC also has similarities to RORC. The heat from state 6 in Fig. 2.8 is recovered from the turbine outlet, in a process similar to the regenerative process of a RORC. Ho et al. [36] opted for several flash architectures, which improve the efficiency of the SOFC. The first idea is a double flash cycle, where the liquid phase of the mixer is flash evaporated again in a second mixer where the gas phase is expanded in a low-pressure expander. This configuration can produce between 15 % and 20 % more work than a SOFC.

Flash cycles with multiple flash stages have also been analysed by Sarr et al. [37] for geothermal use. They claim that a multiple stage flash evaporation gives more output work. In addition, it decreases the required cooling capacity of the plant and the moisture level decreases, which is particularly beneficial in low-pressure expanders.

The second cycle proposed by Ho et al. [36] is an SOFC, which recuperates heat at the turbine exhaust and the saturated liquid of the flash vaporizer in a mixer in a low pressure expander. As a consequence, the fluid is less superheated and more fluid is expanded. They also mentioned a two-phase Organic Flash Cycle in which the WF is expanded before the flash vaporizer. These cycles are in the research and development phase and authors assume isentropic efficiencies. In order to improve their efficiency, they suggest a two-phase ORC, in which the heat of the saturated liquid and the fluid exhaust of the expander are exploited throughout a mixer in a low-pressure turbine.

Lee et al. [38] compared organic flash cycles with a two-phase expander and ORC. Using R245fa, R123, and o-xylene as WFs, they concluded that the efficiency of both cycles depends on the WF and heat source temperature. However, in accordance with their simulations, organic flash cycles are more suitable. for low-grade heat sources.

2.3 Commercial marine WHRS

Shipping uses commonly industrial Diesel engines. WH is commonly used to produce steam or to raise the mechanical conversion efficiency. This section reviews commercial systems from Wärtsilä® and MAN® that are intended to increase the mechanical energy conversion. From a thermodynamic point of view the following systems are Brayton and Rankine based. For decades research and development of these systems has been focused on the efficiency of its components rather than the thermodynamic aspect of the system. Therefore, its efficiency is thermodynamic limited. It ranges typically between 4 % to 10 %. As a consequence, their design is intended to be used in ships with a considerable WH production and from certain load. Fuel savings and pollution reduction are the most beneficial aspects of these systems. Shu G. et al. [39] found that in a specific case, a WHRS can save up to 2.25 million US$/year.

The WHRS of the mentioned manufacturers are based on exactly the same strategy, which consists in using heat from the water cooling jackets, scavenge air and exhaust gas to generate vapor in a single or dual gas boiler. In addition, they all offer the possibility to bypass part of the exhaust gas, typically 10 % of the flow to expand directly into an exhaust-turbine. The main components are:

- Dual/Single pressure exhaust gas boiler

- Steam turbine generator

- Exhaust gas power turbine

- Heat exchangers for the main engine scavenging air and the jacket water

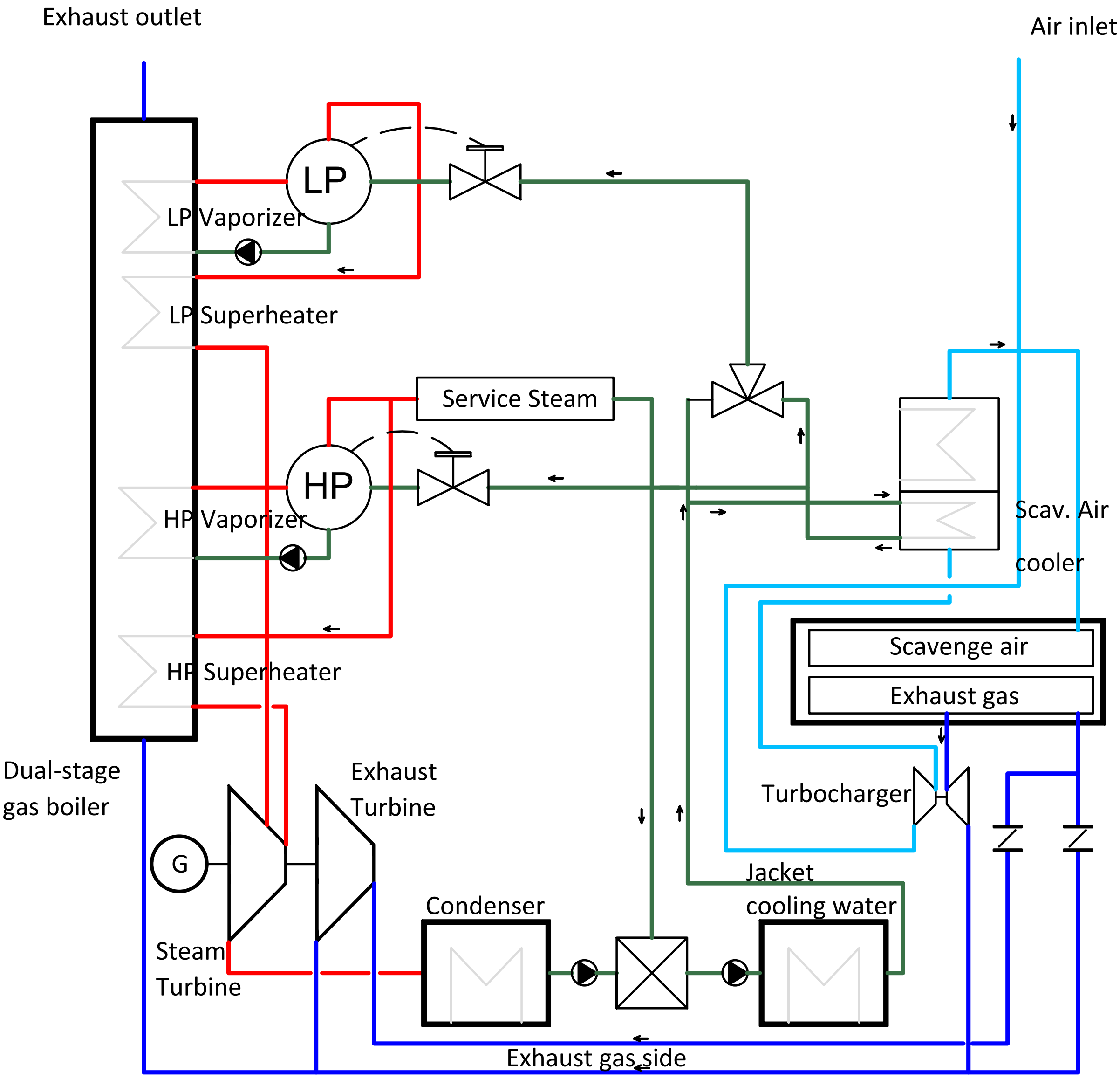

Fig. 2.9 Simplified diagram of the strategy used by MAN, Wärtsilä® and ABB to recover heat.

Fig. 2.9 Shows how the WHRS extract heat from the scavenge air, cooling water and exhaust gas. The exhaust gasses are used to run directly a turbine and to power the gas boiler. Some configurations use only a direct expansion turbine and others only a steam system. However, the most efficient systems always include both as shown in the Fig. 2.9, which stands for a dual-stage gas boiler but a single-stage can be used. Jacket cooling water preheats the water before recovering heat from the scavenge air and then the water is heated in the gas boiler.

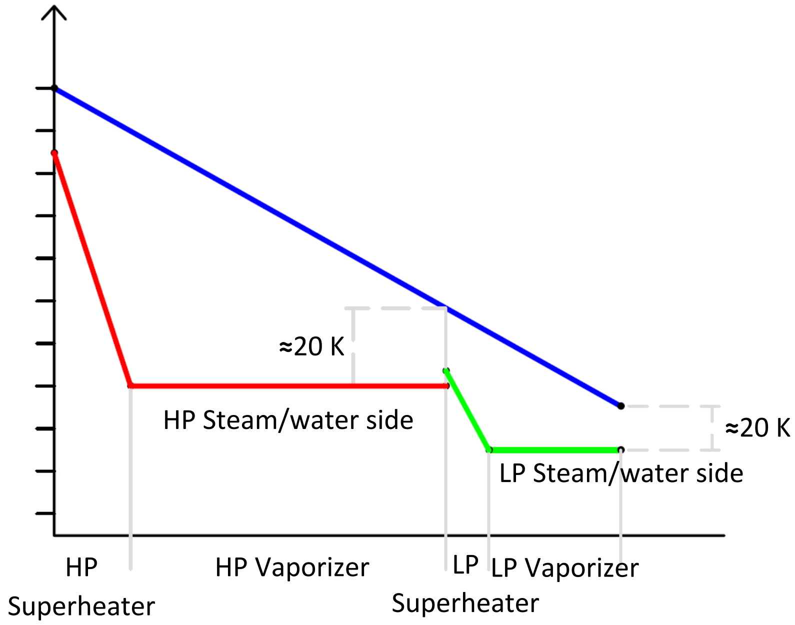

Fig. 2.10 Typical temperatures across the different components of Fig. 2.9.

Fig. 2.10 illustrates the variation of temperature of the water in different components of the WHRS. In the graph we can see a progressive growth of the temperature across the different parts of the boiler, which seems to be rational. In contrast, there is an important part of the energy neglected when the water/steam enters the LP vaporizer and when it leaves the boiler. Energy is unused for feasibility reasons in the condenser, but also when recovering heat from the scavenge air using preheated water from the engine water jacket cooling. In addition, the employment of low-quality materials force to limit the gas exhaust temperature to at least 533 K to avoid being condensated.

2.3.2 MAN

MAN [40] offers technology, which can improve up to 11 % of the main engine power. They are able to obtain this power implementing the approach described in this section, which are direct expansion of exhaust gases and steam expansion involving a dual-pressure gas boiler. MAN® proposes the following WHRS types:

Power turbine and generator (PTG)

As shown in Fig. 2.11, it is a power turbine, which runs on bypassed exhaust gas (between 8 to 12 % bypass). It is the most economical option in terms of installation and space. This option requires at least 40 % load of the main engine. As a result, this simple configuration can recover up to 5 % of the main engine power. (See Tab. 2.4) This option is recommended for Main Engines (ME) with a power output between 15 and 25 MW.

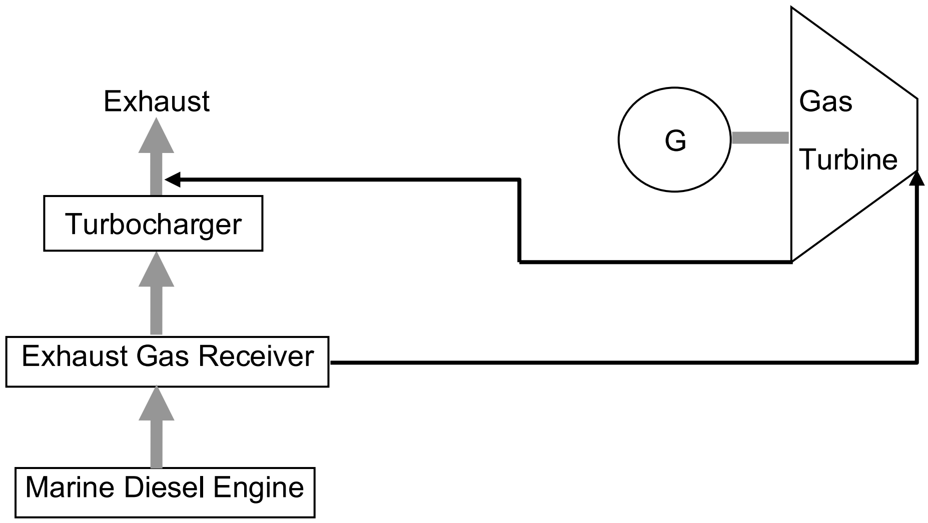

Fig. 2.11 PT WHRS Architecture proposed by MAN.

Steam turbine and generator (STG)

MAN [40] denominates STG the steam system used to convert the exhaust heat into mechanical work expanding steam as shown in Fig. 2.12. It is more complex and occupies more space than the PTG. This option does not bypass exhaust gas. Therefore, the exhaust temperature can be higher. In contrast, this system can recover more than the PTG, around 8 % of the main engine power output. (See Tab. 2.4) This option is recommended for ME with less than 25 MW.

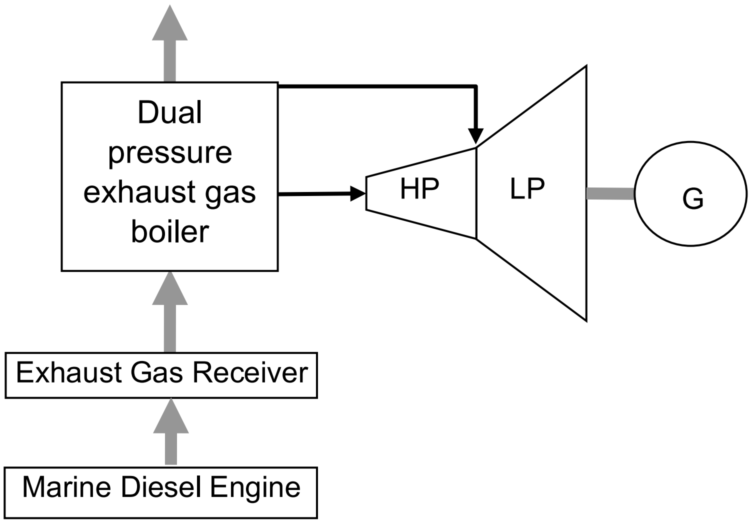

Fig. 2.12 ST WHRS Architecture proposed by MAN.

Steam turbine, power turbine and generator (ST-PT)

The ST-PT systems proposed by MAN® rely on a by-pass of exhaust gas, as shown in Fig. 2.14, part of the exhaust gas is used to expand directly. As a consequence, there is a high temperature exhaust gas that can expand and convert its energy into mechanical work. Then, the residual heat of the bypassed exhaust gas heats water to run a steam turbine, which can be single or dual pressure. In the ST-PT, the steam turbine is coupled to the same generator. This option is recommended in ME with more than 25 MW. ST-PT require less ME load and can be used from 30 % load. Typically, between 8 and 11 % of the ME output can be recovered. (See Tab. 2.4) The application of such system was done in a 57 MW engine. MAN® [41] was able to extract between 4.3 MW and 4.7 MW from WH. In addition, the cash flow analysis showed that the payback time is between three and four years of operation (In terms of auxiliary engine fuel savings). Therefore, MAN® offers an interesting payback ratio. Tab. 2.4 shows MAN® recovery capacity for their WHRS. MAN® WHRS are also able to produce enough electricity to supply the total electricity needed by the power plant.

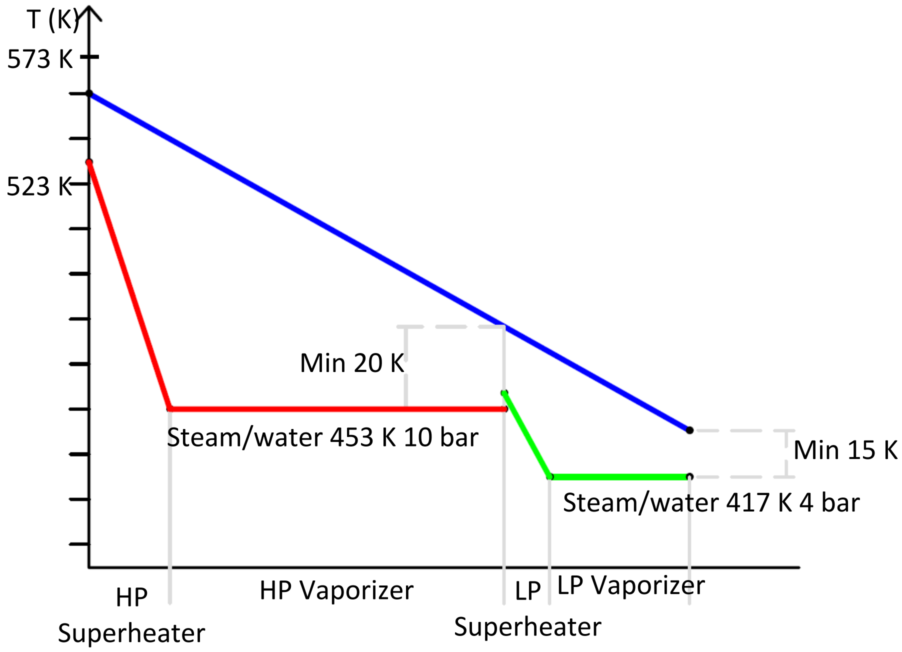

Fig. 2.13 Temperature of the water/steam across different components in a MAN® WHRS [40].

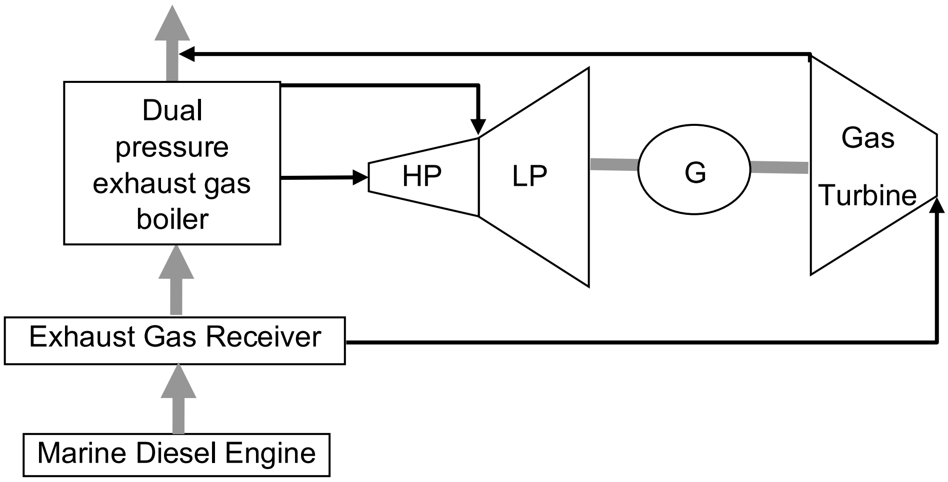

Fig. 2.14 ST-PT Architecture proposed by MAN.

Fig. 2.13 shows typical values of the temperature of the steam/water in different steps of the process. Values are similar to the other two manufacturers reviewed. MAN® also offers the possibility to use a single-pressure system. In this case, the graph would be divided into 3 main sections instead of four: Superheater, Vaporizer and Preheater. The single pressure system is less efficient than the dual pressure.

Tab. 2.4. Maximum efficiency of different MAN® WHRS [40].

WHRS	η_{net} **(%)**
PTG	3-5
STG-Single pressure	4-7
STG-Dual pressure	5-8
Full WHRS (ST-PT)	8-11

2.3.3 Wärtsilä®

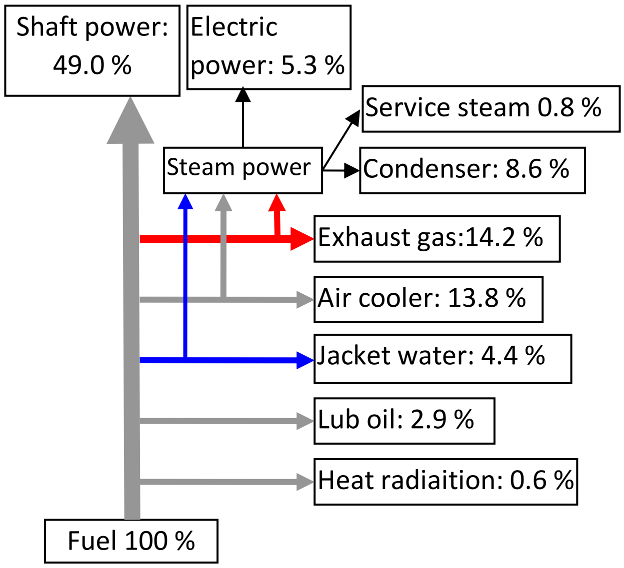

Fig. 2.15 Flow diagram of a heat balance describing the flows of energy occurring in an engine room equipped with Wärtsilä® WHRS [42].

This company recovers heat directly from exhaust gases and generating steam with residual heat to convert into mechanical work in a generator [43]. Their high-efficiency WHRS applied to a RTA96 C improved the energy conversion, the WHRS was able to convert 5.3 % of the total fuel input, as shown in the flow diagram in Fig. 2.15. In this case, exhaust gasses are used in a dual-pressure gas boiler from Aalborg Industries to convert heat form the exhaust to superheated steam, which is expanded in a 6 MWe turbogenerator from Peter Brotherhood. The turbogenerator is equipped with a multi-stage dual-pressure turbine and a turbine running on exhaust gases. The power output is directly supplied to the ship's main switchboard and consumed where required.

The main components are described below and the temperature across the low-pressure and high-pressure superheaters and vaporisers are shown in Fig 2.16.

Gas boiler

This company selected a dual-pressure gas boiler so it has a High and Low-pressure sections. The pressure is typically 9.5 bar and 3.8 bar respectively. The exhaust outlet has to be at least 433 K to preserve the integrity of the materials chosen to construct this system.

Steam-turbine

This component is adapted to the gas boiler. Therefore, it is a dual-pressure steam turbine, which runs at about 6750 rpm and reduced to 1800 rpm when coupled to the generator. Pressures for the high- and low-pressure are slightly lower than the boiler outlet and range between 8 to 9 bar for the high-pressure side and 3 to 3.5 bar for the low-pressure side.

Exhaust-turbine

This turbine needs about 10 % of the exhaust flow from the main engine. Its design is derived from a turbocharger. Therefore, it has about the same expansion ratio and efficiency as turbochargers. Thus, its outlet temperature is similar. It is able to operate when the main engine is at least at 55 % load.

Shaft Motor/Alternator

It requires variable electrical supply frequency at 6600 V and can receive or deliver power. It can be utilized to reduce power consumption, to increase speed or to supply electricity if demanded.

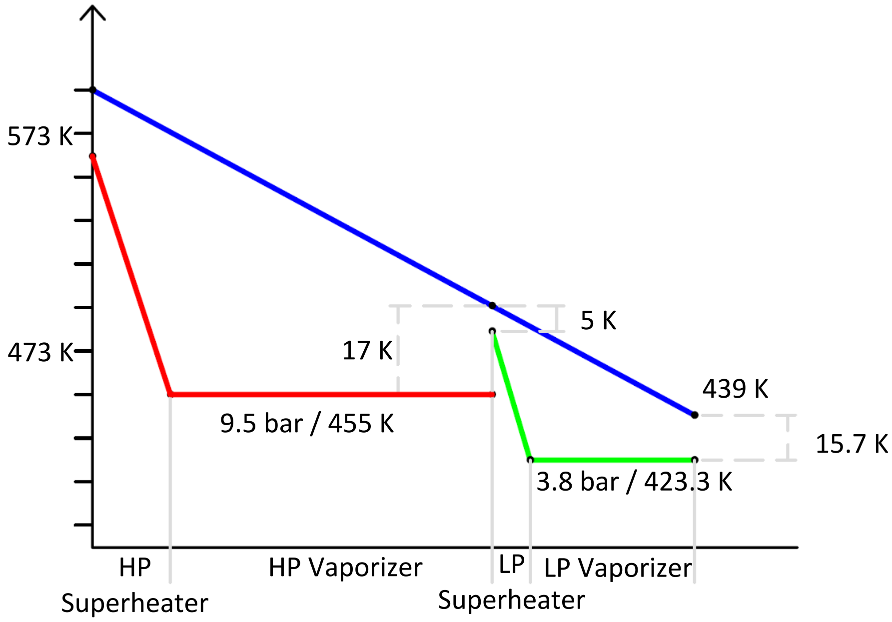

Fig. 2.16 Typical temperatures across the different Wärtsilä® components [43].

2.3.4 Applications of commercial systems

2.3.4.1 MAN® WHRS 12S90ME-C9.2

MAN studied the implementation of a WHRS in a large industrial diesel engine (69.72 MW), as shown in Fig. 2.17 The shaft power output was originally 49.3 %. After implementing a Steam turbine, with dual pressure and a direct expansion of exhaust gasses as shown in Fig. 18, they improved the energy conversion from 49.3 % up to 54.2 %. This amount of energy can represent 36 million US$ for the ship lifetime and 11.2 t of CO_2 per year.

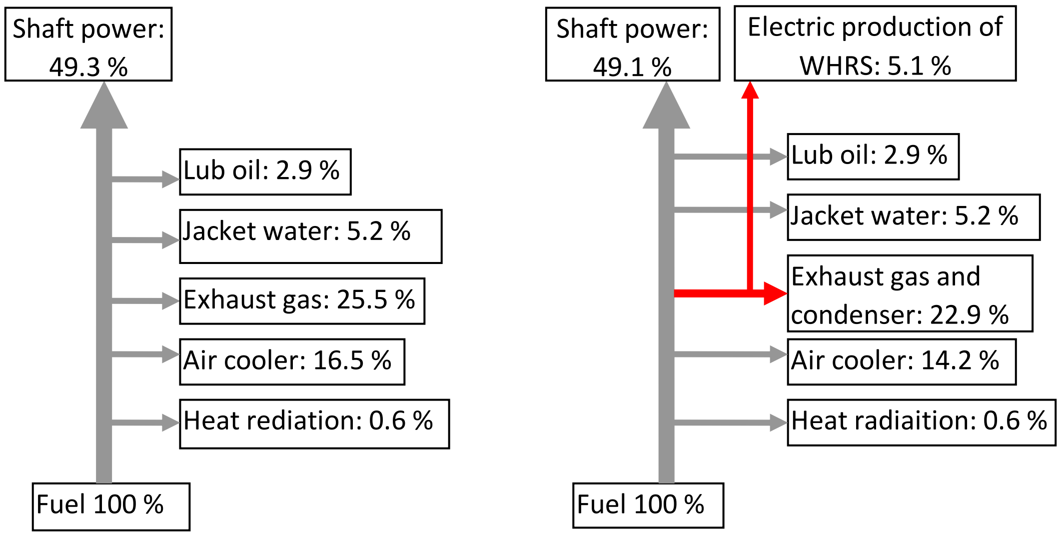

Fig. 2.17 Flow diagram of a heat balance describing the flows of energy occurring in a marine propulsion plant equipped with a dual-pressure MAN® WHRS coupled to a MAN® 12S90ME-C9.2 [40].

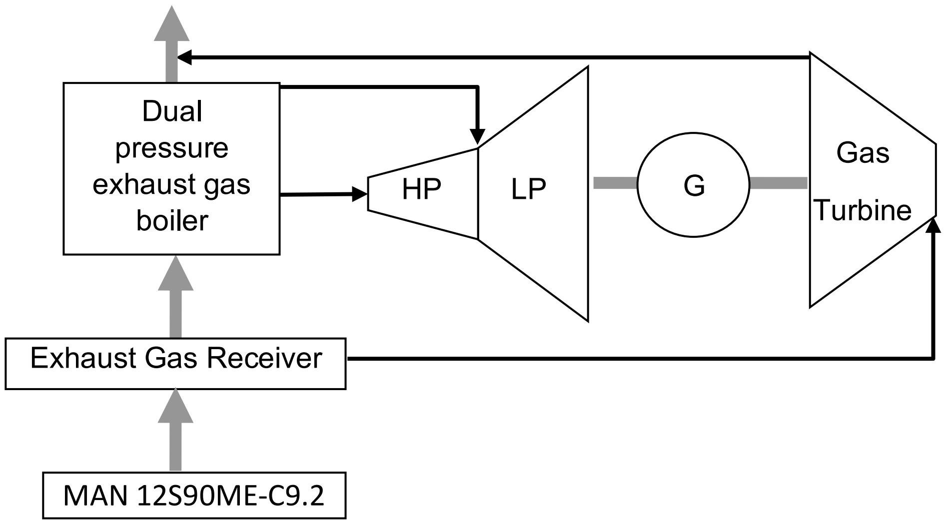

Fig. 2.18 MAN® 12S90ME-C9.2 equipped with a WHRS.

As previously mentioned, the WHRS takes heat from exhaust gases to power a Rankine system, which has a dual pressure turbine. It also takes advantage of the simplest WHRS available technique, which is the direct expansion of exhaust gases from the Diesel engine (See Fig. 2.18). Following the description of the manufacturer, in this particular case jacket water and scavenge air heat is not used. They are probably required in other processes such as service steam or heating Thermal Fluid (TF) for other purposes. Another possibility is that the additional heat from the jacket water and scavenge air does not make an efficient system in terms of space occupied in the engine room and cost.

2.3.4.2 Wärtsilä® 12RT-flex96 C

Wärtsilä® [43] implemented a WHRS in a modern low-speed engine with 49.3 % of efficiency. Fig. 2.19 illustrates the energy flow when coupling a WHRS. Even with a small reduction in performance in the main engine itself, they achieved a 54.9 % efficiency implementing their WHRS as shown in Fig. 2.20.

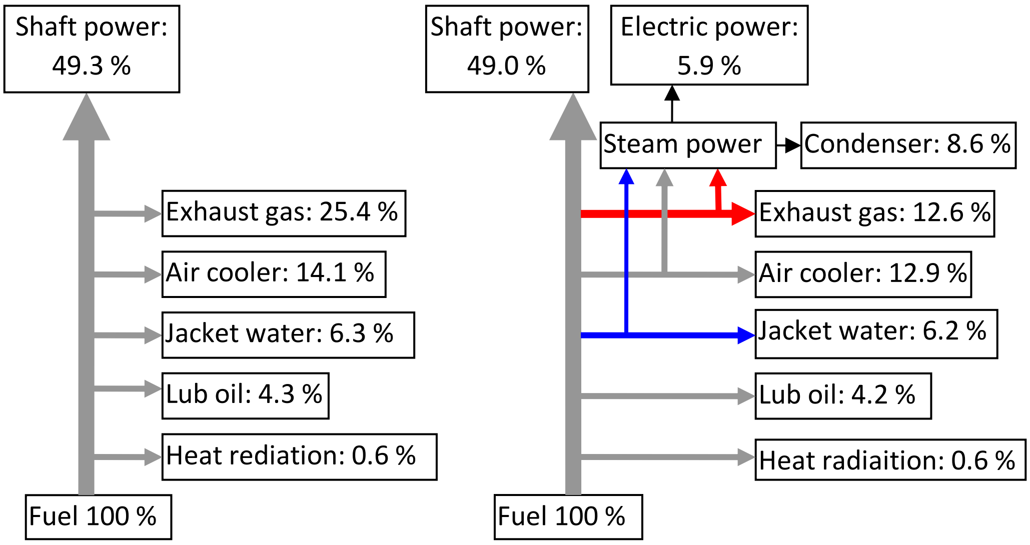

Fig. 2.19 Flow diagram of a heat balance describing the flows of energy occurring in an engine room equipped with Wärtsilä® WHRS [43].

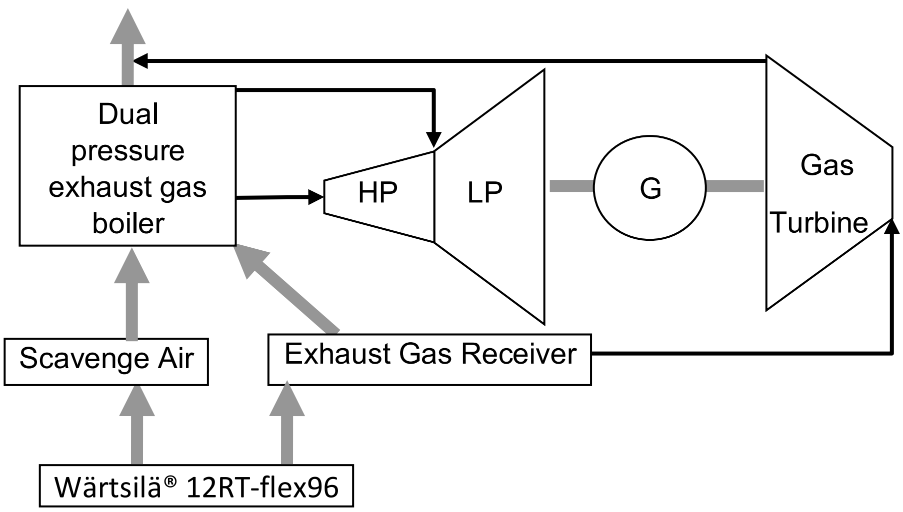

Fig. 2.20 Wärtsilä® 12RT-flex96 C equipped with a WHRS.

This WHRS architecture uses jacket water scavenge air and exhaust gases to power the WHRS. Jacket water and scavenge air is used to preheat the water of the RC mainly powered by the exhaust gasses, which after the turbo compressors powers the dual stage gas boiler. The heat from the jacket water and scavenge air preheats the low-pressure side of the boiler as shown in Fig. 2.10. The preheat process cannot extract efficiently heat from the scavenge air because water is already preheated close to the jacket water temperature before entering the heat exchanger to extract heat from the scavenge air.

2.3.5 State of the art

The two manufacturers offer results in the same range of performance. This is intuitive because they both share the same strategy to recover heat. Therefore, differences will be dictated by the performance of the components. The data provided is always given as the percentage of main engine power output. Therefore, the efficiencies by themselves are function of the power output, which is not consistent for a direct comparison because the power output of the engines suggested by the manufacturers are not the same.

Tab. 2.5. Comparison of the reviewed marine WHRS.

System	Description	η_{Th} (%)
Wärtsilä® [43]	Steam turbine and direct expansion of exhaust gases	12
MAN [40]	Steam turbine and direct expansion of exhaust gases (STG-PTG)	12

Efficiencies of the current technology can get better This chapter is dedicated to the poor capacity of WHRS to convert energy. In addition, these systems have almost no capacity to convert energy when there is a small difference temperature. This is due to the fact that the reviewed thermodynamic cycles and machines are based on Stirling-Rankine-Ericsson cycles and are constrained by its limitations. These cycles have important constraints, such as, having open-process, doing phase changes or experiencing thermodynamic transformations at constant pressure.

2.4 Definition of heat utilisation factor

To assess the ability to convert thermal energy into mechanical work it is necessary to define a parameter, which takes into account this fact. Conventionally the 1st law efficiency is used to quantify or compare the work output according to a heat input. It is defined by the useful work divided by the fuel-based heat energy. In contrast, when non-conventional energy conversion systems are used, it is more suitable to define the relation of the total work output to the heat input.

Tab. 2.6. Nomenclature of Fig. 2.21.

System	HUF (%)
Fossil fuel-based heat power supply	Q_S
Net work delivered by the propulsion thermal engines	W_N
Residual heat	Q_{RES}
Recovered heat as work/electric power/heat services	W_R
Unrecovered heat	Q_{UN}

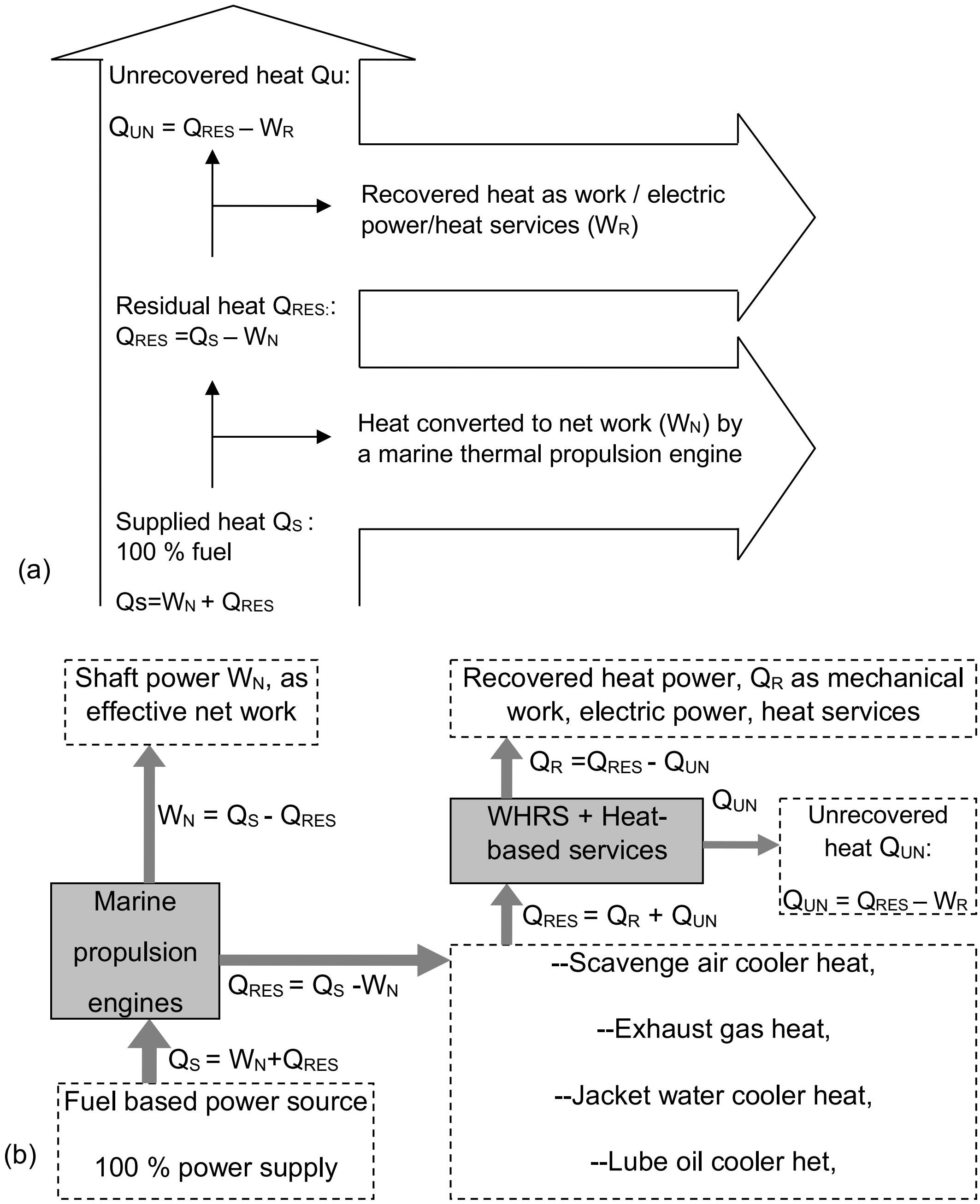

Fig. 2.21 Heat flow diagram for a conventional marine power plant.

Heat can be recovered by direct utilisation in the form of rejected heat and / or by turning the rejected heat into useful work. Fig. 2.21 shows that an important amount of energy is not or cannot be used to convert into mechanical work. Tab 2.6 shows the nomenclature. The objective of the Heat utilisation factor (HUF) is to quantify all the work output and compare it to the heat input. As a result, Recovery factor or heat utilisation factor is:

- The ratio of the utilized heat to the supplied heat.
- The ratio of the net useful amount of heat and work to the supplied heat.
- The ratio of the net work to the supplied heat. (When no available heat is utilised)

Thermal efficiency of a thermal plant (conventional thermodynamics):

$$\eta_{Th} = \frac{Q_S - Q_{RES}}{Q_S} = 1 - \frac{Q_{RES}}{Q_S} = \frac{W_N}{Q_S} \qquad \text{Eq. 2.1}$$

Residual heat:

$$Q_{RES} = Q_S - W_N \qquad \text{Eq. 2.2}$$

Thermal efficiency of a heat recovery thermal plant (conventional thermodynamics):

$$\eta_R = \frac{Q_{RES} - Q_{UN}}{Q_{RES}} = 1 - \frac{Q_{UN}}{Q_{RES}} = \frac{W_R}{Q_{RES}} \qquad \text{Eq. 2.3}$$

Recovered heat in mechanical work mode:

$$W_R = Q_{RES} - Q_{UN} = (Q_S - W_N) - Q_U \qquad \text{Eq. 2.4}$$

Rejected heat:

$$Q_{UN} = Q_{RES} - W_R \qquad \text{Eq. 2.5}$$

Then HUF:

$$HUF = \frac{W_N + W_R}{Q_S} \qquad \text{Eq. 2.6}$$

The following Tab. 2.7 collects information that reflects the heat utilisation factor of the former examples seen in Section 2.3.4:

Tab. 2.7. Comparison of the reviewed marine heat recover systems.

System	HUF (%)
Wärtsilä® 12RT-flex96 C [42]	54.9
MAN WHRS 12S90ME-C9.2 [40]	54.2

The HUF of the systems mentioned in Tab. 2.8 describe the large amount of heat wasted by modern marine power plants. The HUF reflects the work conversion performed for a specific supplied heat. In these cases, both systems waste more than 45 % of their energy input. Thus, there is clearly potential for improvement in the WH recovery field.

Chapter 3 introduces a new concept that overcomes some of the limitations of the cycles reviewed in this chapter.

3 Proposed solutions

This chapter deals with the architecture of the proposed thermal cycle that overcomes limitations of the reviewed cycles. These cycles operate with closed processes-based transformations experimentally validated by a successful proof of concept implemented under a double acting reciprocating cylinder, equipped with heat source and heat sink and its heat transfer exchangers (See Section 3.6). In order to accurately study a thermal cycle characterized by doing useful mechanical work by means of the addition and extraction of heat, the fundamental principles) are first revised for such a particular cycle structure. Furthermore, aiming at verifying compliance of the second law, an entropic analysis for the specific given cycle is performed.

It is assumed that the performance limit of a thermal cycle is given only by Clausius and Kelvin–Planck statements, summarised as: "*The performance of any irreversible thermal cycle must be less than 100 %".* As reviewed in the second section, current technology is well-below the 100 % efficiency. This is the motivation to create a new concept that can outperform existing energy conversion technology obeying the thermodynamic laws and the Clausius and Kelvin-Planck statement.

Research methodology was conceived considering the research objectives, summarised as:

- Energy balance analysis
- Entropy analysis
- Empirical validation

The research objectives have the main purpose of investigating at least a thermal cycle characterised by:

- Doing work through the extraction of heat instead of rejecting heat to a heat sink.
- Operating under closed thermodynamic processes.
- Having an engine structure exhibiting the ability to convert a linear reciprocating motion into continuous rotating motion.

The most appealing feature of the proposed Closed-based 4 Process (C4L) machine is its capability to efficiently exploit heat with low temperature, which can be geothermal, ocean thermal, residual or industry, marine or solar. the so-called C4L is convenient when it is necessary to convert low grade heat and hence, can outperform the state-of-the-art conversion systems

3.1 Analysis of the first law for heat-work interactions in power cycles

With regards to the conventional energy conversion techniques for reversible processes, the principle of conservation of energy requires that: "*the total amount of energy before and after a transformation in both, the heat source and the heat sink are equal*". Therefore, focusing the problem definition task on the efficient conversion of heat energy to useful work, the amount of energy (in the form of heat and mechanical work, included in the heat sink and the heat source) before and after a process, transformation or cycles is constant. Apparently, the energy lost by the heat source is gained by the heat sink in any known form (useful work and rejected het). This fact is validated experimentally in a wide range of empirical processes by means of energy balances. However, although the final energy equals to the initial energy, it is impossible to recover the initial energy by using both, the mechanical work and the rejected heat. The ability to do work of the final energy (work and rejected heat) is inferior than the initial energy. Rejected heat has been converted into low-grade heat, so that it has been degraded undergoing entropy increasing. However, if only conservative magnitudes (such as kinetic or gravitational potential energies) energy is not degraded during any transformation, so that only in this case energy is conserved.

Likewise, under neglected kinetic and potential energies, the amount of energy (in the form of heat and mechanical work) of the heat sink and heat source before and after a cyclic transformation for an isolated system composed by the heat source, the heat sink and the thermal cycle, is constant.

Fig. 3.1(a) depicts the flow graph of the manipulated energy into a thermal cycle. Thus, the first law fulfilment for a cycle involving heat-work interactions undergoing non-conservative magnitudes can be expressed as:

$$\sum E_i(q,w) + \sum E_o(q,w) = \Delta U = 0 \qquad \text{Eq. 3.1}$$

In terms of closed processes-based heat-work interactions, this means that:

$$\sum q_i + \sum w_i + \sum q_o + \sum w_o = \Delta U = 0 \qquad \text{Eq. 3.2}$$

Which in a compact form can be expressed as:

$$\sum q + \sum w = 0 \qquad \text{Eq. 3.3}$$

Equation 3.3 means that for a thermodynamic cycle the principle of conservation of energy fulfils de facto. Furthermore, apparently, this principle cannot be violated in a single thermal cycle because the rejected energy is not accounted for successive heat work interactions, since the amount of energy at the initial state and final state of thermal cycle is constant.

The energy balance taking into account an isolated system where the heat source, heat sink, and the thermal cycle are included into the isolated system, is described as follows, where the cycle surroundings are included into the isolated system. The scheme depicted in Fig. 3.1(b) shows the isolated system with the energy balance equations:

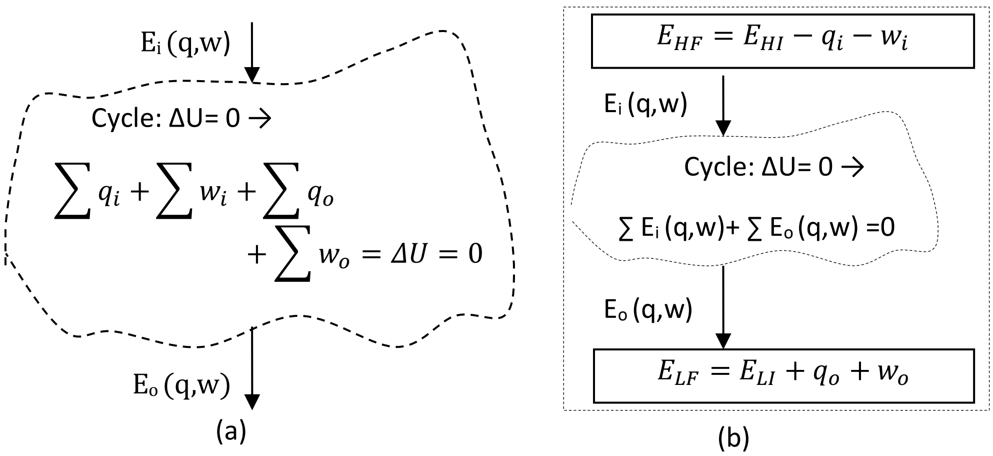

Fig. 3.1. The layout of an energy conversion system. (a), system undergoing closed processes based thermal cycle. (b) isolated system undergoing a closed process based thermal cycle.

Thus, considering a system composed of a heat source, a heat sink and a thermal cyclic such as a thermal cycle under analysis, the energy balance must be used to verify compliance with causality law, the first and second laws of the thermodynamics, where what is taken into account is that the total amount of energy (q,w) into the heat source and the heat sink before and after a cycle is constant. Therefore, the energy balance before and after a cyclic transformation is expressed as:

"The total amount of energy in the heat source and the heat sink before a transformation or a cycle ($E_{HI} + E_{LI}$) must be equal to the total energy of the heat source and the heat sink after the transformation or cycle ($E_{HF} + E_{LF}$)". That is expressed as:

$$E_{HF} + E_{LF} = E_{HI} + E_{LI} \qquad \text{Eq. 3.4}$$

Proof:

According to the scheme depicted with Fig. 3.1(b) follows that:

$$E_{HF} = E_{HI} - q_i - w_i$$

$$E_{LF} = E_{LI} + q_o + w_o \qquad \text{Eq. 3.5}$$

Eq. 3.6 is solved by using Eq. 3.5 into Eq. 3.4, as follows:

$$E_{HI} - q_i - w_i + E_{LI} + q_o + w_o = E_{HI} + E_{LI} \qquad \text{Eq. 3.6}$$

Simplifying Eq. 3 .6, yields:

$$q_o + w_o = q_i + w_i$$

or

$$q_i - q_o = w_o - w_i \qquad \text{Eq. 3.7}$$

which implies that: $\sum q + \sum w = 0$, as expresses Equation 3.3

However, bearing in mind that the rejected energy from any transformation or cycle to the heat sink at ambient temperature (temperature of its surroundings), in the form of low-grade heat and increased entropy, some rejected energy is vanished or annihilated since it cannot be recovered in any way, (the system has lost part of its ability to do useful work). Therefore, such controversial aspect needs a deeper analysis to lead to a consistent conclusion.

That is, into a thermal cycle apparently, energy is conserved. However, in the outside the thermal cycle energy is not conserved since finally it is annihilated or vanished in the form of electromagnetic radiation (low-grade heat).

With the purpose of concluding the above comments about the principle of conservation of the energy so far one can assert that: $\sum q + \sum w = 0$, then:

$$E_{HF} + E_{LF} = E_{HI} + E_{LI} \qquad \text{Eq. 3.8}$$

As stated by the principle of the conservation of energy, according to Equation 3.7 for conventional thermal cycles:

$$q_i - q_o = w_o - w_i = w_n \qquad \text{Eq. 3.9}$$

In all the conventional thermal cycles known by now, the transformation of thermal energy to mechanical work involving a thermal cycle where useful work is done does not alter the overall balance of the input-output energy. However, summing the useful work to the rejected heat so as to obtain the initial input energy (the initial ability to do work), it is observed that it is not possible to recover the input spent energy due to the existence of non-conservative magnitudes (low-grade heat with increased entropy).

Based on experimental evidence the following lemma can be stated:

Lemma 1

If the energy is degraded through thermodynamic transformations involving heat transfer and / or non-conservative magnitudes, then, energy is destroyed being unrecoverable and unavailable.

Lemma 2

The energy conversion task based in transferring heat from a high temperature heat source to a low temperature heat sink, after several cascade conversion cycles the residual heat lost the ability to do useful work, or cannot be completely converted into useful work so that an important amount of low-grade heat is wasted in the heat sink.

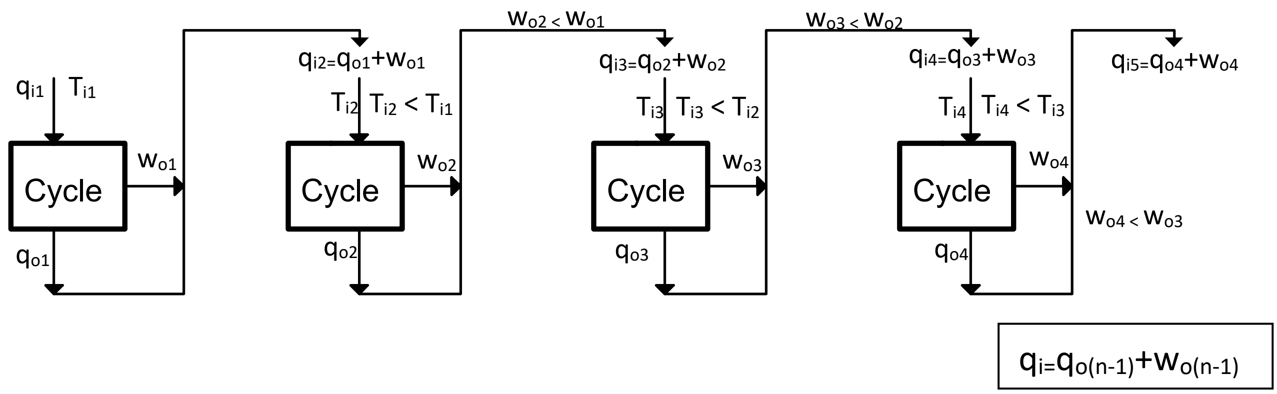

Fig. 3.2. Scheme of the flow graph depicting the heat energy degradation.

The flow graph of energy degradation undergoing heat-work interaction modes, which is based on non-conservative magnitudes, is depicted with Fig. 3.2. As shown, every successive cycle operates with the same amount of energy as input heat but at lower temperature so that the amount of useful work delivered will be also lower. Therefore, the ability to do work is being lost as energy passes from a given process cycle to the next one.

To say, after several processing phases the heat energy transferred to the heat sink exhibits so low temperature (low-grade heat), that finally cannot be converted into useful work.

Consequently, it is observed that energy is not conserved, because of the non-conservative characteristics of involved magnitudes (heat).

The conclusion expressed by the energy balance Equation 3.9 asserts compliance with the law of conservation of energy in the case of a conventional thermal engine for converting heat energy to useful work only for conservative magnitudes (heat is excluded) in which a fraction of the heat absorbed from a heat source is converted into mechanical work, while the remaining heat fraction is rejected towards the heat sink.

3.2 Axiomatic facts related to the evolution of the entropy of an isolated system

Entropy balance is a common conventional procedure to verify the fulfilment of the second law in any thermodynamic transformation in which heat transfer exists as well as in thermal analysis of the cycle. Thus, in what follows the method to verify the cycles to be analysed is described. In an isolated system composed by a heat source A emitting energy (transferring heat) to a heat sink B assumed as the surroundings at its local temperature, it is observed that the entropy of the isolated system is increasing if such isolated system is composed by condensed matter (instead of radiated energy trough the electromagnetic field towards a deep space). The assertion is related to the fact that the heat sink B considered as the surroundings whose condensed matter is at local temperature, absorbs the heat transferred from A to B, so that the final entropy increases while it can never be lower than zero in a real system

Fig. 3.3 shows the scheme of an isolated system in which a heat source A, transfer heat to a heat sink B, according to Clausius statement, absorbs the heat rejected by the A.

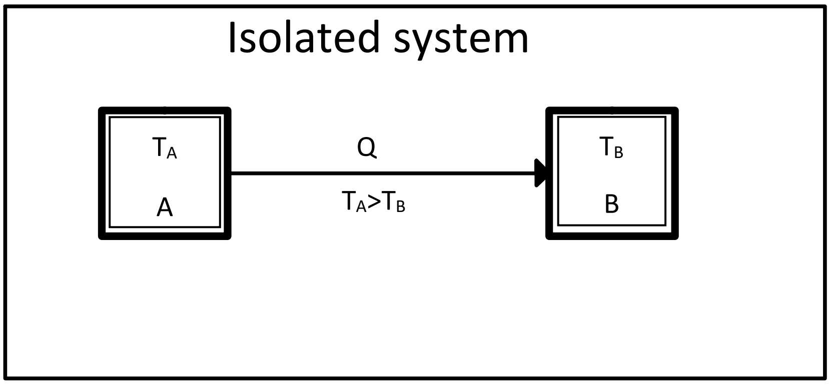

Fig. 3.3. Scheme of an isolated system for the analysis of entropy.

The change in entropy of the heat source A:

$$\Delta S_A = S_{A2} - S_{A1} = \int_1^2 \frac{\delta Q}{T} = \frac{1}{T_A}\int_1^2 \delta Q = \frac{-1}{T_A} Q \quad \text{Eq. 3.10}$$

The change in entropy of the heat sink B:

$$\Delta S_B = S_{B2} - S_{B1} = \int_1^2 \frac{\delta Q}{T} = \frac{1}{T_B}\int_1^2 \delta Q = \frac{1}{T_B} Q \qquad \text{Eq. 3.11}$$

The total change in entropy of the isolated system ΔS_U composed by a heat source A, and a heat sink B, is then:

Initial entropy,

$$\Delta S_{U1} = S_{A1} + S_{B1} \qquad \text{Eq. 3.12}$$

Final entropy,

$$\Delta S_{U2} = S_{A2} + S_{B2} \qquad \text{Eq. 3.13}$$

The change in entropy is the difference between final and initial total (entropy of the heat source and entropy of the heat sink) entropies ΔS_U. Then:

$$\Delta S_U = \Delta S_{U2} - \Delta S_{U1} =$$

$$(S_{A2} + S_{B2}) - (S_{A1} + S_{B1}) =$$

$$(S_{A2} - S_{A1}) + (S_{B2} - S_{B1}) = \Delta S_A + \Delta S_B =$$

$$Q\frac{-1}{T_A} + Q\frac{1}{T_B} = Q(\frac{1}{T_B} - \frac{1}{T_A}) = Q\frac{T_A - T_B}{T_A \cdot T_B} \qquad \text{Eq. 3.14}$$

From the above result follows that:

$$T_A \geq T_B \rightarrow \Delta S_U = S_{U2} - S_{U1} \geq 0 \qquad \text{Eq. 3.15}$$

Which means that the change in entropy of the isolated system is greater or equal than zero, provided that the heat sink is able to absorb the heat rejected by the system by means of condensed matter at local temperature. This involves that the rejection of energy in the form of electromagnetic radiation (heat included) to the deep space, where no condensed matter exists, destroy the ability to do work (so, destroy energy) without increasing entropy due to the lack of condensed matter and losing its ability to be converted into work. The evidence of this assertion obeys daily experimental validation.

The increasing entropy principle is a consequence of the energy degradation due to transformations involving non-conservative magnitudes (all heat-work interactions are non-conservative transformations), although the amount of energy at the initial and final of any transformation is constant. (energy balance, energy equivalence and energy conservation obey different meanings)

3.3 Searching alternative solutions to the conventional techniques

In order to find a thermal cycle with the ability to provide better performance in terms of thermal efficiency the following question arises: is it possible the existence of another type of thermal cycle (a thermal cycle that performs useful mechanical work by extracting heat from the engine to a heat sink) capable of performing useful mechanical work by using a different approach of the energy balance in a thermal cycle by means of another strategy of heat utilisation.

Aiming at a thermal cycle with the ability to render more thermal efficiency than the conventional, for finding an answer to the above question by a rigorous and reasonable analysis, let's observe a Lenoir thermal cycle [44], which is characterized by three processes as shown in Fig. 3.4 and Tab. 3.1, so that it is a trilateral cycle:

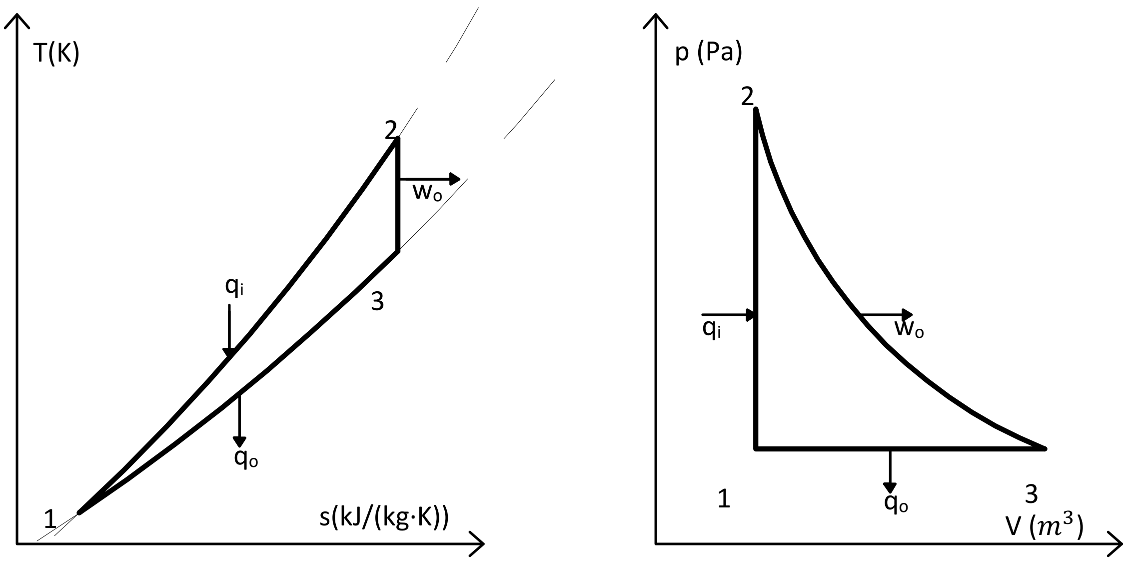

Fig. 3.4. The basic Lenoir thermal cycle. (a), p-V diagram. (b), T-s diagram.

Tab. 3.1 The Lenoir thermal cycle, showing the transformations carried out for every path function.

Cycle process	Thermal process and path function	Function model
1-2	Closed isochoric heat addition, q_i	$q_i = Cv \cdot (T_2 - T_1)$
2-3	Closed isentropic expansion, w_o	$w_o = Cv \cdot (T_2 - T_3)$
3-1	Open isobaric heat rejection, q_o	$q_o = Cp \cdot (T_3 - T_1)$

Energy balance of Lenoir thermal cycle

The flow graph of the energy balance for the Lenoir cycle is depicted with Fig. 3.5. There can be observed that the rejected heat obeys an open process-based transformation, in which flow work is involved so that a relevant degradation of the rejected heat energy is implicit to such cycle.

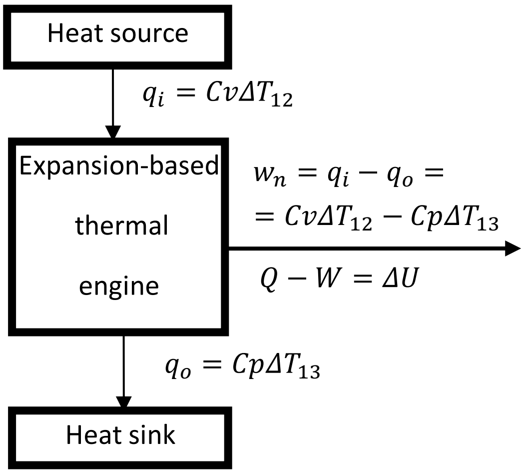

Fig. 3.5. Energy flow of the Lenoir thermal cycle.

According to the processes carried out in Tab. 3.1,it follows that the thermal efficiency $\eta_L$of the Lenoir cycle is given as:

$$\eta_L = 1 - \frac{q_o}{q_i} = 1 - \frac{Cp \cdot (T_3 - T_1)}{Cv \cdot (T_2 - T_1)} = 1 - \gamma \frac{(T_3 - T_1)}{(T_2 - T_1)} \qquad \text{Eq. 3.16}$$

It is observed that the analysed thermal cycle rejects the inherent heat to the heat sink, (environment) by means of an open process consisting in some flow work and internal energy rejected to the heat sink, specified as:

$$q_o = Cp \cdot (T_3 - T_1) = \Delta h = h_3 - h_1 \qquad \text{Eq. 3.17}$$

So as to overcome the drawbacks encountered with this thermal cycle, it’s necessary to start the search task by using another form of heat-work interaction mode, in which it is feasible to:

1. Rejecting heat in the form of internal energy to the sink without flow work.

2. Doing mechanical work by extracting heat from the WF by means of closed processes-based heat-work interactions, instead of refusing heat to the heat sink as in conventional thermal cycles (open processes transformations).

To explore the possibility of such cycle, we can take a scientific approach and list all the polytropic, isochoric, isothermal and isobaric closed interactions. Tab. 3.2 lists the 32 possible closed thermodynamic interactions [45] and a first-law balance is done for each interaction, according to the pressure of the system (p_{sy}) and the surroundings (p_{su}). RE represent the possibility of delivering work using heat (Y) or the impossibility to convert heat into work (N).

Firstly, the proposed mode of heat-work interaction differs from the conventional Lenoir cycle in that a thermal cycle implemented undergoing closed processes-based heat-work interaction modes has not the loses inherent to the flow work encountered in all open processes. The next section deals with a C3L based on Lenoir cycle, which overcomes this limitation.

Tab. 3.2 Polytropic, isochoric, isothermal and isobaric transformations for closed-processes based systems [45].

Transfer mode	ΔV	p_{sy} versus p_{su}	balance 1[st] law	RE	RON
		Polytropic processes			
	$\Delta V > 0$	$p_{sy} > p_{su}$	$\Delta u = q - w_{o\_exp}$	Y	1
	expansion	$p_{sy} < p_{su}$	$\Delta u = q + w_{i\_cont}$	Y	2
q > 0	$\Delta V < 0$	$p_{sy} > p_{su}$	$\Delta u = q + w_{i\_comp}$	Y	3
heating	compression	$p_{sy} < p_{su}$	$\Delta u = q - w_{o\_cont}$	Y	4
	$\Delta V > 0$	$p_{sy} > p_{su}$	$\Delta u = -w_{o\_exp}$	Y	5
	expansion	$p_{sy} < p_{su}$	$\Delta u = w_{i\_cont}$	Y	6
q = 0	$\Delta V < 0$	$p_{sy} > p_{su}$	$\Delta u = w_{i\_comp}$	Y	7
adiabatic: $n=\gamma$	compression	$p_{sy} < p_{su}$	$\Delta u = -w_{o\_cont}$	Y	8
	$\Delta V > 0$	$p_{sy} > p_{su}$	$\Delta u = q - w_{o\_exp}$	Y	9
	expansion	$p_{sy} < p_{su}$	$\Delta u = q + w_{i\_cont}$	Y	10
q < 0	$\Delta V < 0$	$p_{sy} > p_{su}$	$\Delta u = q + w_{i\_comp}$	Y	11
cooling	compression	$p_{sy} < p_{su}$	$\Delta u = q - w_{o\_cont}$	Y	12
		Isochoric processes			
q > 0	$\Delta V = 0$	$p_{sy} > p_{su}$	$\Delta u = q$	Y	13
heating	isochoric	$p_{sy} < p_{su}$	$\Delta u = q$	Y	14
q < 0	$\Delta V = 0$	$p_{sy} > p_{su}$	$\Delta u = q$	Y	15
cooling	isochoric	$p_{sy} < p_{su}$	$\Delta u = q$	Y	16
		Isothermal processes			
	$\Delta V > 0$	$p_{sy} > p_{su}$	$q - w_{o\_exp} = 0$	v	17
	expansion	$p_{sy} < p_{su}$	$q + w_{i\_cont} = 0$	Y	18
q > 0	$\Delta V < 0$	$p_{sy} > p_{su}$	Not applicable	N	19
heating	compression	$p_{sy} < p_{su}$	Not applicable	N	20
	$\Delta V > 0$	$p_{sy} > p_{su}$	Not applicable	N	21
	expansion	$p_{sy} < p_{su}$	Not applicable	N	22
q < 0	$\Delta V < 0$	$p_{sy} > p_{su}$	$q + w_{i\_comp} = 0$	Y	23
cooling	compression	$p_{sy} < p_{su}$	$q - w_{o\_cont} = 0$	Y	24
		Isobaric processes			
	$\Delta V > 0$	$p_{sy} > p_{su}$	$\Delta u = q - w_{o\_exp}$	Y	25
	expansion	$p_{sy} < p_{su}$	$\Delta u = q + w_{i\_cont}$	Y	26
q > 0	$\Delta V < 0$	$p_{sy} > p_{su}$	Not applicable	N	27
heating	compression	$p_{sy} < p_{su}$	Not applicable	N	28
	$\Delta V > 0$	$p_{sy} > p_{su}$	Not applicable	N	29
	expansion	$p_{sy} < p_{su}$	Not applicable	N	30
q < 0	$\Delta V < 0$	$p_{sy} > p_{su}$	$\Delta u = q + w_{i\_comp}$	Y	31
cooling	compression	$p_{sy} < p_{su}$	$\Delta u = q - w_{o\_cont}$	Y	32

3.3.1 The closed processes expansion based thermal cycle

Different configurations of closed-processes based, non-condensing C3L were studied [46–48]. Isobaric, isothermal and adiabatic expansion were considered. Authors found that the specific work output is largely higher for isothermal expansion but, for feasibility reasons, adiabatic expansion is chosen. Ferreiro et al. [49] proposed a variation of the traditional Otto cycle, where they explored the potential for closed adiabatic expansions. They found that for a wide range of temperatures this cycle can even obtain an efficiency higher than the Carnot efficiency. This fact confirms the relevance and advantage of this type of expansion. The thermal cycle shown in Fig. 3.6 represents the T-s diagram of a physically realisable, reversible, non-condensing, closed-processes based thermal cycle, which is completed by means of three closed process based transformations: closed isochoric heating, absorbing heat from an external heat source, adiabatic quasi-isentropic expansion, doing useful work, and quasistatic isobaric cooling extracting heat to a heart sink at lower temperature. Thus, the cycle path functions are modelled as:

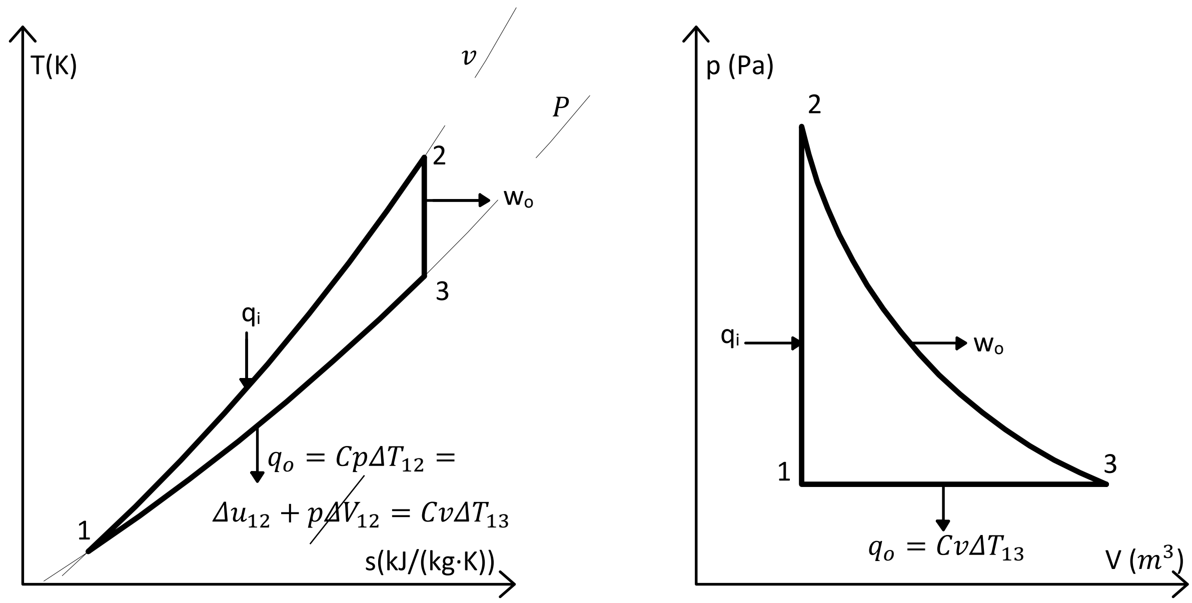

Fig. 3.6. The expansion based thermal cycle. (a), p-V diagram. (b), T-s diagram.

Cooling at constant pressure during a quasistatic isobaric contraction without useful work is used to contract the WF (flow work) and to decrease the internal energy according to Eq. 3.18:

$$q_o = q_{13} = \Delta h_{13} = Cp\Delta T_{13} = \Delta u_{13} + p\Delta V_{13} = Cv\Delta T_{13}$$ Eq. 3.18

Tab. 3.3 The closed processes-based expansion based thermal cycle, showing the transformations carried out for every path function.

Cycle process	Thermal process and path function	Function model
1-2	Closed isochoric heat addition, q_i	$q_i = Cv \cdot (T_2 - T_1)$
2-3	Closed isentropic expansion, w_o	$w_o = Cv \cdot (T_2 - T_3)$
3-1	Closed quasistatic isobaric heat rejection, q_o	$q_o = Cv \cdot (T_3 - T_1)$

Tab. 3.3 depicts the C3L expansion based thermal cycle, showing the transformations carried out for every path function. Observing the cycle process 3 it can be noted that the process of extracting heat to the heat sink is carried out by means of a closed transformation so that flow work is not involved, and some loses are avoided. According to the Tab. 3.3, follows that the thermal efficiency η of the expansion based thermal cycle is given as:

$$\eta_{Th_{C3L}} = 1 - \frac{q_o}{q_i} = 1 - \frac{Cv \cdot (T_3 - T_1)}{Cv \cdot (T_2 - T_1)} = 1 - \frac{(T_3 - T_1)}{(T_2 - T_1)}$$ Eq. 3.19

3.3.1.1 Energy flow of the expansion based thermal cycle

The closed processes expansion based thermal cycle energy balance is depicted in Fig. 3.7, which reflects the principal difference compared to Lenoir Cycle: The heat output at constant volume instead of constant pressure.

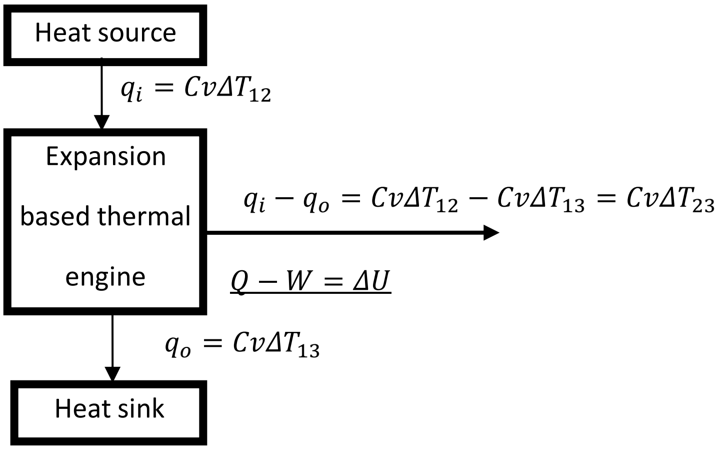

Fig. 3.7 Energy flow of the expansion based thermal cycle.

Comparing the thermal efficiencies of Lenoir cycle to that of the closed processes expansion based thermal cycle follows that:

$$\eta_L = 1 - \gamma \frac{(T_3 - T_1)}{(T_2 - T_1)} \quad \text{Eq. 3.20}$$

$$\eta = 1 - \frac{(T_3 - T_1)}{(T_2 - T_1)} \quad \text{Eq. 3.21}$$

Thus, since:

$$\gamma \frac{(T_3 - T_1)}{(T_2 - T_1)} > \frac{(T_3 - T_1)}{(T_2 - T_1)} \quad \text{Eq. 3.22}$$

Implies that:

$$\eta_L \prec \eta_{Th_{C3L}} \quad \text{Eq. 3.23}$$

Equation 3.23 expresses that the thermal efficiency of the closed processes expansion based thermal cycle is greater than that of the Lenoir cycle.

This result means a useful guide to focus the study on the basis of extracting heat from the cycle according to a closed process, which avoids the waste of heat due to the flow work involved in open processes, such as, in the case of Lenoir cycle.

3.3.2 The closed processes contraction based thermal cycle

RON 8 of Tab. 3.2 is implemented in a C3L and represented in Fig. 3.8. The T-s diagram of a physically realisable reversible thermal cycle, which is composed by three transformations as described in Tab. 3.4.

While for conventional cycles:

$$q_i - q_o = w_n \qquad \text{Eq. 3.24}$$

in this cycle, a C3L contraction-based cycle follows that:

$$q_o - q_{ic} = w_{cont} \qquad \text{Eq. 3.25}$$

The heat added at constant pressure during a quasistatic isobaric expansion without useful work is used to expand the WF (flow work) and to raise the internal energy according to the Eq. 3.26:

$$q_i = q_{12} = \Delta h_{12} = Cp\Delta T_{12} = \Delta u_{12} + p\Delta V_{12} = Cv\Delta T_{12} \qquad \text{Eq. 3.26}$$

where,

Δu_{12}, is the boost of internal energy along the transformation 1-2

$p \cdot \Delta V_{12}$, is the flow work due to the isobaric expansion (isobaric increment of volume without doing useful work).

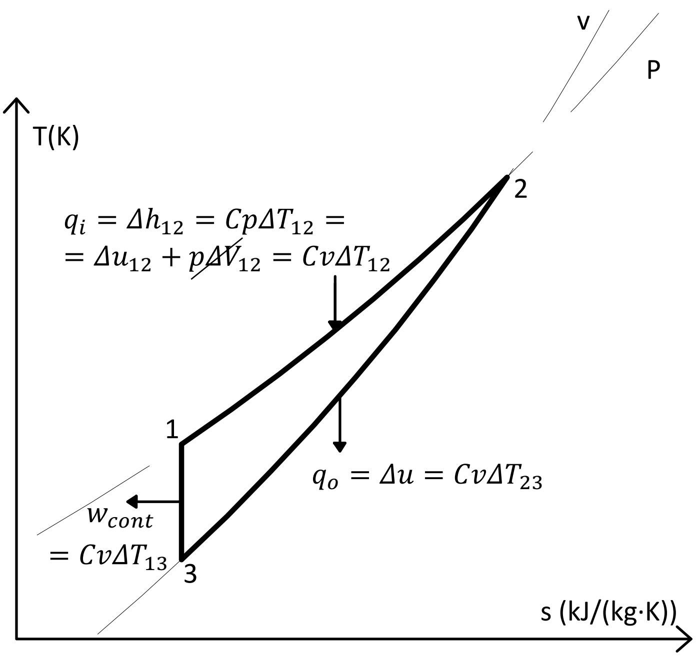

Fig. 3.8 The T-s diagram of a C3L isochoric cooling cycle.

Tab. 3.4 The closed processes-based expansion based thermal cycle, showing the transformations carried out for every path function.

Cycle process	Thermal process and path function	Function model
1-2	Closed quasistatic isobaric expansion, q_i	$q_i = Cv \cdot (T_2 - T_1)$
2-3	Closed isochoric cooling, q_o	$q_o = Cv \cdot (T_2 - T_3)$
3-1	Closed quasistatic isentropic contraction, w_{cont}	$w_{cont} = Cv \cdot (T_3 - T_1)$

3.3.2.1 Energy balance of the closed processes contraction-based thermal cycle

The net input heat to the cycle (heat absorbed by the WF) at point 2 is then:

$$q_{ic} = q_i - p\Delta V_{12} = q_i - w_{12} = \Delta u_{12} = Cv(T_2 - T_1) = Cv\Delta T_{21} \quad \text{Eq. 3.27}$$

The useful work as contraction-based compression work is given as:

$$w_n = w_{cont} = q_o - q_{ic} = Cv \cdot \Delta T_{23} - q_{ic} = Cv\Delta T_{23} - Cv\Delta T_{21}$$
$$= Cv\Delta T_{13} \quad \text{Eq. 3.28}$$

which differs from the conventional definition stated as: $w_n = q_i - q_o$

The ratio of the useful work or energy conversion due to the adiabatic contraction-based compression to the heat transferred to the heat sink as a performance index is given as:

$$\frac{w_n}{q_o} = \frac{u_{13}}{u_{23}} = \frac{Cv \cdot \Delta T_{13}}{Cv \cdot \Delta T_{23}} = \frac{\Delta T_{13}}{\Delta T_{23}} \quad \text{Eq. 3.29}$$

These results suggest that a more thorough analysis is necessary involving heat-work interactions, nor formerly considered.

Although the thermal cycle analysis obeys rigorously the principle of conservation of energy and the first principle as well as the second law, the expression (3.29) does not follow conventional interpretation of the 1st Law $w_n = q_i - q_o$. The flow graph of the energy balance is depicted with Fig. 3.9, where the work done (the contraction-based compression work is defined as $w_n = W_{cont} = q_o - q_{ic} = Cv\Delta T_{13}$.

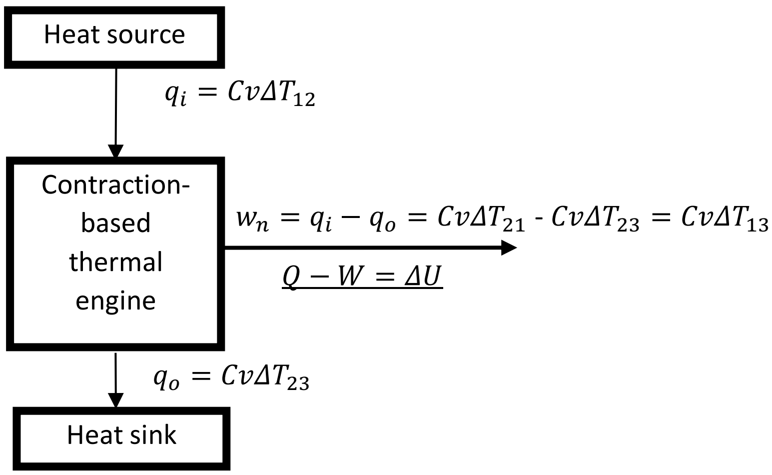

Fig. 3.9 Energy flow of the closed-processes based delivering work after cooling its WF.

The fact of finding the possibility of doing net mechanical work by means of extracting heat to the heat sink, provides with the opportunity to open the pace to try to use all the available high grade heat to use in a heating mode thermal cycle and all the available cold to include in a cooling mode thermal cycle, which let us combine both actions into a double acting reciprocating cylinder, as shown in next section.

3.3.3 The closed processes expansion-contraction based thermal cycle

The thermal cycle consists of a Cooling-Heating Closed-processes Non-condensing Cycle composed of four legs (C4L), which includes heating and cooling closed processes. This cycle is the result of joining the two previous cycles described previously in section 3.3.1 and 3.3.2 (See Fig. 3.10): The expansion-based and the contraction-based. As a result, two thermodynamic processes can be implemented to convert thermal energy into mechanical work in a single cycle. As previously explained, these cycles exploit thermal energy from an opposite point of view and their respective energy wasted in the heat sink can be utilised complementary to convert more energy. Cycles using this adiabatic contraction and expansion were used by Ferreiro et al. [50] combined with a heat pump. Results obtained show that they have potential for converting heat.

The first cycles published of this kind chose isobaric expansion and contraction [51–53]. The thermal cycle developed in this thesis involves isochoric heating, adiabatic expansion working, isochoric cooling and adiabatic contraction working. The main advantage is the possibility of reducing the expansion or contraction time, even if the work output will remain proportional to the heat absorption. This represent a mechanical advance to materialise the theoretical thermal cycle in a machine. With the aim of satisfying the 1st law, the corresponding energy balance is arranged subjected to the following rule: If $q_{12} - q_{13} - w_{23} = 0$, and $q_{31} - q_{34} - w_{41} = 0$, then it follows that:

$$(q_{12} - q_{13} - w_{23)} = (q_{31} - q_{34} + w_{41}) = q_{12} - w_{23} - q_{34} + w_{41} = 0 \qquad \text{Eq. 3.30}$$

which yields the energy balance as:

$$q_i - q_o = w_o - w_i \rightarrow q_{12} - q_{34} = w_{23} - w_{41} \qquad \text{Eq. 3.31}$$

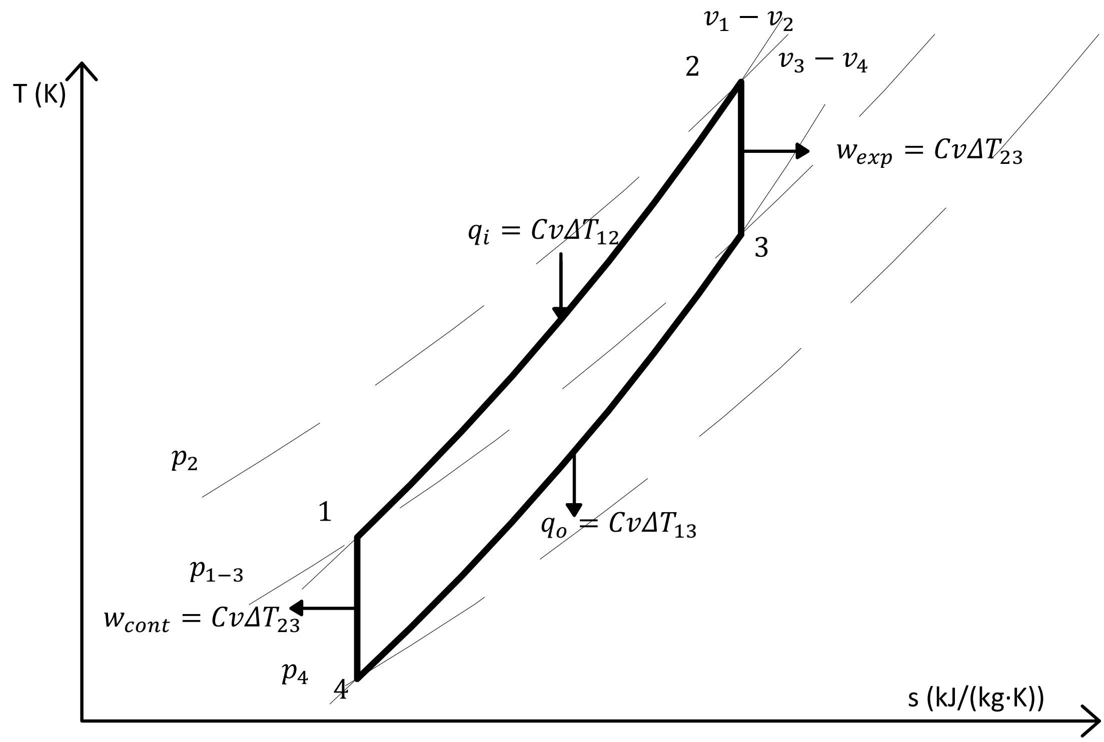

Fig. 3.10 Temperature-Entropy C4L diagram.

However, while for thermal cycles associated with work-based heating, the net work has been conventionally defined as $w_n = q_i - q_o = w_{23} - w_{41}$, when dealing with contraction-based work, the net work of the cycle is the absolute value of the partially useful work which yields:

$$w_n = w_{23} + abs|w_{41}| \qquad \text{Eq. 3.32}$$

instead of $q_i - q_o$

The thermal efficiency from (12) is given as:

$$\eta_{th} = \frac{w_n}{q_i} = \frac{w_{23} + abs|w_{41}|}{q_{12}} \qquad \text{Eq. 3.33}$$

Which written as function of its state temperatures, yields:

$$\eta_{th} = \frac{w_n}{q_i} = \frac{Cv(T_2 - T_3) + Cv(T_1 - T_4)}{Cv(T_2 - T_1)} = \frac{(T_2 - T_4) + (T_1 - T_3)}{(T_2 - T_1)} \qquad \text{Eq. 3.34}$$

Tab. 3.5. Thermodynamic processes of the C4L.

Cycle process	Associated path functions	Function model
1-2	Closed isochoric heat addition, q_i	$q_i = Cv \cdot (T_2 - T_1)$
2-3	Closed quasistatic isentropic expansion, w_{exp}	$w_{exp} = Cv \cdot (T_2 - T_3)$
3-4	Closed isochoric cooling, q_o	$q_o = Cv \cdot (T_3 - T_4)$
4-1	Closed quasistatic isentropic contraction, w_{cont}	$w_{cont} = Cv \cdot (T_1 - T_4)$

This expansion-contraction based cycle is characterized by performing useful mechanical work both by adding heat transferring heat from an external heat source at a certain temperature to the cycle, and by extracting heat from the cycle by transferring heat from the cycle to a heat sink at a lower temperature. (See Tab 3.5) For structural and operational purposes, this thermal cycle is equivalent to the two cycles described above.

The structure of such thermal cycle is depicted in Fig. 3.10, representing the T-s diagram of a physically realisable reversible thermal cycle composed of two closed isochoric processes consisting of adding and extracting heat to / from the WF and two closed adiabatic processes consisting of adiabatic- isentropic expansion and adiabatic-isentropic contraction-based compression of the WF which undergoes useful mechanical work.

Following the conclusion about expression Eq. 3.15 the C4L based on adding and extracting heat to / from a WF is subjected to an analysis which fulfils the principle of the conservation of energy if the appropriated conditions are met. The objective of section 3.3.3.1 is to verify the fulfilment of the principle of the energy conservation.

3.3.3.1 Closed processes expansion-contraction based cycle balance:

To verify the fulfilment of the 1st law of thermodynamics an energy balance of the cycle is required.

$$q_i = u_2 - u_1 = Cv(T_2 - T_1)$$

$$q_o = u_3 - u_4 = Cv(T_3 - T_4) \qquad \text{Eq. 3.35}$$

$$w_{exp} = w_{23} = \frac{p_2 \cdot v_2 - p_3 \cdot v_3}{\gamma - 1} = u_2 - u_3 = Cv(T_2 - T_3)$$

$$w_{cont} = w_{41} = \frac{p_4 \cdot v_4 - p_1 \cdot v_1}{\gamma - 1} = u_4 - u_1 = Cv(T_4 - T_1) \qquad \text{Eq. 3.36}$$

From Equation 3.35 and Equation 3.36 flows that:

$$q_i - q_o = (u_2 - u_1) - (u_3 - u_4) = (u_2 - u_3) - (u_1 - u_4) = w_{23} - w_{14} \qquad \text{Eq. 3.37}$$

But $q_i - q_o = w_{23} - w_{14}$ in Equation 3.37 is not the net useful work, then this unexpected result does not follow the definition of the net work used so far for conventional thermal cycles, which expressed as $q_i - q_o = w_n$ is conventionally correct.

Thus, Equation 3.37 is not an appropriate interpretation of principle of conservation of energy.

As demonstrated, the cycle net work is the result of adding the partial useful works w_{23} and w_{14}, as shown in Fig. 3.11, so that it must be expressed as:

$$w_n = w_{23} + w_{14} \qquad \text{Eq. 3.38}$$

In terms of performance, the ratio of the useful work due to adiabatic expansion to the heat transferred to the cycle from the heat source is given as:

$$\frac{w_{23}}{q_i} = \frac{u_{23}}{u_{12}} = \frac{Cv \cdot \Delta T_{23}}{Cv \cdot \Delta T_{12}} = \frac{\Delta T_{23}}{\Delta T_{12}} \qquad \text{Eq. 3.39}$$

and the ratio of the useful work due to adiabatic contraction-based compression to the heat transferred to the heat sink is given as:

$$\frac{w_{14}}{q_o} = \frac{u_{14}}{u_{34}} = \frac{Cv \cdot \Delta T_{14}}{Cv \cdot \Delta T_{34}} = \frac{\Delta T_{14}}{\Delta T_{34}}$$

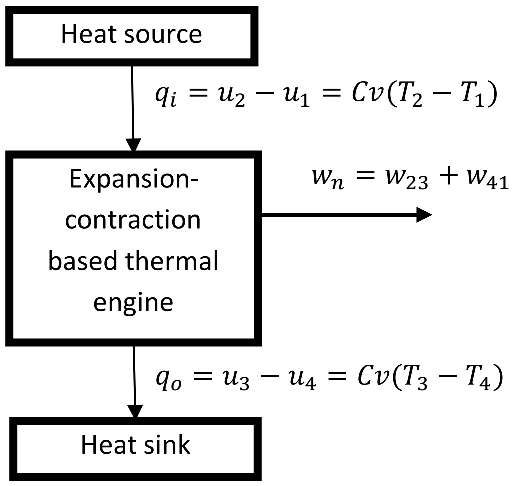

Fig. 3.11 Energy flow of the expansion-contraction thermal cycle.

The controversial result given in Equation 3.33 should be accommodated in the first principle as a particular case of a general cycle modelling mode.

Some facts verified in both C3L and C4L are:

(a) Heat extracting to the heat sink or heat extraction has been carried out under a closed isochoric process.

(b) Mechanical work has been performed by means of a closed contraction-based compression undergoing an adiabatic-isentropic process.

3.4 Entropy balance of the proposed thermal cycles

Formulation of an entropy balance is required for the proposed analysis. Thus, it has been observed that in all irreversible heat transfer processes, entropy increases. As a consequence of this experimental fact, the second law states that entropy can be created but it cannot be destroyed. Therefore, the increase of entropy principle is expressed by means of an entropy balance formulation as:

$$\Delta S_{sys} = S_{trans} + S_{gen} \qquad \text{Eq. 3.40}$$

which means that the change in entropy equals the entropy transfer plus entropy generation. The transfer of entropy assumed also as entropy flow or entropy associated with a heat transfer process at constant temperature *T* is described as:

$$S_{trans} = \frac{q}{T} \qquad \text{Eq. 3.41}$$

and for the general case:

$$S_{trans} = \sum \frac{q_k}{T_k} \qquad \text{Eq. 3.42}$$

Since the temperature during a transformation in which entropy transfer and entropy generation take place cannot be constant, an approach to the mean temperature is applied.

A heat transfer process evolving from an initial state to a final state, based on irreversible heat transfer, undergoes a change in entropy which is described as the difference between the final entropy and the initial entropy. Therefore, using Equation 3.41 and Equation 3.42 yields:

$$\Delta S_{sys} = S_{final} - S_{init} = S_k - S_0 = \sum \frac{q_k}{T_k} + S_{gen} \qquad \text{Eq. 3.43}$$

As stated by Equation 3.43, the entropy change of a closed transformation is due to the entropy transfer accompanying heat transfer and the entropy generation within the system boundaries. Considering a closed transformation, it follows that no mass flow is included across its boundaries, and the entropy change is the difference between the initial and final entropies of the transformation. What occurs is that the entropy change of a closed system is due to the entropy transfer associated with heat transfer and the entropy generation within the system boundaries, as expressed with Equation 3.43. Here it has been assumed that the total entropy generated during a process can be determined by applying the entropy balance to an extended system that includes both the system and its immediate surroundings, where external irreversibilities may exist.

In the task of entropic analysis of the proposed expansion based thermal cycle two heat transfer processes are possible: the heat transfer with entropy transfer associated to entropy generation when adding heat to the cycle, and the heat rejection process with entropy decreasing. Since the isentropic-adiabatic expansion based path function carried out during points (2) – (3) does not alter the entropy of the proposed expansion based thermal cycle (there is no entropy generation), it follows that in order to fulfil the entropy balance as well as the second law, the increase in entropy during the process of adding heat (1) – (2) must be exactly equal to the decrease of entropy during the process of heat rejection (3) – (1):

$$\Delta S_{sys}\big|_{(1)-(2)} - \Delta S_{sys}\big|_{(3)-(1)} = 0 \qquad \text{Eq. 3.44}$$

The fulfilment of 2nd Law for Equation 3.44 requires that entropy generation be positive since cannot be zero which implies that some internal irreversibilities ($T_0 * S_{gen}$) must exists.

Since the heat rejection process is the only way in which the entropy of a fixed mass of WF can be diminished, in order to equalize the entropy during the cycle evolution, a necessary condition to balance the changes in entropy of the proposed expansion based thermal cycle, is required and thus carried out by means of heat rejection during transformation (3) – (1). This fact indicates that a necessary condition to even the cycle entropy implies a heat rejection process, transferring entropy to the surroundings, the environment or to any heat sink reservoir.

3.4.1 Entropy balance of the expansion based thermal cycle

Fig. 3.12 is the entropy-temperature diagram of a simulation of an expansion-based C3L between 300 K and 700 K. Tab. 3.6 quantifies temperatures, pressure, specific volume, specific entropy and specific internal energy. This is the information collected to analyse the entropy balance of the cycle.

Tab. 3.6 Example of cycle parameters of the C3L expansion-based using air as a WF.

State	T (K)	p (kPa)	v (m^3/kg)	s (kJ/kg·K)	u (kJ/kg)
1	300.0	100.00	0.861	5.706	214.3
2	700.0	233.30	0.861	6.333	512.7
3	556.1	100.00	1.596	6.333	401.8

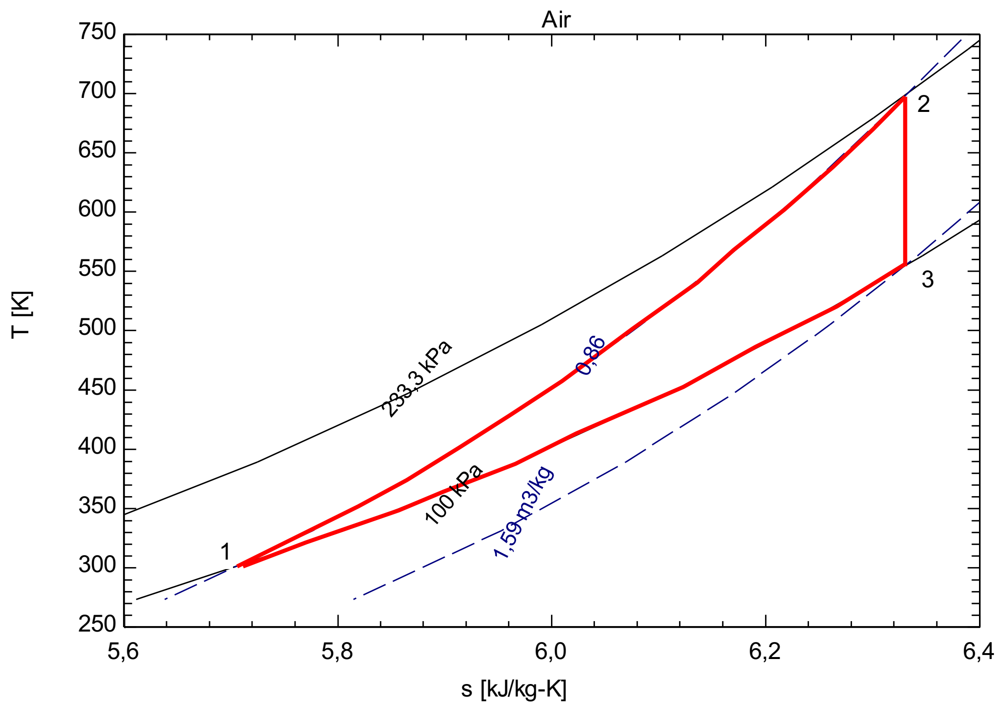

Fig. 3.12 The C3L expansion-based.

Applying equation 3.43 gives:

$$\Delta S_{sys}\big|_{(1)-(2)} - \Delta S_{sys}\big|_{(3)-(1)} = 0.627 - 0.627 = 0 \qquad \text{Eq. 3.45}$$

Eq. 3.45 demonstrates that the entropy transfer in the thermal cycle fulfils the 2nd Law assuming that there are some internal irreversibilities.

3.4.2 Entropy balance of the contraction based thermal cycle

Fig. 3.13 is the entropy-temperature diagram of a simulation of a contraction-based C3L between 300 K and 700 K. Tab. 3.7 quantifies temperatures, pressure, specific volume, specific entropy and specific internal energy. This information is needed to analyse the entropy balance of the cycle.

Tab. 3.7 Cycle parameters of the C3L operating by extracting heat only using air as a WF.

State	T (K)	p (kPa)	v (m^3/kg)	s (kJ/kg·K)	u (kJ/kg)
1	381.9	100.00	1.096	5.949	273.3
3	700.0	100.00	2.009	6.576	512.7
4	300.0	42.86	2.009	5.949	214.3

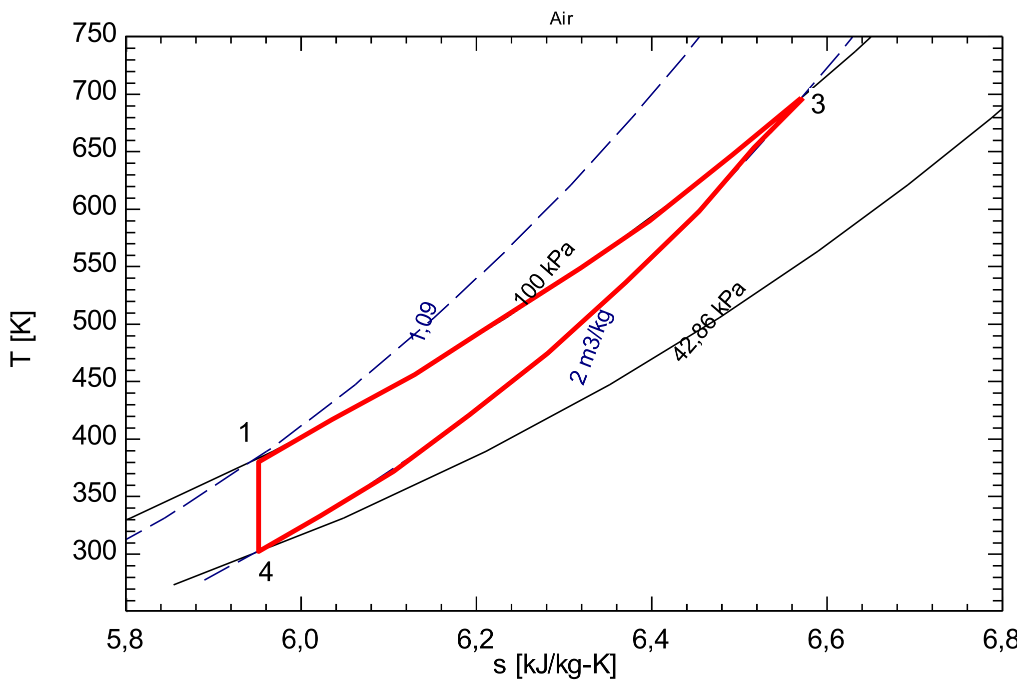

Fig. 3.13 The C3L contraction-based.

Applying equation 3.43 to this specific thermal cycle gives:

$$\Delta S_{sys}\big|_{(1)-(3)} - \Delta S_{sys}\big|_{(3)-(4)} = 0.627 - 0.627 = 0 \qquad \text{Eq. 3.46}$$

Eq. 3.46 proves that the entropy transfer in the thermal cycle fulfils the 2nd Law assuming that there are some internal irreversibilities.

3.4.3 Entropy balance of the expansion-contraction based heat thermal cycle

Fig. 3.14 is the entropy-temperature diagram of a simulation of a closed 4-Legs expansion-contraction-based (C4L) between 300 K and 700 K. Tab. 3.8 quantifies temperatures, pressure, specific volume, specific entropy and specific internal energy for a single and double acting mode respectively. This information is taken to analyse the entropy balance of the cycle and done using air as WF.

Tab. 3.8 Cycle parameters of the C4L in singe acting mode.

State	T (K)	p (kPa)	v (m^3/kg)	s (kJ/kg·K)	u (kJ/kg)
1	360.9	100.00	1.036	5.896	258.1
2	700.0	194.00	1.036	6.386	512.7
3	581.9	100.00	1.670	6.386	421.3
4	300.0	51.55	1.670	5.896	214.3

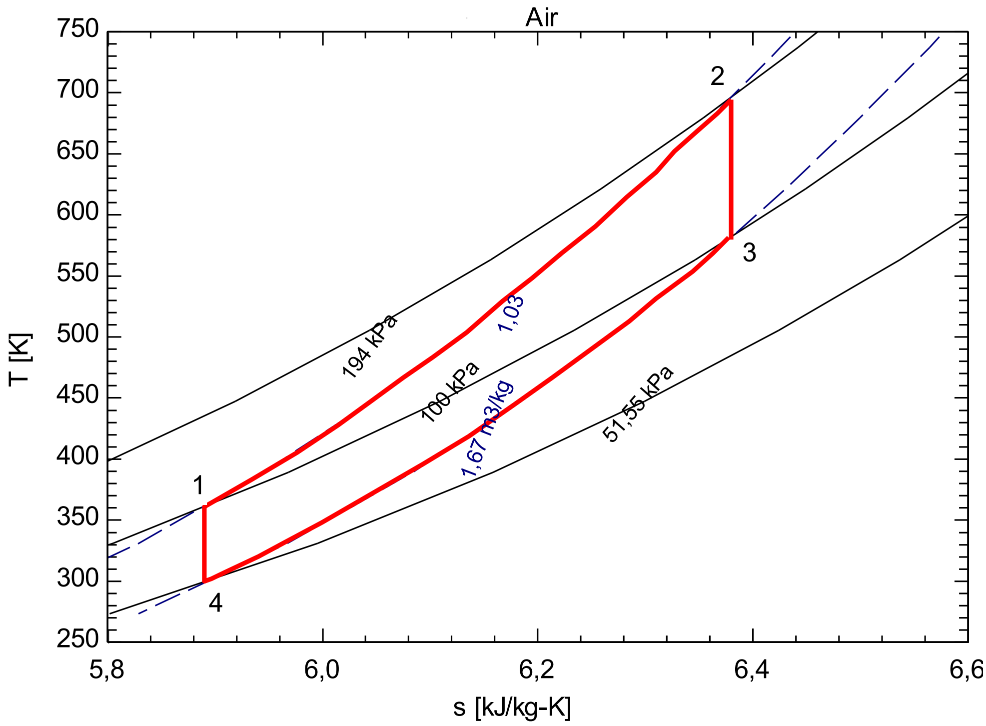

Fig. 3.14 The proposed expansion-contraction cycle performing contraction and expansion mechanical work.

Eq. 3.45 and 3.46 confirms that the entropy transfer in both thermal cycles fulfil the 2nd Law assuming that there are some internal irreversibilities.

Using data from Tab. 3.8 in equation 3.43 to this specific thermal cycle gives:

$$\Delta S_{sys}\big|_{(1)-(2)} - \Delta S_{sys}\big|_{(3)-(4)} = 0.49 - 0.49 = 0 \qquad \text{Eq. 3.47}$$

Eq. 3.47 confirms the fulfilment of the 2^{nd} Law for the cycle analysed in this section.

3.5 Experimental validation

An experimental validation is necessary for the suggested systems, delivering work while compressing had never been thought of to convert thermal energy into work. A process, which heats, increases pressure, and reduces volume being able to deliver work in theory.

Suction work is used in the industry, such as compressed air, vacuum pumps. In contrast, there is no trace in the literature of a closed, non-condensing, adiabatic system able to compress a fluid delivering mechanical work. The adiabatic compression is gotten cooling a fluid in non-condensing conditions in a closed adiabatic chamber, when the cooling process is finished the pressure and temperature of the mentioned chamber will decrease. Assuming an initial temperature and pressure equal or lower than the surroundings, the temperature and pressure at the end of the process has to be lower than in the surroundings. Therefore, the system can deliver work if the surrounding pressure compresses adiabatically the chamber. For instance, in a piston-cylinder.

Current literature assumes indisputably that compression processes use energy. Stirling, Ericsson and RC are constrained by this fact. However, this limitation is not a physical restriction and can be overcome in theory. For this reason, an experimental validation is carried on, to put into practice one of the simplest thermodynamic transformations, which can change radically the efficiency of a thermodynamic power cycle.

3.5.1 Setup

The setup is intended to realize as close as possible the thermodynamic processes of the proposed cycles. The main necessary components to validate the experiment must be able to heat/cool at constant volume, expand and contract as adiabatically as possible. In addition, the WF cannot condensate undergoing these processes. The components chosen were:

- Air as WF.
- Two copper heat exchangers.
- Water as cool and hot fluid.
- A pneumatic piston.
- Water heater
- Diverse material, such as, pipes, valves (V1, V2, V3, V4, V5, V6, V7, V8, V9 and V10) and manometer.

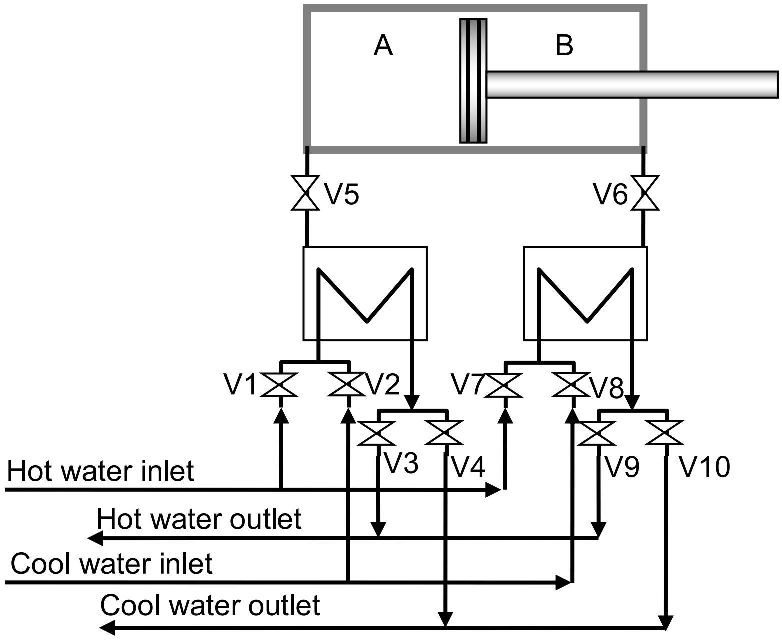

Fig. 3.15 Experimental setup.

As Fig. 3.15 shows this system is limited in terms of thermodynamic performances but it is intended to demonstrate that a compression in the mentioned conditions can deliver work. There are two main constraints in this experiment, which affects the efficiency but not its aim.

The first condition is the important amount of hot water wasted. The second limitation is thermal inertia of the components and the heat wasted while cooling and heating copper.

The experiment confirms the theory and demonstrates that work can be delivered while compressing in the mentioned requirements. The next chapter proposes physical mechanisms and thermodynamic architectures to implement the C4L, which can be found in the European Patent Office.

4 Proposed Alternative Engine Structures and Implementation

4.1 Introduction

The above engine architectures are being implemented undergoing isochoric expansion-contraction based thermal cycles. Such implementation alternatives are patented with application numbers P201700181, P201600386 and P201600414 respectively.

In the first proposal with patent number P201700181, the structure of the heat recovery system is suggested, whose objective is to achieve a technical architecture capable of recovering low-grade heat efficiently.

In the second approach with patent application number P201600386, the structure of the system of conversion of discontinuous force reciprocating motion to continuous rotary motion torque is included. Such discontinuous reciprocating movement is due to the need to perform the admission of heat energy or heat absorption, as well as the heat exhaustion under an isochoric thermal process, with which the volume of the enclosure from which heat is transferred (cylinder) must remain at constant volume. Therefore, the cylinder in the conditions of intake and exhaust of thermal energy must remain at constant volume

In the last proposal with patent application number P201600414, the structure of the system of conversion of discontinuous reciprocating motion to continuous rotary motion is set.

In the same way as in the previous case, such discontinuous alternative movement must be transformed into continuous rotation movement. Therefore, since the cylinder in such conditions of admission and exhaust of thermal energy has to remain at constant volume, the rotation movement remains constant

Thermal cycles that involve thermodynamic processes carried out at constant volume (isochoric) have the inherent disadvantage that every isochoric process is unable to perform work simultaneously because at constant volume it is not possible to perform mechanical work, since there is not volume change. However, the capacity of the power plant to convert low-grade energy into mechanical work justifies isochoric processes.

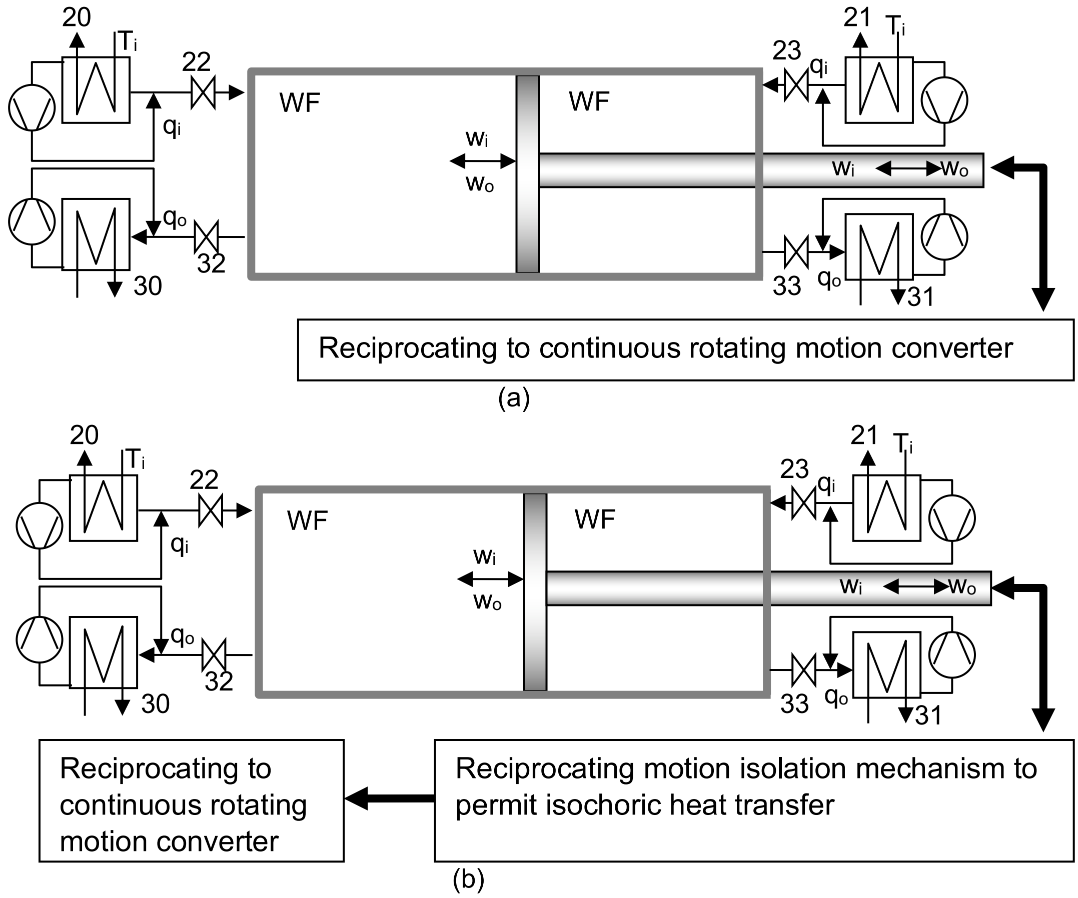

Fig. 4.1. Basic structural differences between reciprocating continuous motion and reciprocating discontinuous or impulsion-based motion.

Thus, in order to solve such problem, according to the scheme shown in Fig. 4.1, there are several options among which these can be outstood:

a. Implementation of the isochoric heat transfer process in an isolated enclosure of the actuator cylinder responsible for performing mechanical work.

b. Realization of the isochoric heat transfer process in the actuator cylinder responsible for performing mechanical work, so that while the piston remains motionless during the isochoric heat transfer process, the piston rod remains uncoupled from the mechanism responsible for transforming the thermal energy into mechanical work (connecting rod-crankshaft or another equivalent device)

The implementation of both proposed techniques as part of the innovations of the thesis, require different principles dealing with cycle timings, methodologies, operating strategies and mechanical structures. The main differences between both explained architectures are depicted in Fig. 4.1(a) and (b). Although apparently, they are very similar, there is a great difference with regards to the cycle timing. The cycle timing sequence is shown in Tab. 4.1.

From a practical point of view, it is interesting to consider a double-acting cylinder-based cycle, so that the tasks of adding and extracting heat to / from the double-acting actuator cylinder must be possible for every cylinder chamber. The heat transfer by thermal convection is only effective if it is carried out counter-current. As observed in Fig. 4.1 (a) and (b), the tasks of isochoric heating and cooling are carried out by thermal forced convection.

Tab. 4.1 Cycle tasks associated with a double-acting cylinder considering the status of the inlet and outlet cylinder valves and its associated heat exchangers.

Left to right cylinder stroke motion	
Left cylinder chamber Adiabatic expansion: reservoir (20), Valve (22). Isochoric heating reservoir (21)	Right cylinder chamber adiabatic contraction: reservoir (31), Valve (33). Isochoric cooling reservoir (30)
Right to left to cylinder stroke motion	
Left cylinder chamber adiabatic contraction: reservoir (30), Valve (32). Isochoric cooling reservoir (31)	Right cylinder chamber adiabatic expansion: reservoir (21), Valve (23). Isochoric heating reservoir (20)

4.2. Description of some proposed thermal engine structures

4.2.1 P [1] Regenerative double-acting reciprocating thermal engine

This section describes the regenerative double-acting reciprocating thermal engine which is based on the patent **P201700181** referenced as P [1].

Objective of the technology

The technology denominated regenerative double-acting reciprocating thermal engine, of closed and open processes and its operating procedure, has the objective of the conversion of thermal energy to mechanical and / or electrical energy via mechanical energy by means of a regenerative thermal cycle of double effect , implemented on an alternative thermal machine of double regenerative effect that operates with helium between two heat sources (hot source that yields heat to the cycle and cold source that absorbs heat from the cycle) destined to the conversion of thermal energy into mechanical work and / or electric power.

Description of the technology

The regenerative double-acting reciprocating thermal engine with, closed and open processes and its operating procedure, consists of an unconventional thermal cycle implemented by an alternative double-effect, regenerative thermal machine that operates with closed and open processes with a WF. It is constituted by an unconventional thermal cycle implemented by means of an alternative thermal machine of double effect, regenerative that operates with closed and open processes using a WF (helium), which is equipped with two thermal focus (one of high temperature for heat the working TF that acts as a heat source and another of low temperature to cool the TF that acts as a heat sink), where both focuses are constituted by heat exchangers, and where the hot focus consists of at least two heat exchangers, which alternatively transfer heat from a WF to the TF of the thermal cycle implemented by the thermo-actuator cylinder, and where the heat to heat the heat transfer fluid comes from any available source of heat such as compressor WF, nuclear energy, fossil fuel thermal energy, solar thermal, geothermal, high residual, medium heat and even at low temperature, and where the cold focus or heat sink consists of at least two heat exchangers, which alternatively transfer heat from a TF of the thermal cycle implemented by the thermo-actuator cylinder, to the WF, which is cooled by means of a refrigerant that can be air or water at room temperature, a steam compression refrigeration machine, the evacuation of a turbo-expander at sub-ambient temperature, or a conventional cooling tower by air or water, equipped with two high and low temperature thermal bulbs formed by heat exchangers), where the hot bulb consists of a heat exchanger that transfers heat from a TF to the WF of the thermal cycle implemented by the reciprocating double-acting alternative thermo-actuator cylinder, where the heat comes from any Available from heat such as compressor WF, nuclear energy, fossil fuel thermal energy, thermosolar, geothermal, high, medium and even low temperature residual, and where the cold focus or heat sink consists of a heat exchanger that transfers heat from the WF of the thermal cycle implemented by the reciprocating double-acting reciprocating thermo-actuator cylinder, to a TF, which is cooled by means of a refrigerant that can be air or water at room temperature, a steam compression refrigeration machine, the evacuation of a turbo expander at sub-ambient temperature, or a tower conventional cooling by air or water.

Description of the components

In this section, the components that make up the regenerative double-acting reciprocating thermal engine, to facilitate the understanding of the technology are described in an illustrative and non-limiting manner, where reference is made according to Tab. 4.2:

Tab. 4.2 Structure of the alternative double-effect regenerative thermal machine.

Ref.	Component	Ref. Number	Component
1	double-acting thermo-actuator	20, 21, 30, 31	Heat exchanger
2	Piston	22, 23, 32, 33	Intercommunication
14	Unloading duct	24, 25, 34, 35	Pressure regeneration
13	Compressor	26, 27, 36, 37	Valves
12	Expander	15	Return duct
17	Return conduit	16	Evacuation duct

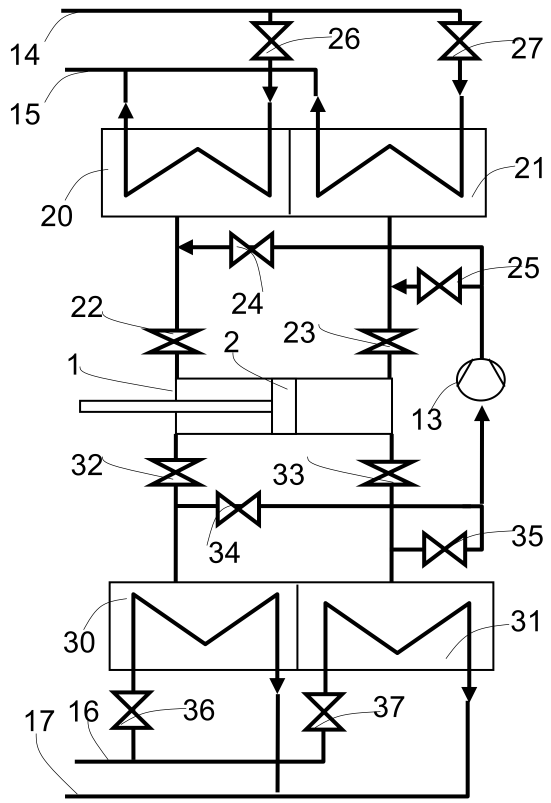

Fig. 4.2. alternative double-effect regenerative thermal machine.

Fig. 4.3 shows the structure of two power units coupled in series formed by their respective double acting thermos-actuator cylinders (1), fed by ducts (14) and (15) for high temperature thermal source and ducts (16) and (17) for the heat sink.

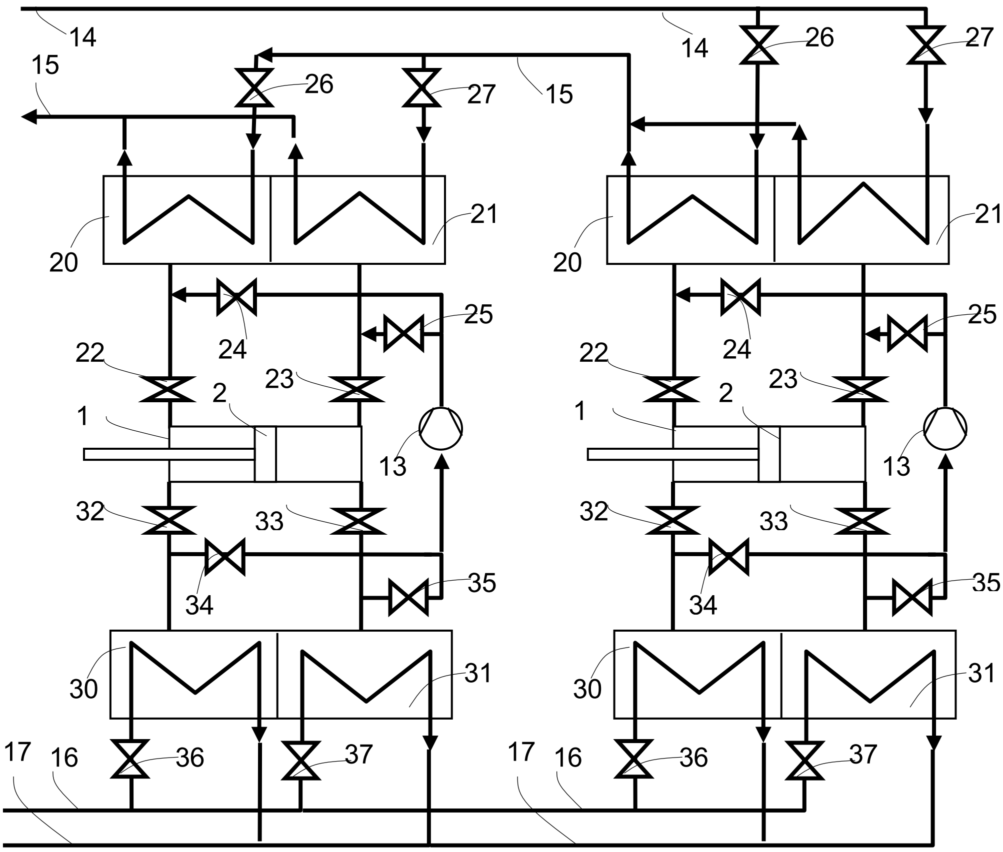

Fig. 4.3 Two power units coupled in series.

Fig. 4.4 shows the structure of the machine object of the technology provided with the means for heating and cooling or extracting heat from the working TF, for which several alternative conventional technical methods of supplying thermal energy for heating the WF are proposed, as well as means of heat extraction for cooling the WF. For instance, for the case of heating and cooling by reverse Brayton cycle.

In case heat of another type of energy is used as WF of the compressor, nuclear energy, or thermal fossil combustion, or thermosolar, or geothermal, or residual high temperature, medium and even low temperature, as well as cold coming from air or water at room temperature, from vapour compression based refrigerating or sub-environmental temperature, or the evacuation of turbo-expanders at sub-environmental temperature, or a conventional cooling tower, is equipped with ducting and inlet and outlet valves of the heat TF both for heat supply to heat the WF and for heat extraction to cool the WF, which includes the following components:

Tab. 4.3 Additional components of the architecture in Fig. 4.4.

Ref. Number	Component
44, 45, 46, 47	Valves
40, 41, 42, 43	Valves
10	Starter motor
11	Compressor
12	Turbo-Expander

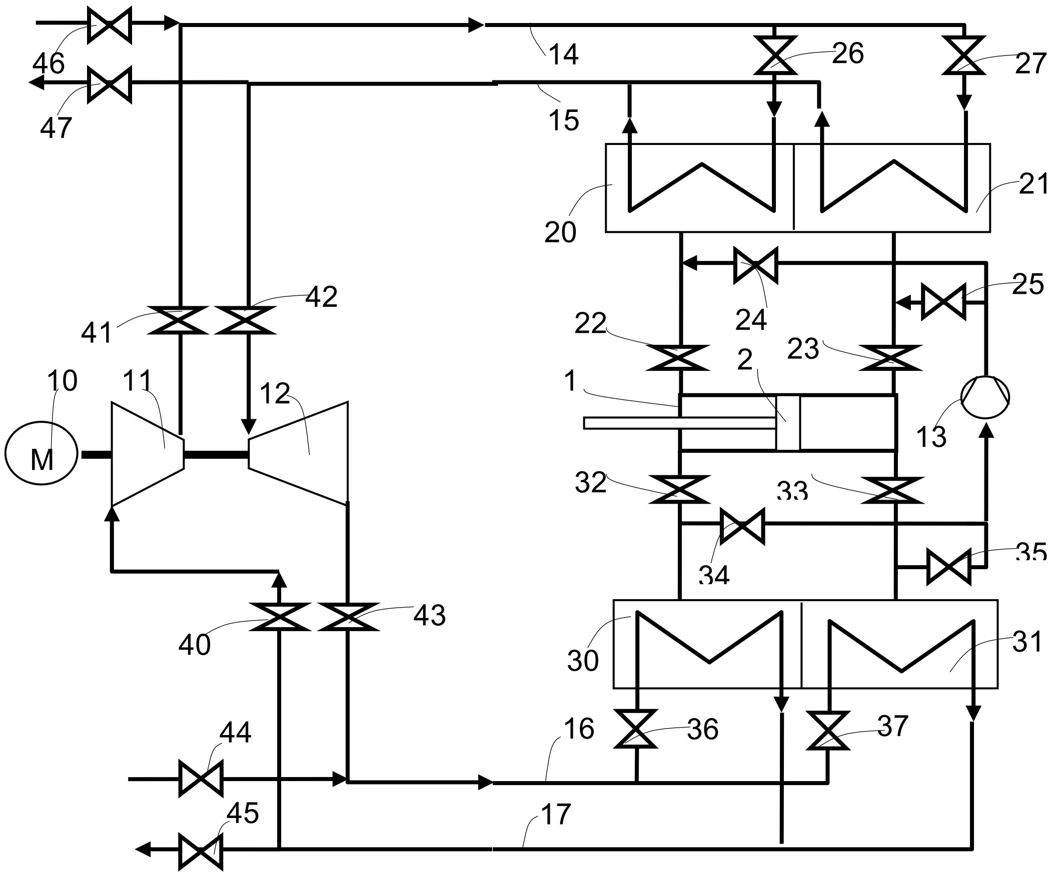

Fig. 4.4 Structure of the machine.

4.2.2 P [2] Reciprocating force convertor, discontinuous to continuous rotary torque

This section describes the reciprocating force convertor, discontinuous to continuous rotary torque which is based on the patent **P201600386** referenced as P [2].

Objective of the technology

The object of the present technology called reciprocating force convertor discontinuous to continuous rotary torque, is the implementation of thermal energy converted to mechanical energy by a conventional thermo-mechanical cylinder that operates with a thermal cycle including closed processes at constant volume during which the piston remains at rest, while the power supply shaft rotates at a continuous speed. Between the thermo-mechanical cylinder and the shaft of crankshafts, two hydraulic cylinders are installed, being the first to block the movement of the piston and rod of the thermo-mechanical cylinder to achieve thermo-mechanical processes at constant volume, while the second is enabled to connect the force provided by the first hydraulic cylinder with rod-connecting rod-crankshaft to achieve a torque of rotation.

Description of the technology

The technology denominated Reciprocating force convertor discontinuous to continuous rotary torque consists in a mechanism destined to capture the force developed by a thermo-mechanical cylinder (engine or actuator) by means of two hydraulic cylinders and a movement converter of translation to rotation movement based on connecting rod-crankshaft in the crossover-connecting-crankshaft system.

The reciprocating force convertor discontinuous to continuous rotary torque and operating procedure thereof transfers the discontinuous force of the piston of a thermo mechanical actuator cylinder to a rotary axis by means of two intermediate hydraulic cylinders connected with its hydraulic circuit, being the first of them (locking cylinder) able to block the movement of the piston of the thermo-mechanical actuator cylinder to maintain its constant volume at each dead point of its stroke, while the second (movement coupling cylinder) is disconnected from the first and follows the rotary axis through a converter mechanism of rectilinear reciprocating to continuous rotary movement such as the connecting rod-crankshaft system, so that this shaft can rotate freely when the piston of the thermo-mechanical actuator cylinder remains locked and at rest condition.

Description of the components

In this section, the components constituting the reciprocating force discontinuous to continuous rotary torque are described with the help of an illustration and without limitation to facilitate the understanding of the technology where reference is made to the Fig. 4.5

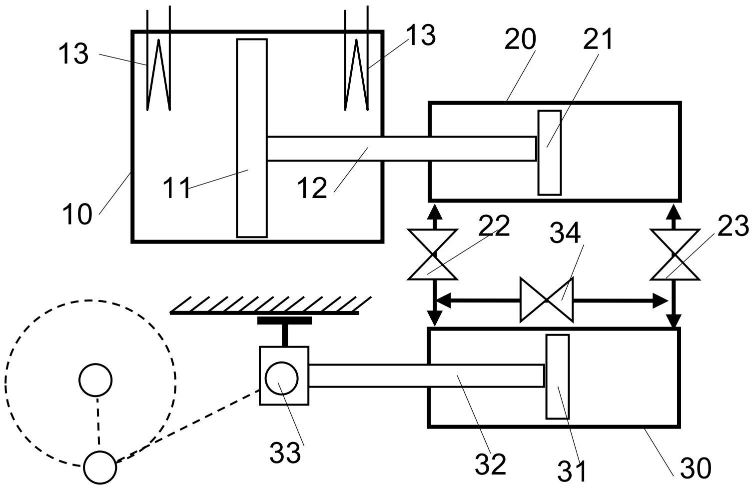

Fig. 4.5 The discontinuous reciprocating force converter to continuous rotary torque.

Fig. 4.5, mechanical structure of the discontinuous reciprocating force converter to continuous rotary torque to capture the force developed by a thermo-mechanical cylinder (engine or actuator) by means of two hydraulic cylinders and a converter of movement from translation to rotation movement based on crosshead -black-crankshaft. Its components are described in Tab. 4.4

Tab. 4.4 Components of the mechanism described in Fig. 4.5.

Ref. Number	Component	Ref. Number	Component
10	Double-acting thermo-mechanical actuator cvlinder	22, 23, 34	intercommunication valve
11, 12	piston rod	30	crankshaft-connecting rod
13	sources of heat or thermal energy	31	Piston
20	hydraulic cylinder for blocking movement	32	Rod
21	piston of the hydraulic cylinder for blocking the movement	33	Crosshead

4.2.3 P [3] Reciprocating discontinuous to continuous rotary motion

This section describes the reciprocating discontinuous to continuous rotary motion, which is based on the patent **P201600414** referenced as P [3]

Objective of the technology

The objective of the present technology called reciprocating discontinuous to continuous rotary motion, is the efficient conversion of thermal energy to mechanical and / or electrical energy by means of conventional reciprocating cylinders that operate with a thermal cycle that includes closed processes at constant volume during which the piston rod remains at rest, while the power supply shaft rotates at a continuous speed.

Description of the technology

The technology called reciprocating discontinuous to continuous rotary motion, with two implementations a and b, consists of the implementation a), in a mechanism formed by two hydraulic cylinders and a converter of movement from translation to rotation movement based on a toothed rack with movement linear reciprocating that engages with a partially toothed wheel in the axial direction with continuous rotation movement; and in the implementation b), in the first hydraulic cylinder capable of blocking the movement of the piston of the actuating cylinder to keep the volume of the cylinder constant at each end of its stroke while to lock and unlock the connection between the piston of the hydraulic cylinder and The rotary shaft uses a mechanical connecting device between rod and connecting rod to transform discontinuous reciprocating rectilinear motion to continuous rotary motion.

Reciprocating machines whose thermal cycles involving processes during which the volume of the cylinder remains constant (piston and piston rod at rest), are subject to a discontinuous reciprocating movement, which is not useful for coupling directly to a crankshaft by means of the conventional connecting rod. For this reason, the rod of the actuator cylinder is coupled to a translation converter from translation to rotation movement by means of two intermediate hydraulic cylinders in implementation a), or a mechanical connection system in implementation b).

The technology called reciprocating discontinuous to continuous rotary motion transfers the discontinuous and reciprocating movement of the piston and its rod from a double acting actuator cylinder to a rotating shaft by means of the discontinuous reciprocating translation motion converter to continuous rotation movement, which is carried out in the implementation a) by means of two intermediate hydraulic cylinders connected by their pistons and rods, the first of which being responsible for blocking the movement of the actuator cylinder piston to maintain the volume of the constant cylinder at each end of its stroke, while the second disconnects the connection between the piston of this hydraulic cylinder and the rotating shaft so that this shaft can move freely when the piston of the actuating cylinder remains at rest, while in the implementation b) The coupling and uncoupling between the rod of the actuator cylinder and the rod is done by a mechanical system.

Description of the components

This section describes with an illustration and not limiting the components that make up the reciprocating discontinuous to continuous rotary motion to facilitate the understanding of the technology where reference is made to Fig 4.6:

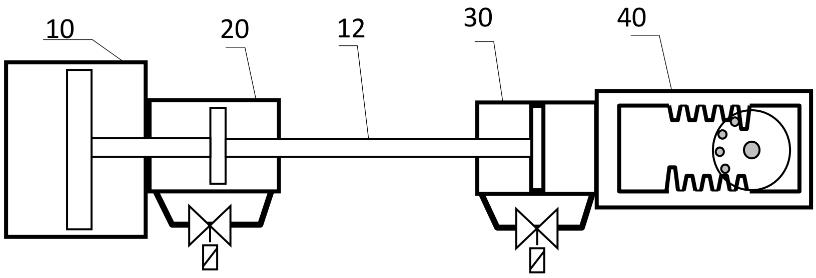

Fig. 4.6 reciprocating and discontinuous translation motion converter to continuous rotation movement.

Tab. 4.5 Components of the mechanism described in Fig. 4.6.

Ref. Number	Component	Ref. Number	Component
10	Double acting actuator cylinder	12	Piston rod
20	Hydraulic cylinder for blocking the movement	40	Converter of reciprocating rectilinear movement to rotation movement
30	Cylinder		

5 Case studies: The implementation of WHRS on ship's propulsion plants

The marine industry offers an attractive field to extract WH, since the weight of the fuel is important. Thus, a more efficient power plant reduces the fuel mass for the same trip and this can easily overcome the additional weight/space lost due to the installation of efficient bottoming cycles. This work will use commercial marine engines to evaluate the potential of a C4L as bottoming cycle. Ming M. et al. [54] studied the variation of emissions of carbon dioxide. They concluded that carbon dioxide emissions are sensitive to electricity efficiency. In addition, they came to the statement that the amount of carbon dioxide emissions will increase at least until 2030. In the year of the publication, the carbon dioxide emissions are expected to be between 60 % and 80 % higher than in 2011. Zaman K. et al. [55] found that there is a direct relation between economic growth and carbon dioxide emissions. Economic growth tends to be globally positive. Therefore, pollutant emissions in power production are an important challenge and the marine sector has the potential to reduce global warming. The highly efficient model presented can definitively help to diminish marine emissions.

Heat recovery in marine power plants is explored and so are three different architectures. Commercial systems use mainly serial-based heat recovery, which wastes a high amount of heat due to the impossibility to transfer a significant quantity of heat from the source to the WF on account of the temperature differences. For instance, water preheated with Jacket water will probably be heated up to 353 K, then, this water will not be able to recover lower temperature heats, for instance from the scavenge air. This fact limits the heat recovery capacity of the system.

This section also examines the potential of an innovative C4L thermodynamic cycle used to convert low-grade heat into mechanical work. A patented prototype of a machine working with a C4L is described in this section. The machine suggested in this section is used in architectures that follows P [1].

The machines exhibited work out with waste energy from the reviewed marine engines to assess its potential. The thermal efficiencies of power plants using the displayed model are evaluated and compared. The analysis of this section, shows that the proposed machine using closed processes achieve outstanding results compared to other machines running on low-grade heat. Thus, this section demonstrates that the application of non-condensing closed thermal power cycles reviewed in this thesis can increase significantly the thermal efficiency of existing power plants. Its HUF surpasses the current technology. Therefore, this innovative concept can convert efficiently energy from low-grade heat sources. Results of the preliminary study shows that the suggested machine can convert low grade heat to mechanical work extracting heat from a wide range of low-grade heat sources, not utilised by the current technology.

5.1 Marine engines used to evaluate the potential of the Cooling-Heating Closed Non-Condensing Machine

Six industrial engines were selected to review the recovery potential of the WH in the marine industry. The mechanical power output of these engines ranges from 82440 kW to 11120 kW. Tab. 5.1 and 5.2 show data from their respective waste-heat in the 4 main sources of low-grade waste-heat: Exhaust gas, scavenge air, lubrication oil and cooling water. This data is used to quantify and define the WH.

Tab. 5.1 Waste-heat temperature and flows from exhaust gases and scavenge air.

Model-Power at full load	$\dot{Gas}_{Exh}$ (kg/s)	T_{exh} (K)	$\dot{Air}_{SC}$ (kg/s)	T_{SC} (K)
S90 – 69.72 MW [11]	168.0	513	164.8	499.4
G95 – 82.44 MW [11]	155.1	518	163.7	505.2
G80 – 42.39 MW [11]	90.2	508	88.3	503.4
G70 – 29.12 MW [11]	62.8	508	56.6	508.2
G60 – 22.70 MW [11]	49.0	508	48.0	503.3
G45 – 11.12 MW [11]	23.7	513	23.2	495.9

Tab. 5.2 Waste-heat temperature and flows from lubrication oil and cooling water.

Model-Power at full load	$\dot{m}_{oil}$ (kg/s)	T_{oil} (K)	$\dot{m}_{water}$ (kg/s)	T_{Water} (K)
S90 – 69.72 MW [11]	276.0	330.7	150.0	353
G95 – 82.44 MW [11]	337.3	329.5	158.3	358
G80 – 42.39 MW [11]	204.4	327.8	83.3	358
G70 – 29.12 MW [11]	145.7	327.8	55.5	358
G60 – 22.70 MW [11]	109.9	327.7	44.4	358
G45 – 11.12 MW [11]	61.3	327.0	22.2	353

5.2 Marine WHRS

There are two main strategies to recover WH: Serial and Parallel systems. The serial architecture is the simplest with the drawback of having an inherent low efficiency. The low efficiency of a serial recovery system is caused by the increase of coolant temperature in each residual heat source. In contrast, parallel architectures can overcome the mentioned disadvantage of serial architectures. Commercial systems use serial architectures, which limits their potential. In addition, parallel architectures can deliver a convenient temperature of the WH. For instance, heat from the lubrication oil can be bypassed in order to increase the final temperature of the TF, with the drawback of having less energy recovered. Therefore, a parallel architecture involving the three and two higher temperature sources is also examined. The following architectures will be coupled to residual heat from internal combustion marine engines formerly referenced to on Tab. 5.1-5.2.

5.2.1 Parallel WHRS

Parallel architectures offer better performances than serial architectures. The main advantage of these systems is the low temperature of the coolant inlet. In addition, this architecture allows to tune the flow for each section. Therefore, outlet temperatures of the coolant can be controlled or adapted to match a minimum temperature. For instance, lubrication oil cannot be cooled infinitely. The oil has a minimum temperature, typically about 320 K. This architecture makes possible the use of more than one flow of coolant. Fig. 5.1 shows how 4 different flows can extract heat separately. Fig 5.2 applies the architecture using heat from S90 – 69.72 MW [11]. Thus, these systems can use a coolant for low-temperature heat sources and another one for higher temperature residual heat. This idea can improve the thermal efficiency of thermal cycles using residual heat to extract power.

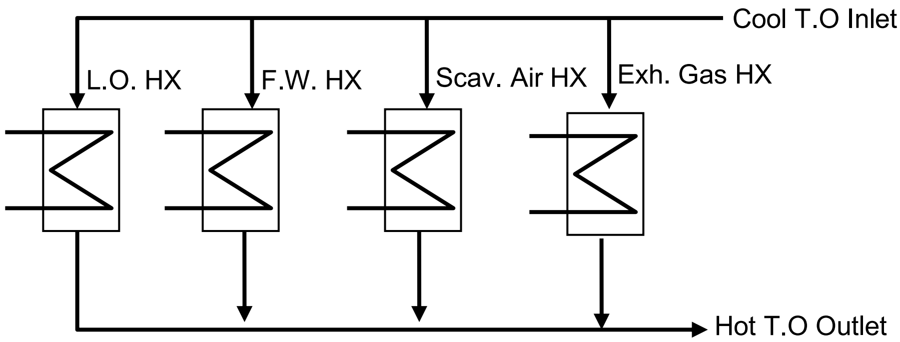

Fig. 5.1 Parallel architecture.

As an example, this recovery system guarantees the engine cooling and delivers 482.9 kg/s of oil with a specific heat of 1670 J/(kg·K) at 394 K for a S90 – 69.72 MW [11]. This temperature is below the temperature acquired by the same WH with the serial architecture. However, there is more than twice the amount of flow. Therefore, the heat output is obviously higher.

Performances for the other engines analysed in this chapter can be found in Tab. 5.3. The table shows the temperature and flow of thermal oil obtained for the different engines proposed in this chapter.

Temperature values are low. In contrast, the thermal energy absorption is efficient. Therefore, the heat absorption capacity is high. Low temperatures do not allow an efficient heat transfer to the WF, which is a typical problem when dealing with low grade heat. As described in the corresponding section, the aim of this thesis is to look into the conversion of low-grade heat into mechanical work. Thus, having higher amounts of heat with the constraint of having low temperatures is not an obstacle for the development of this thesis. Such kind of low-grade energy is convenient to apply to a conversion system, whose efficiency is not as dependant in temperatures as it is for conventional engines.

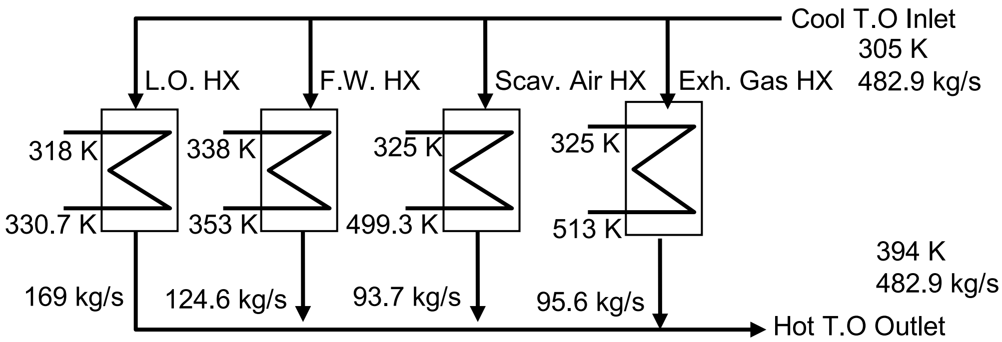

Fig. 5.2 Parallel architecture using heat from S90 – 69.72 MW [11].

Tab. 5.3 Coolant output obtained for the different engines with a parallel architecture.

Model-Power at full load	T (K)	$\dot{m}$ (kg/s)	$\dot{P}_{th}$ (MW)	Case
S90 – 69.72 MW [11]	394.0	482.9	71.8	1
G95 – 82.44 MW [11]	390.9	507.6	72.8	2
G80 – 42.39 MW [11]	388.4	281.0	39.1	3
G70 – 29.12 MW [11]	387.1	195.0	26.7	4
G60 – 22.70 MW [11]	388.8	151.4	21.2	5
G45 – 11.12 MW [11]	382.2	80.6	10.4	6

5.2.2 Parallel Scavenge Air and Exhaust Gas WHRS

Figure 5.2 shows a separated parallel marine heat recovery system. For a S90 – 69.72 MW [11], this recovery system guarantees the respective cooling and delivers 189.3 kg/s of oil with a specific heat of 1670 J/(kg·K) at 486.2 K. Parameters for the other engines scrutinized in this chapter can be found in Tab. 5.4. Thermal power regained is lower than in the previous architecture. Fig 5.4 applies the architecture using heat from S90 – 69.72 MW [11], in this case heat can be extracted at 486.2 K.

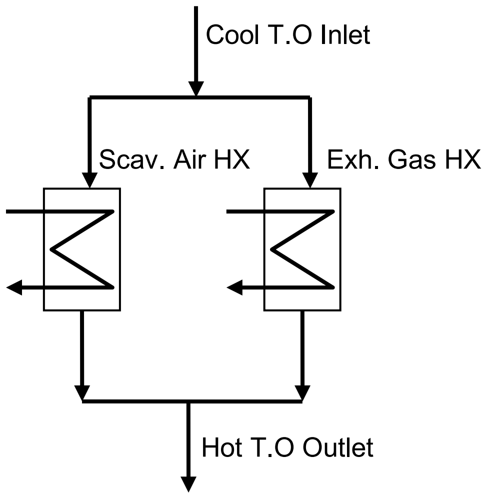

Fig. 5.3 Parallel separated marine heat recovery system configuration.

The architecture can give heat at higher temperature (hot side). If higher temperatures are available, heat transfer to the WF is more efficient. Therefore, more power can be gained from highest temperatures.

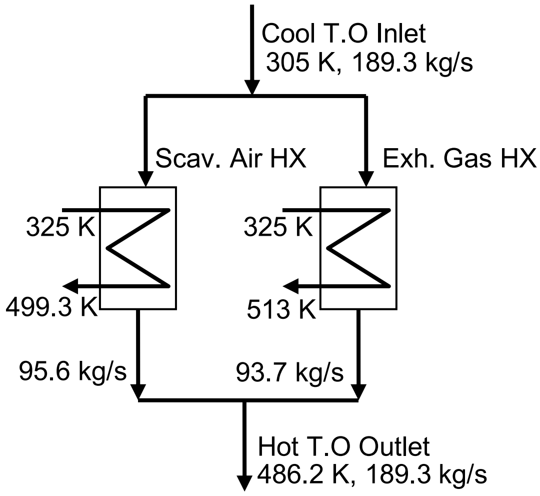

Fig. 5.4 Parallel separated marine heat recovery system configuration using heat from S90 – 69.72 MW [11].

Tab. 5.4 Coolant output of the exhaust and scavenge air heat obtained for the different engines with a parallel separated architecture.

Model-Power	T_2 (K)	$\dot{m}_2$ (kg/s)	$\dot{P}_{th_2}$(MW)	Case
S90 – 69.72 MW [11]	486.2	189.3	57.3	7a
G95 – 82.44 MW [11]	491.4	181.4	56.5	8a
G80 – 42.39 MW [11]	485.8	101.5	30.7	9a
G70 – 29.12 MW [11]	488.1	67.9	20.8	10a
G60 – 22.70 MW [11]	485.7	55.2	16.7	11a
G45 – 11.12 MW [11]	484.6	26.7	8.0	12a

5.2.3 Parallel Scavenge Air, Exhaust Gas and Jacket Water WHRS

Jacket water stands for an important amount of heat, the temperature is not high compared to the scavenge and exhaust heat. In contrast, compared to these two sources, it means around the 15 % of the WH in a system that recovers heat from the exhaust, scavenge and jacket water. Accordingly, the final temperature is dumped with the advantage of adding more heat to the recovery system. Fig 5.5 shows the architecture and Fig. 5.6 illustrates the diagram of such system using heat from S90 – 69.72 MW [11].

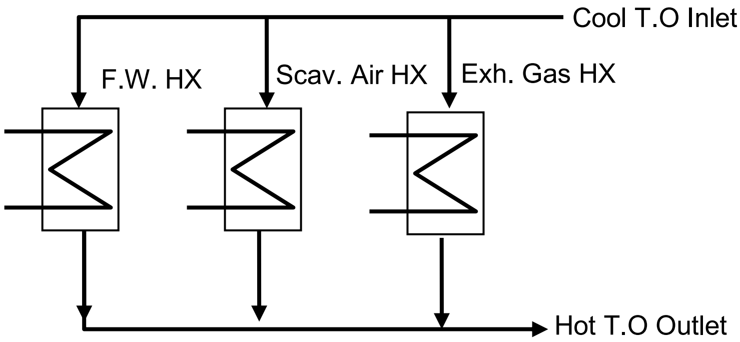

Fig. 5.5 Parallel separated marine heat recovery system configuration.

Tab. 5.5 Values of the coolant extracting heat from exhaust, scavenge air and water coolant.

Model-Power	T (K)	$\dot{m}$ (kg/s)	$\dot{P}_{th}$ (MW)	Case
S90 – 69.72 MW [11]	431.4	313.9	66.2	13
G95 – 82.44 MW [11]	434.3	308.7	66.7	14
G80 – 42.39 MW [11]	433.4	167.7	36.0	15
G70 – 29.12 MW [11]	433.4	114.2	24.5	16
G60 – 22.70 MW [11]	433.6	90.8	19.5	17
G45 – 11.12 MW [11]	425.9	46.8	9.4	18

Tab. 5.5 show the parameters for the studied engines. Data demonstrate a clear decrease in temperature and an increase in power extracted when comparing with the separated parallel system. Besides, it also has a higher temperature than the simple parallel system. Therefore, this system is in terms of power and temperature between the simple parallel and the separated parallel.

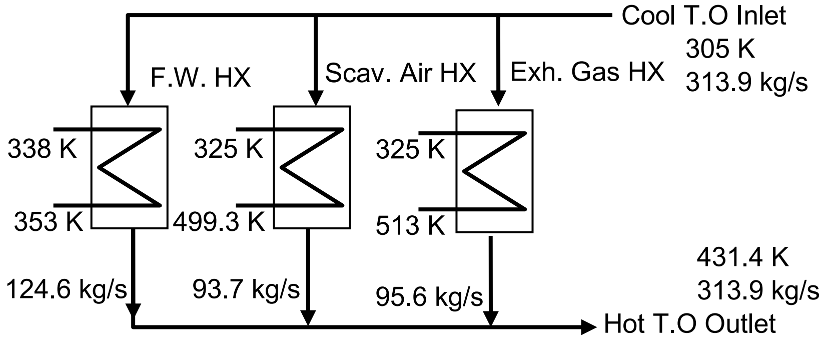

Fig. 5.6 Parallel separated marine heat recovery system configuration using heat from S90 – 69.72 MW [11].

5.2.4 Analysis of results

Fig. 5.7 illustrates these different temperatures and suggests that in most cases, heat is wasted between 370 K and 500 K, which is in average at 435.8 K. This fact will be recalled in the simulation of the projected machine and the power architectures.

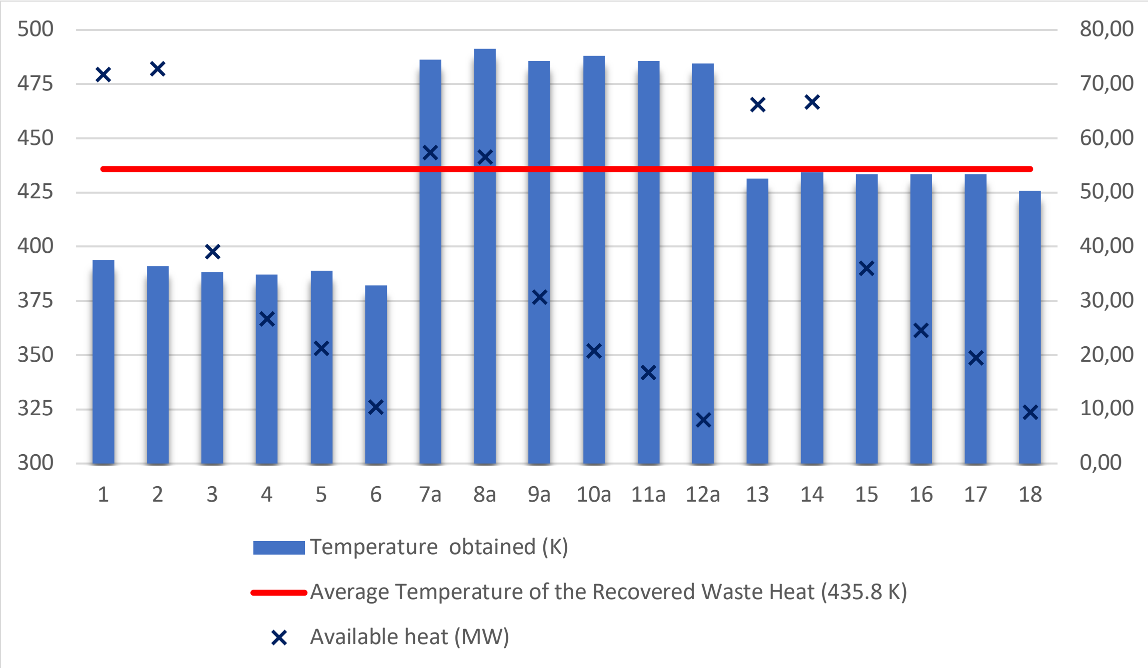

Fig. 5.7 Temperatures of the heat wasted by the two-strokes engines reviewed.

The machine provided is not as influenced as others by temperature. However, there are heat exchange processes, which are directly affected by temperature difference. Thus, the WHRS to be considered are those whose temperatures are higher, which according to Fig. 5.7 are those not including heat from lubrication oil. The inclusion of this heat reduces considerably the temperature of the heat recovered.

The next section introduces the machine using the C4L cycle to convert energy. Where WHRS based on C4L cycles coupled to the MAN® S90 – 69.72 MW [11] will be analysed and compared. This Engine will be referred as MAN® S90 [11] in further sections.

5.3 C4L Power Unit

The machine we suggest uses basic components. It is designed to work with all kinds of low-grade residual heat. Thus, there is no need to implement complex systems, such as, high pressure injectors for combustion systems. Power units (PU) constitute a materialisation of the C4L cycles. PU recreates closely the thermodynamic path that characterise the C4L.

The potential of PU architectures is analysed using WH from a commercial industrial Diesel engine. The operation of the power unit will be described in this section. Finally, the parameter for evaluating and comparing the different architectures based on C4L Power units is the thermal efficiency. The thermal efficiency is used to assess the HUF of a marine power plant. This variable indicates the amount of energy consumed and wasted in a power plant according to the description in section 2.4. Results show that the implementation of C4L cycles can improve considerably the amount of energy converted in a power plant. This is due to its ability to transform heat at low temperature, which is the main barrier of the available technology.

5.3.1 Components of a power unit

The machine is a on the invention detailed in Section 4.3.1. The main components are: a double effect piston four heat exchangers, fans and valves. Tab. 5.6 depicts a detailed description of the main components of the single power unit. The device which turns the thermal energy of the TF into mechanical energy is a double-acting cylinder. The machine we have projected works with a double-effect piston in order to optimise space and performance of heat exchangers. Fig. 5.8 illustrates the main components of the machine. The machine planned can be coupled to patented devices able to change a linear alternative movement into a continuous rotary movement (See P [1] and P [2]).

Tab. 5.6 Detailed description of the main components of the proposed machine.

Component	Description
F (F1, F2, F3, F4)	Circulation fan. It is responsible for the recirculation of the WF inside the heat exchanger, improving heat transfer effectiveness.
FT	Transfer fan. It is responsible for the transfer of the WF from the cool side to the hot side
HX-(20) and HX-(21)	Heat exchanger, that heats the WF.
HX-(30) and HX-(31)	Heat exchanger, that cools the WF.
Double-acting piston cylinder (1-2)	It is responsible for converting the thermal energy into both: hot energy for doing mechanical work by means of adiabatic expansion and cold energy for doing work by means of adiabatic contraction-based compression.
14, 15	Inlet (14) and outlet (15) hot TF.
16, 17	Inlet (16) and outlet (17) cooling water.
22, 23, 32, 33	Piston control valves.
13	Transfer valves.

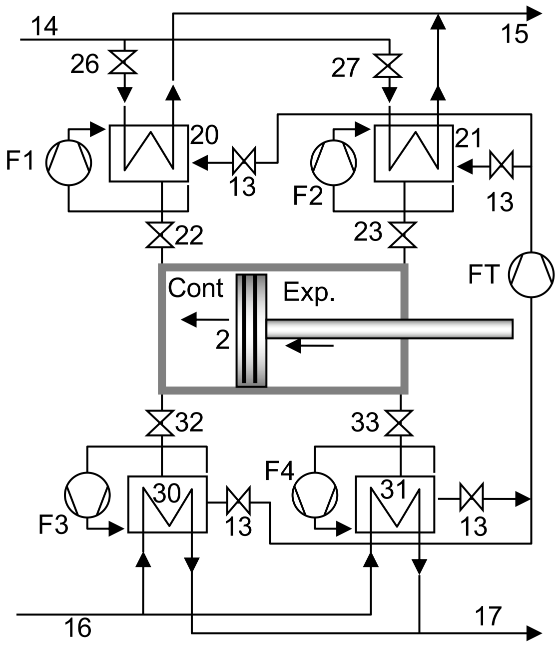

Fig. 5.8 Basic system which cool and heat the WF of a piston-cylinder unit. Tab. 5.6 details the main components of the machine.

5.3.2 Operation of a power unit based on a C4L cycle

The cycle of the proposed machine is made up by 2 different steps, which are ruled by the control valves. Neglecting the initial condition, Step 1 and 2 are the two possible states of the machine. After Step 2, the mechanical cycle of the machine ends and another one begins following the same two steps. This architecture needs a valve (13), which balances the mass of WF in the cool and hot heat exchangers. The system is always slightly pumping WF from the hot side to the cool side. This is due to the contraction in the cool side assisted with the expansion of the hot side, which pumps the WF to the cool heat exchangers (20 et 21). The main advantage of this system is the low thermal inertia in heat exchangers. For instance, if the machine worked with one heat exchanger inside a tank cooling and heating, the thermal inertia of the mass of the heat exchanger prevent the realization of the system. This obstacle is overcome with the system suggested. Fig. 5.9 and Fig. 5.10 illustrates the two Steps and Tab. 5.7 and 5.8 describe the steps.

Tab. 5.7 Operation of the proposed machine (Heat exchangers).

Step	Final Piston Position	Cylinder chamber right	Cylinder chamber left	HX - (20)	HX - (21)	HX - (30)	HX - (31)
0	N/A	N/A	N/A	N/A	N/A	N/A	N/A
1	Left	Adiabatic expansion	Adiabatic contraction	OFF	Heating	Cooling	OFF
2	Right	Adiabatic contraction	Adiabatic expansion	Heating	OFF	OFF	Cooling

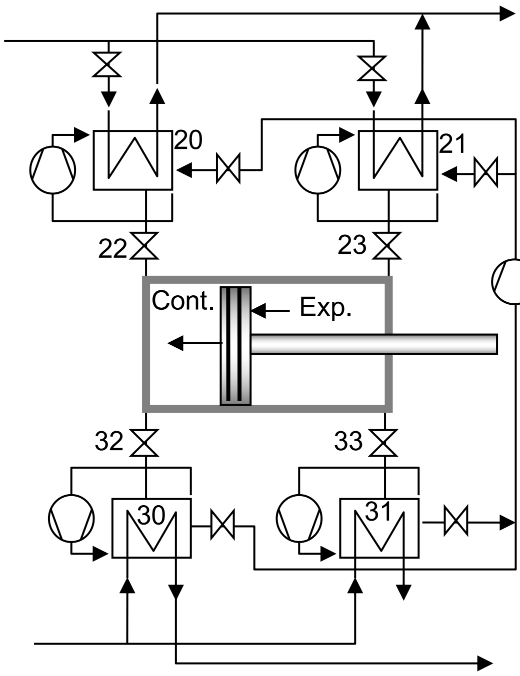

Fig. 5.9 Step 1 described in Tab. 5.7 and 5.8.

Tab. 5.8 Operation of the proposed machine (Valves).

Step	Final Piston Position	Cylinder chamber right	Cylinder chamber left	Valve (22)	Valve (23)	Valve (32)	Valve (33)
0	N/A	N/A	N/A	OFF	OFF	OFF	OFF
1	Left	Adiabatic expansion	Adiabatic contraction	OFF	ON	ON	OFF
2	Right	Adiabatic contraction	Adiabatic expansion	ON	OFF	OFF	ON

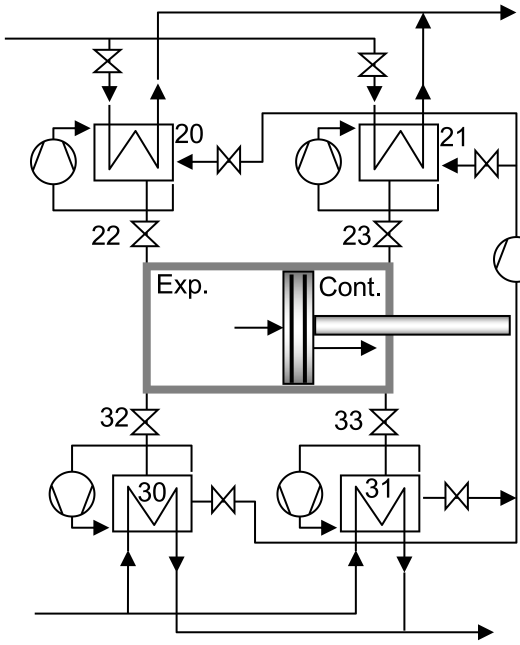

Fig. 5.10 Step 2 described in Tab. 5.7 and 5.8.

5.4 Study of different C4L architectures

PU are proposed to materialize the C4L cycle included in this thesis. The following machines depend heavily on the efficiency of the heat transfer processes. In order to simulate these machines, it is necessary to assume parameters, which assess the heat exchange behaviour in the task of heat transfer. Temperature differences will be taken as a parameter to consider heat transfer. Tab. 5.9 shows the temperatures differences assumed for the heat exchange processes of the machines.

The WFs cannot condense in the range of temperatures of the machine. The proposed machines will work with hydrogen, air and helium. These gases have very different properties. In terms of heat transfer, helium properties are considerably more favourable than air properties. In contrast, Air is widely available compared with Helium. The work of an expansion/contraction described in Section 3 is directly proportional to the specific heat at constant volume. Therefore, Hydrogen has a significant advantage in terms of WF mass or PU size.

The different properties of these fluids are the reason why they were selected. So, it can be demonstrated that the following PU can achieve a reasonable conversion ratio of energy with a wide range of different WFs. Simulations are done using an EES code derived from the Appendix 1, which takes into account losses and secondary fluids. Tab 5.9 also includes the losses of the machine.

Both case study and initial simulations were done assuming a water temperature of 300 K.

Tab. 5.9 Temperature differences assumed in the heat exchangers with different fluids assumed for the analysis of the case studies and other losses.

Type of Loss	Air	Helium and Hydrogen
Work Fluid transfer (kJ)	$\frac{dV}{v_1}\Delta h_{A-B} / \eta_f$	
Fan Efficiency η_f	0.8	0.9
Convective fans (kJ)	$m_{\substack{working \\ fluid}}\, dpv_{gas}\, v_1 / \eta_f$	
Pressure difference dpv_{gas} (kPa)	2	
Water pumps work (kJ)	$m_{\ water}\ V_{\ water}\ dpv_{water}$	
Pressure difference dpv_{water} (kPa)	0.2	
Isentropic efficiency	0.85	0.9
ΔT Thermal Oil (K)	40	15
ΔT Water (K)	35	10

5.4.1 C4L PU using Air as WF

Section 5.3.2 explains in detail the operation of the machine. Thermodynamically, this option is limited in terms of performance. The main limitations are its heat ratio and specific heats.

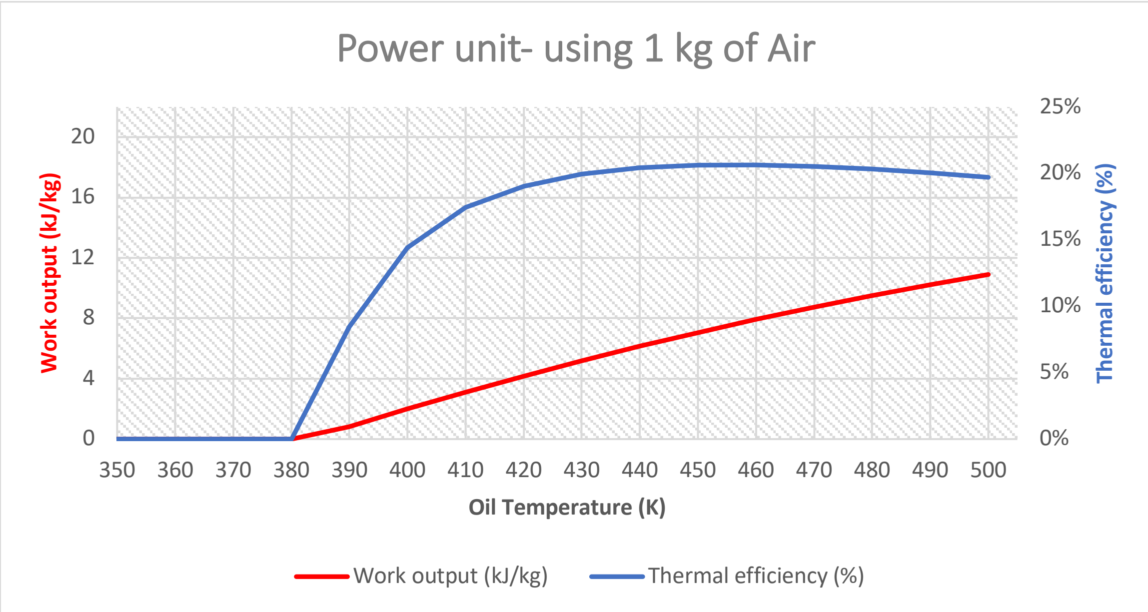

Fig. 5.11 Performances (Thermal efficiency and specific work Output) of the proposed Simple architecture using 1kg of air as WF.

Fig. 5.11 shows the thermal efficiency and the work output according to the temperature of the TF. Firstly, its efficiency increases for lower temperatures when temperature of oil inlet increases. After that, the efficiency reaches a maximum were the efficiency remains almost constant. Finally, after roughly 460 K the efficiency starts to decrease slightly. This can be explained by the indirect dependence of the machine in the temperature. However, limitations, such as, heat transfer, explain why the machine cannot convert energy for lower temperatures. In power plants where heat is required for secondary processes, such as, steam generation or heating, this system is convenient because it extracts a small amount of heat from the TF. The figure also proves an inherent problem of the concept if it is used for higher temperatures, which is its inability to convert a realistic amount of thermal energy when working with higher temperatures.

5.4.2 C4L PU using Hydrogen as WF

Fig. 5.12 illustrates a very similar result compared to air. Despite having a similar heat ratio, Hydrogen have very different properties both Cp and Cv. These important parameters improve dramatically the performance of the machine suggested, as shown in Fig. 5.12. In addition to the energy conversion performance, the high specific heats of the hydrogen require a low WF mass, in the order of 14 times less than the mass required by a machine using air as WF. The shape of the graph is similar to the results obtained with air. The scale differs considerably. In addition, its performance shows the same limitation, which is its inability to convert energy when temperature increases. In this case this occur from roughly 400 K, after this point the performance start to drop.

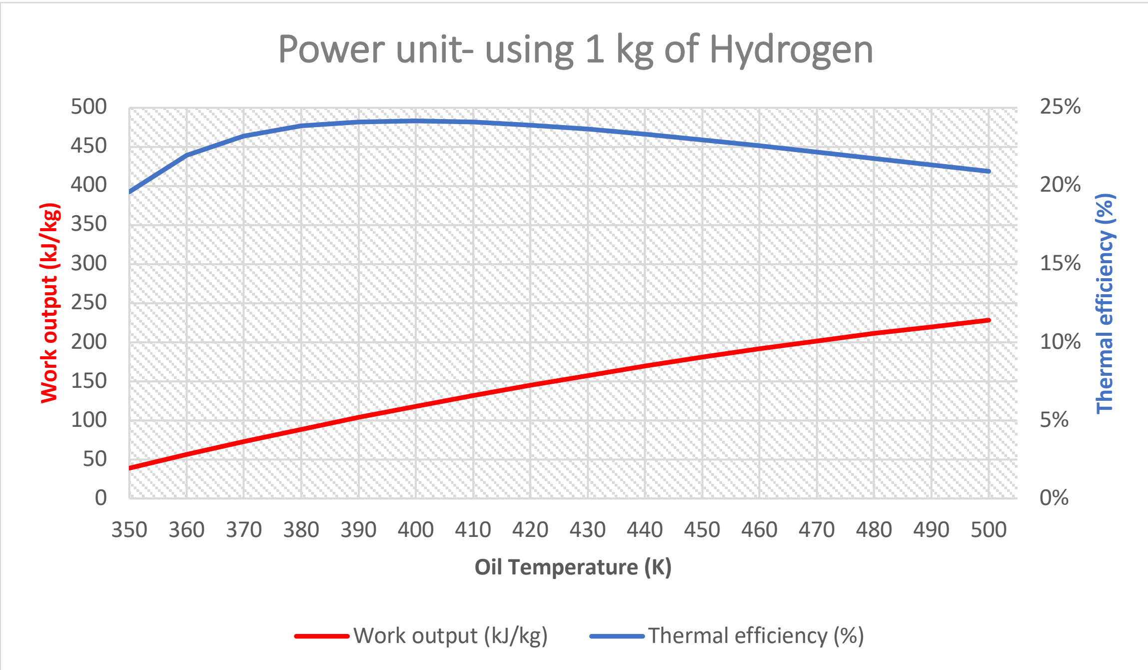

Fig. 5.12 Performances (Thermal efficiency and specific work Output) of the proposed Simple architecture using 1kg of hydrogen as WF.

5.4.3 C4L PU using Helium as WF

Fig. 5.13 illustrates that both the range of temperatures and performances increase when using a convenient WF. The first third of temperatures shows particularly interesting results. In this region, commercial WHRS are not able to convert efficiently thermal energy into mechanical power. The mechanical work output is lower than the machine using hydrogen but higher than the machine using air. The performance indicates that this machine is able to maintain its capacity to convert thermal energy even when temperatures increase. For instance, from 430 K to 500 K it losses less than 1 %. This is probably due to the better thermodynamic performance of the Helium, its heat ratio is considerably higher than the other two WFs reviewed. The work output for 1 kg of WF is lower for Helium than Hydrogen. However, it requires more than the double of thermal energy to heat 1 kg of Hydrogen than 1 kg of Helium due to the differences in specific heat. Therefore, for the same amount of heat available to convert into mechanical energy, helium will obtain better performances.

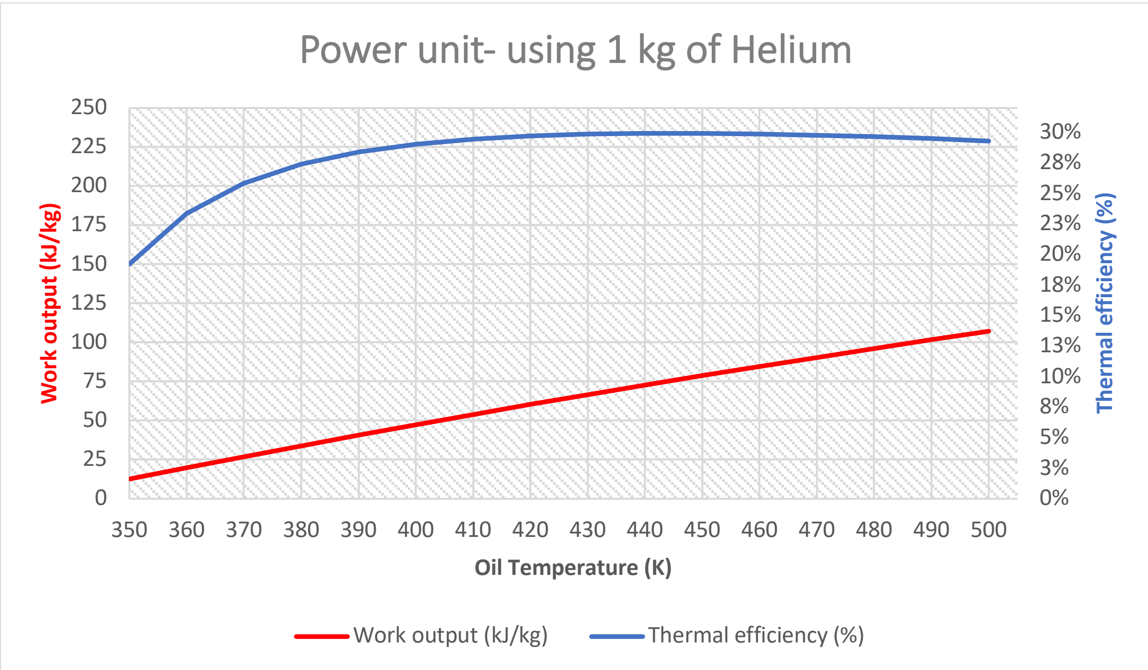

Fig. 5.13 Performances (Thermal efficiency and specific work Output) of the proposed Simple architecture using 1kg of helium as WF.

5.4.4 Comparison of the architectures with commercial systems.

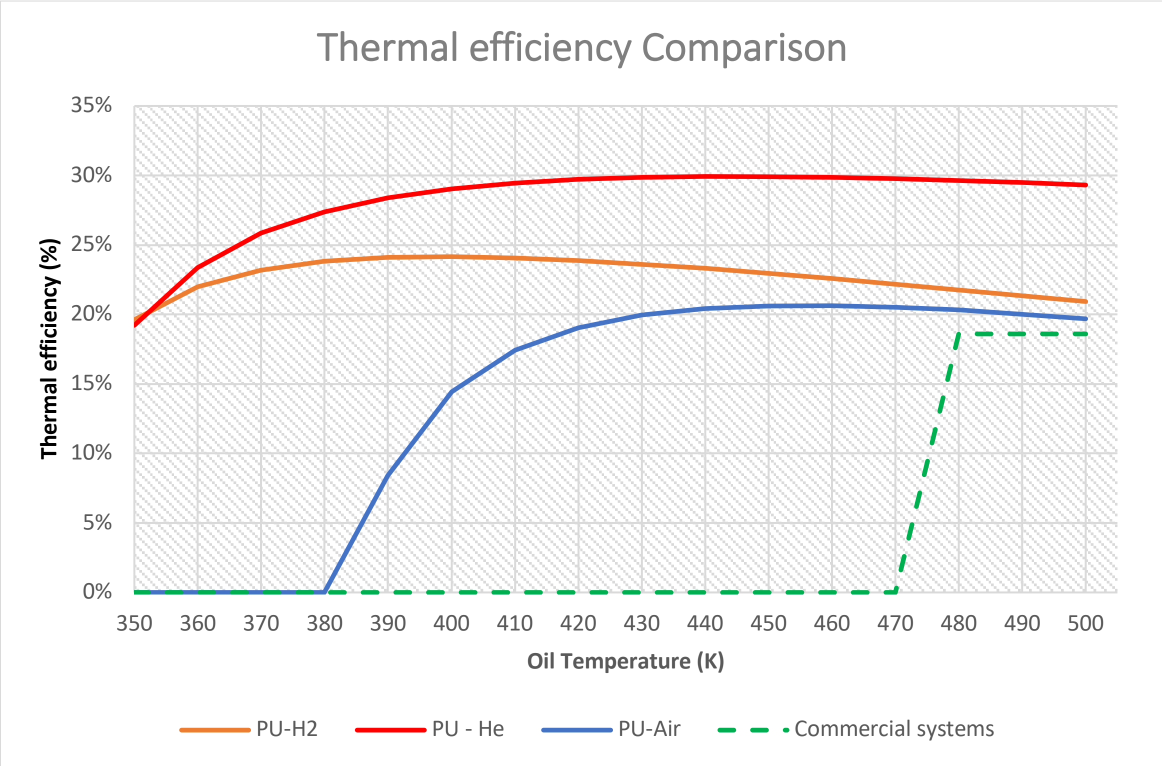

Fig. 5.14 Thermal efficiency comparison of the proposed architectures.

The comparison is done assuming that both Wärtsilä® and MAN® systems can convert the same percentage of heat into mechanical work at all the temperature range indicated. As they use mainly heat dependant on the top temperature of the scavenge and exhaust, this range of temperature was proposed. Both using Air, Hydrogen and particularly Helium outperforms these two commercial systems in all the analysed conditions. All simulated cases shown in Fig. 5.14 can offer an undisputable good performance. However, it is worth mentioning that the machine working with Air can obtain considerable better results than the commercial systems for lower temperatures using a widely available WF and neither health issues nor any sort of pollution are produced. Fig. 5.14 also illustrates that the machines suggested can use heat that current technology cannot. For instance, the heat from the jacket water, which represents an important amount of energy wasted.

There is also a technical aspect worth mentioning: the relativity of the efficiencies. Fig 5.14 show an efficiency close to 18 % for commercial systems. This value was achieved using data from Fig. 2.19 where 5.9 % is obtained from the 31.7 % of the exhaust gas. This value appears to be close to the other efficiencies in the figure. However, there is an important difference, commercial WHRS are not able to recover heat at lower temperatures, as shown in Fig. 5.14. Lower temperatures as shown in section 5.2 have an important amount of WH.

5.4.5 Case Study – Simulation of C4L machines coupled to a MAN® S90 [11]

Previous sections demonstrated that a simple power unit using C4L can outperform current technology for low grade heat conversion systems. This section simulates the power unit applied to the mentioned MAN® engine. Section 5.2 explained the sources of WH of industrial diesel engines. Current technology can extract heat mainly from exhaust gases and cannot convert thermal power efficiently from other sources. Therefore, efficiencies claimed by the manufacturers of such devices can only be applied to a small portion of the WH. Thus, even in the same range of efficiency, the output is higher when considering more WH. The main reason for considering Hydrogen in the study was to illustrate an important factor of the PU, which is the size or mass of WF required. This parameter will be given in each calculation to assess to feasibility of the PU. Mass does not translate directly in size but when comparing WFs using the same pressures in state 1 of the cycle, size is directly proportional. Thus, size depends on the mass and pressure range applied. Pressures close to atmospheric pressure are used to propose a machine that has less possibilities to leak and does not use high pressures for safety.

5.4.5.1 Case – Air simulation

This section simulates the architecture displayed in Fig 5.15 using Air as WF. The residual heat from the engine is extracted from jacket water, scavenge air and exhaust gas through a parallel recovery architecture presented in section 5.2.2. According to Tab. 5.10 this system obtains 19 % of thermal efficiency and 4.5 MW per mechanical cycle of thermal energy converted into mechanical work for this specific case. Fig. 5.15 illustrates the flows INLET and OULTET of the machine. Tab. 5.11 shows the thermodynamic states of the machine for this simulation.

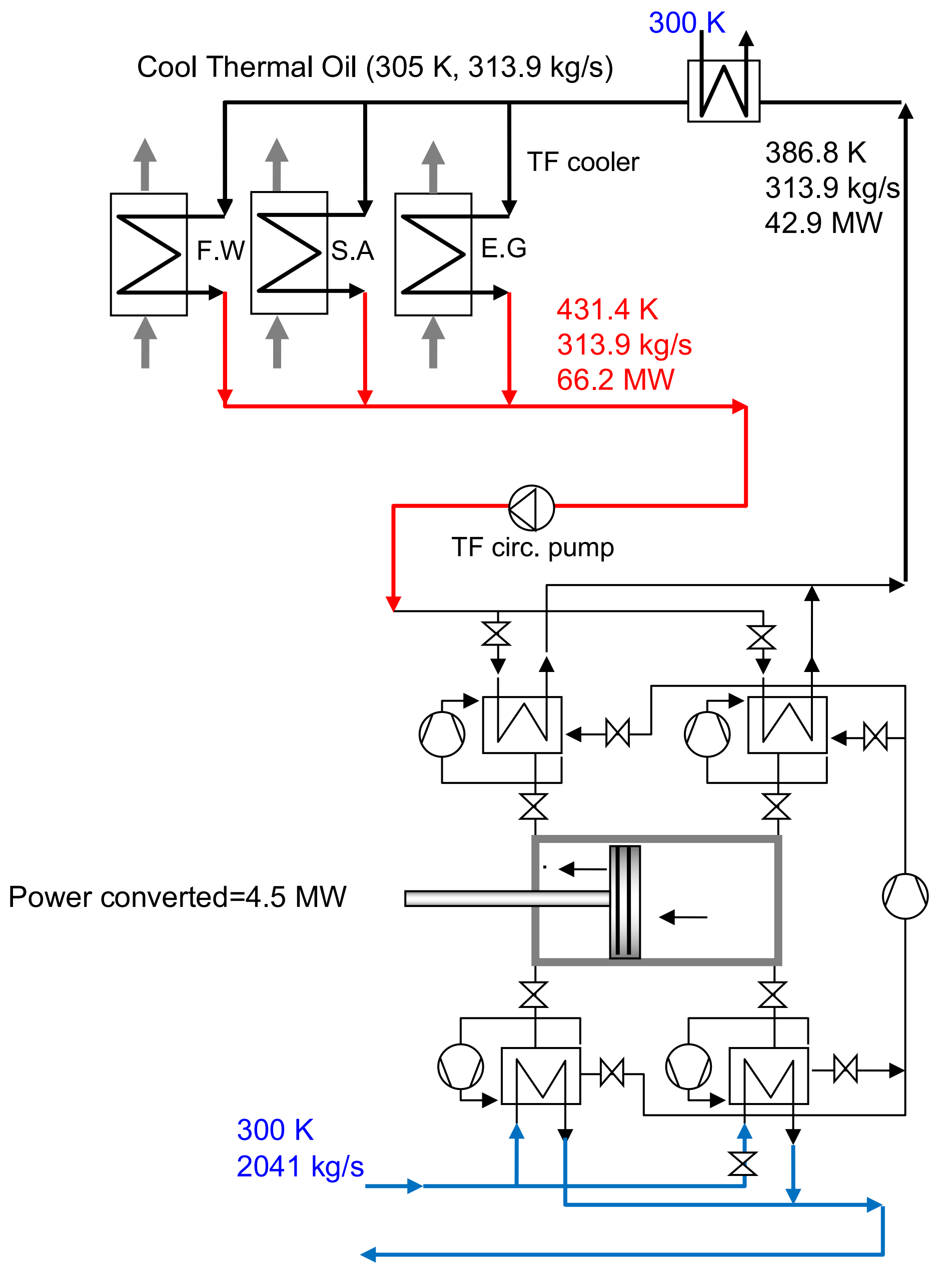

Fig. 5.15 PU architecture inlets and outlets using heat from a Parallel WHRS recovering heat from the MAN® S90 [11] working with Air as WF.

Tab. 5.10 Performances of the machine per mechanical cycle working with Air as WF.

Thermal efficiency (%)	19.0
Work Output (kJ)	4447.0
Convective Fan loss (kJ)	2583.0
Internal pumping work loss (kJ)	928.6
Mass of WF (kg)	1038.2
Efficiency in terms of shaft output (%)	6.4

Compared to the available technology the work output and the conversion potential is by far superior. (See Section 2.4) The fact that machines based on the C4L provided in this thesis are not limited in terms of temperature explains the important difference of efficiency and work output. This solution is convenient if other secondary thermal processes are required, such as steam generation.

Tab. 5.11 Temperature, pressure, internal energy, enthalpy, specific volume and entropy in the different states working with Air as WF.

State	T (K)	P (kPa)	u (kJ/kg)	h (kJ/kg)	v (m^3/kg)	s (kJ/(kg·K))
1	346.8	100.0	247.9	347.5	0.995	5.851
2	391.4	112.9	280.2	392.6	0.995	5.939
3	378.2	100.0	270.6	379.2	1.085	5.939
4	335.0	88.6	239.5	335.6	1.085	5.851

Tab. 5.11 quantifies the thermodynamic values of the power unit. Both pressures and temperatures remain low compared to other WHRS. Therefore, this fact makes the use of less exigent materials for the components a real outcome. The inlet temperature is low.

5.4.5.2 Case – He simulation

This section simulates the architecture using Helium as WF. The simulation obtains 29.7 % of thermal efficiency and 12.3 MW per mechanical cycle of thermal energy converted into mechanical work for this specific case (See Tab 5.12). Fig. 5.16 illustrates the diagram and the flows INLET and OULTET of the machine. Thermodynamic states of the cycles shown in Tab

5.13 demonstrates that this system can work with lower pressures and attain higher efficiency than the commercial systems.

Tab. 5.12 Performances of the machine per mechanical cycle working with Helium as WF.

Thermal efficiency (%)	29.7
Work Output (kJ)	12311.0
Convective Fan loss (kJ)	3143.0
Internal pumping work loss (kJ)	1782.0
Mass of WF (kg)	202.0
Efficiency in terms of shaft output (%)	8.9

Tab. 5.13 Temperature, pressure, internal energy, enthalpy, specific volume and entropy in the different states working with Helium as WF.

State	T (K)	P (kPa)	u (kJ/kg)	h (kJ/kg)	v (m^3/kg)	s (kJ/(kg·K))
1	337.3	100.0	-497.3	203.2	7.005	32.18
2	416.4	123.5	-250.8	614.0	7.005	32.84
3	382.8	100.0	-355.7	439.3	7.949	32.84
4	310.0	81.0	-582.3	61.5	7.949	32.18

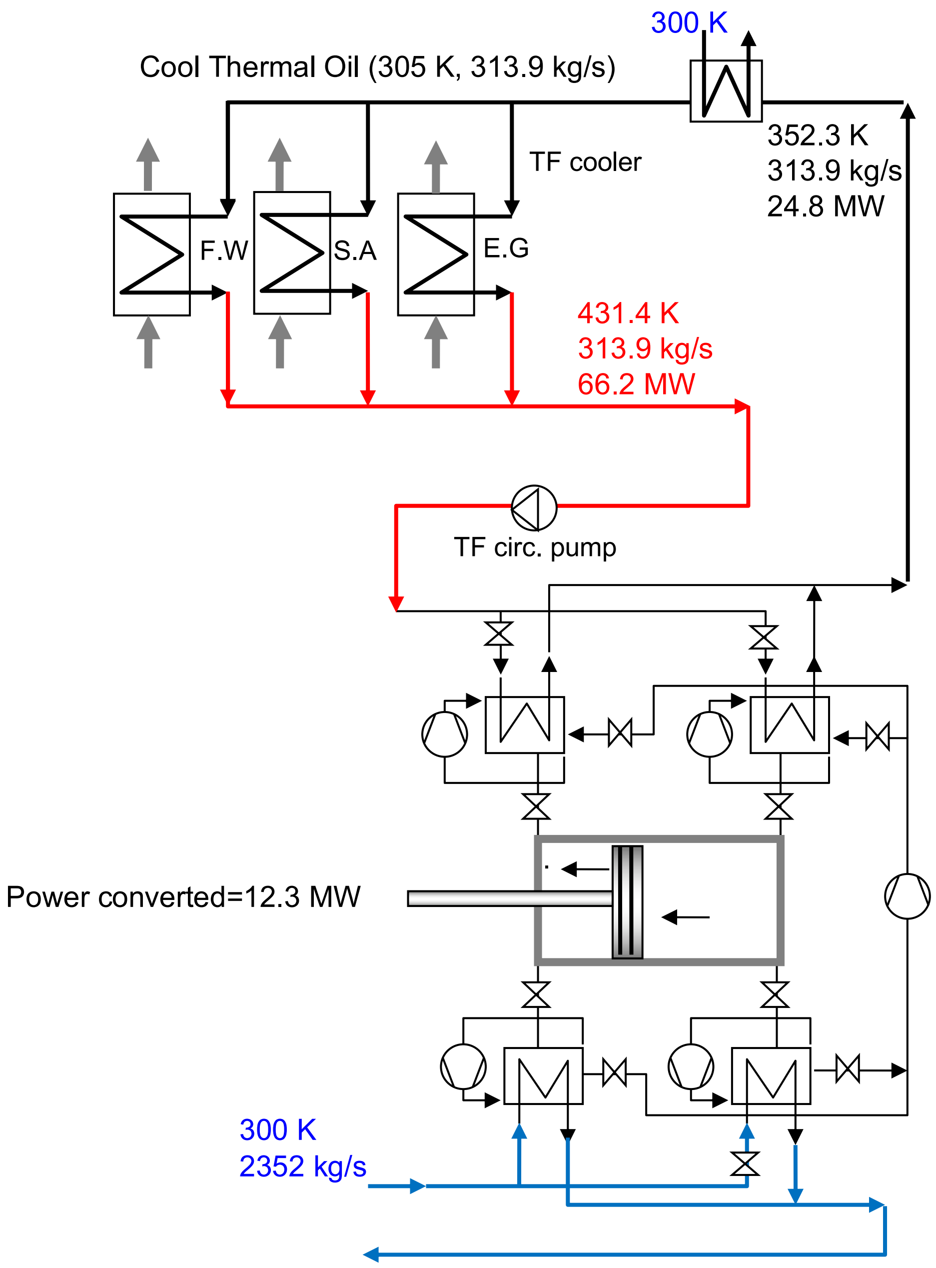

Fig. 5.16 PU architecture inlets and outlets using heat from a Parallel WHRS recovering heat from the MAN® S90 [11] working with Helium as WF.

5.4.5.3 Case – H2 simulation

This section simulates the architecture using H2 as WF as shown in Fig. 5.17. Tab. 5.14 shows better results to ones achieved with Air as a WF: 23.9 % of thermal efficiency and 10.7 MW of thermal energy converted into mechanical work for this specific case (per mechanical cycle). This architecture has the advantage of requiring less mass of WF. For applications where the space is limited, this solution is convenient. Tab. 5.15 quantifies the thermodynamic values of the power unit. Both pressures and temperatures remain low compared to other WHRS.

Tab. 5.14 Performances of the machine per mechanical cycle working with Hydrogen as WF.

Thermal efficiency (%)	23.9
Work Output (kJ)	10706.0
Convective Fan loss (kJ)	2205.0
Internal pumping work loss (kJ)	3051.0
Mass of WF (kg)	72.7
Efficiency in terms of shaft output (%)	7.7

Tab. 5.15 Temperature, pressure, internal energy, enthalpy, specific volume and entropy in the different states working with Hydrogen as WF.

State	T (K)	P (kPa)	u (kJ/kg)	h (kJ/kg)	v (m^3/kg)	s (kJ/(kg·K))
1	331.0	100.0	-892.5	472.8	13.65	66.32
2	416.4	125.8	-13.1	1704.0	13.65	68.69
3	390.0	100.0	-285.6	1323.0	16.08	68.68
4	310.0	79.5	-1108.0	170.1	16.08	66.32

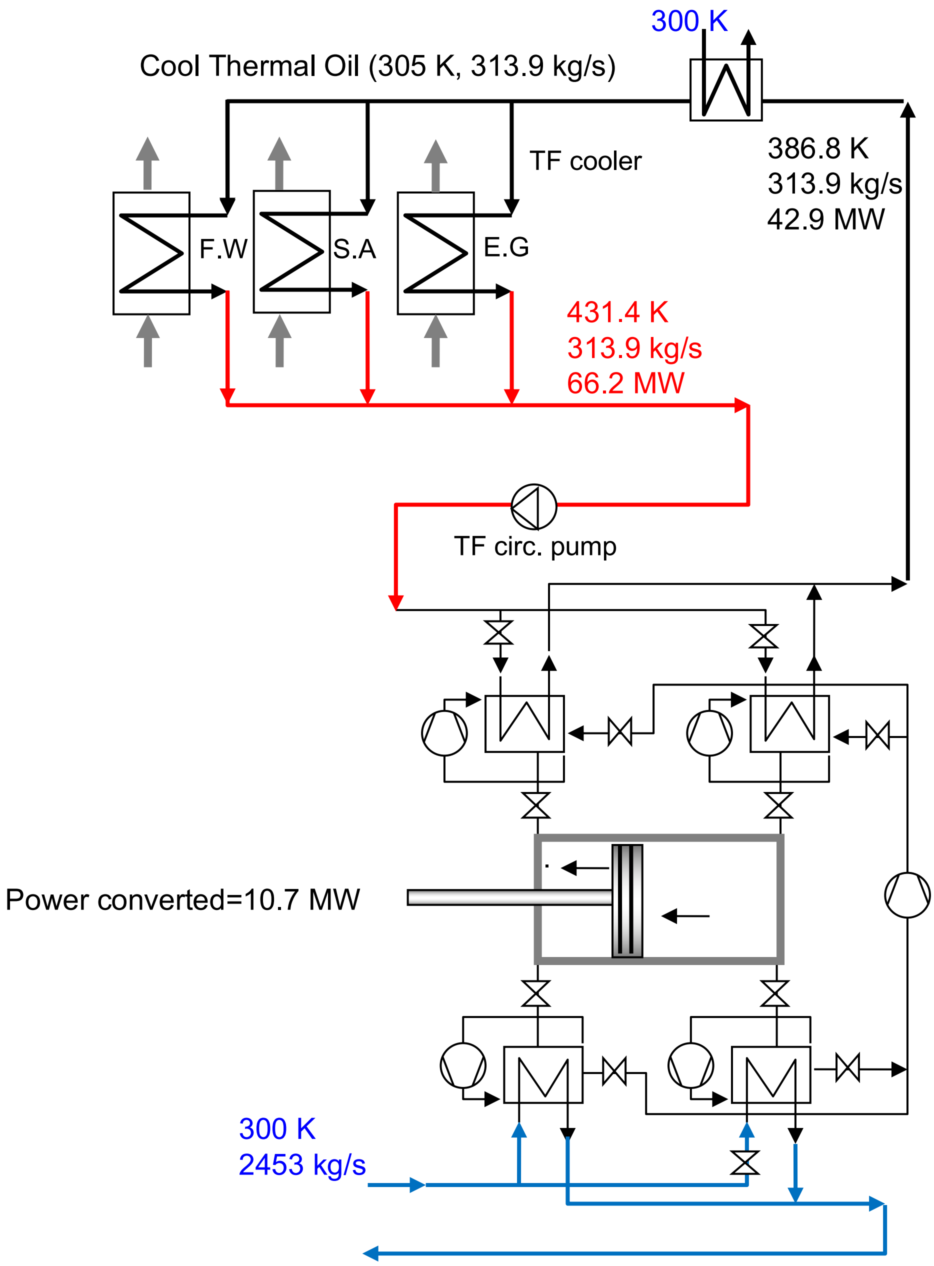

Fig. 5.17 PU Architecture inlets and outlets using heat from a Parallel WHRS recovering heat from the MAN® S90 [11] working with Hydrogen as WF.

5.5 Analysis of results

Throughout this section, three sub-sections are taken into consideration in regards with the general analysis of results, since an analysis of the state of the art is required previously to an analysis of the expected conclusions of the solution provided, for performing afterwards an analysis of the results achieved for the case studies carried out.

Analysis of the state-of-the-art technologies

As a consequence of the review of the state-of-the-art of WH recovery techniques in marine propulsion plants, it has been found that the overall efficiency without any WHRS is excessively low, of the order of 49 %, according to the data shown in Table 5.16. This, undergoes a percentage of heat losses that approaches about 51 % of the total fuel consumption. Likewise, it has been observed that marine propulsion plants equipped with the best current WHRS recently incorporated into the industry only achieve an increase in the overall efficiency of the plant by approximately 5.1-5.9 %, which is also not satisfactory in general terms based on the data shown in the third column of Tab. 5.16 obtained from the review study carried out in Chapter 2. This technology leads to an unrecoverable loss of thermal energy in the order of 45.1-45.8 % of the total energy supplied to the propulsion system.

Tab. 5.16 Current marine propulsion plants equipped with the state-of-the-art technologies concerning to the WHRS.

WHRS Installation type	Shaft power (no WHRS) %	Power gained due to the WHRS %	WH %	HUF %
Wärtsilä® 12RT flex96 [43]	49.0	5.9	45.1	54.9
MAN 12S90ME [40]	49.1	5.1	45.8	54.2

The facts exhibit an opportunity concerning to the availability of a given important amount of WH in the range of 45.1-45.8 % of the total added heat-based fuel available at different temperatures where two of them exhibit sufficient potential to be collected as power source in a WHRS which supposes roughly the 40 % of the total heat at an average temperature that approaches 500 K, therefore meaning a relevant percentage of recoverable heat thanks to the suggested technology.

From the perspective of the gain in global efficiency obtained as a result of the incorporation of conventional WHRS systems, it represents a certain increase that is not at all satisfactory given the possibilities that a change in the technological paradigm should offer related to the discovery of heat-work interactions not explored until now, and which are based on getting useful work through compression-based thermal contraction processes as a result of the extraction of heat from a TF.

The observation of the possible benefit derived from the incorporation of such heat-work interactions gives rise to a hypothesis based on the urge to develop both a thermal cycle and a machine capable of operating with such a cycle, all of which entails the necessary study object of the thesis, aimed at achieving results that provide benefits to heat recovery systems in general and particularly to residual heat recovery systems in marine propulsion plants.

Analysis of results achieved from the ideal C4L cycle computation

In chapter 3 the C4L in single acting mode with air as WF has been studied. The conclusions of the analysis of the C4L in single acting mode with dry air as working have been presented in Table 3.8. Subsequently, researches have been carried out on the same reversible cycle operating with WFs such as hydrogen and helium.

In order to develop a performance analysis, a WHRS equipped with the C4L cycle is implemented in a combined structure composed of a power source consisting of the residual heat from a marine reciprocating internal combustion engine where a p-V diagram to show the C4L cycle is depicted. This structure represented for the cycle analysis purposes is depicted in Fig. 3.9.

The reasons for the selection of such WFs is due to the fact that hydrogen is superior to other WFs for providing a high specific work, while helium is superior to other WFs for easing a high performance, independent of the fact that both hydrogen and helium exhibit other advantageous characteristics as WFs. The data presented in Table 3.8 for air as WF has been extended to hydrogen and helium and their operational parameters in the C4L cycle. Now, in order to perform a general analysis concerning to the achieved results, the data corresponding to the ideal C4L cycle state points for dry air, hydrogen and helium is depicted. In the Table 5.17, the set of parameters for each WF is shown, including temperature, pressure, specific volume, specific internal energy, specific entropy, specific heat capacity at constant volume and specific heat capacity at constant pressure. The data contained in the table 5.17 expresses the characteristic values (WF conditions) of the cycle for each state point and its correspondent WF. It is observed that fixing the top (T_H) and bottom (T_L) temperatures in 700 K and 300 K respectively for the three chosen WFs, the rest of the parameters exhibit very different values according to the inherent characteristics. Thus, the results that can be expected from such cycle data once the data analysis has been performed are very different. Such differences exhibit a lot of information useful to design the engine structure in accordance to the requirements of every fluid.

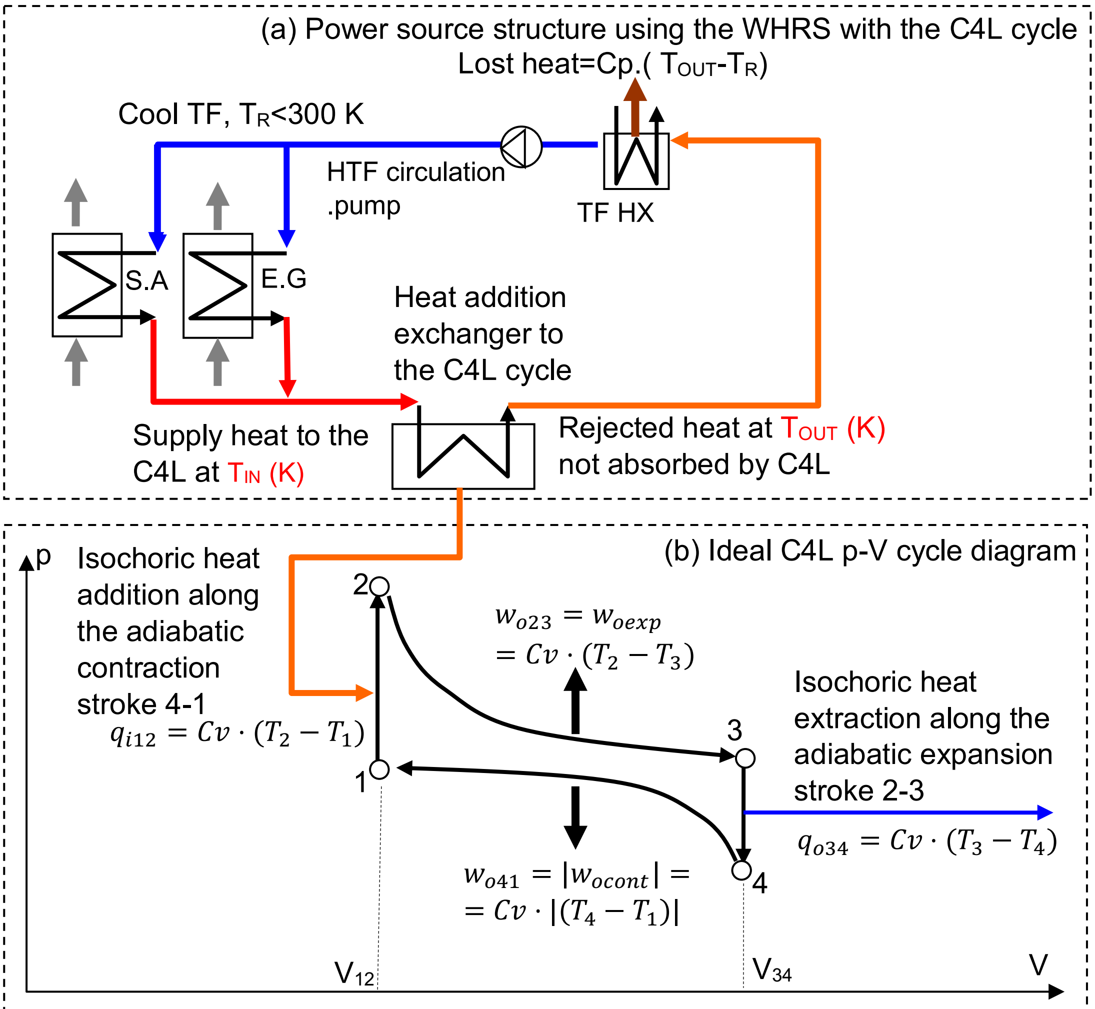

Fig. 5.18 Combined structures composed by a power source consisting of the residual heat from a marine reciprocating internal combustion engine and thermal cycle C4L under the p-V diagram. (a), the power source based on the residual heat of the marine engine. (b), the p-V diagram of the thermal cycle depicting the ideal heat-work interactions carried out.

The C4L thermal cycle operating with dry air, hydrogen and helium coincide in the extreme temperatures (high and low) as well as the cycle start pressures since the high pressure is given by the value of the high temperature as shown in Tab. 5.17. These conditions undergo certain design constraints concerning to the dimensional structure of each machine associated with the characteristics of each WF.

Therefore, using the data of Tab. 5.17 for each WF adequately, a wide set of performance parameters are quantitatively determined. Most of them are represented in Tab. 5.18 and can characterize the thermal cycle in terms of performance. The heat-work interactions carried in the C4L cycle are modelled in the p-V diagram depicted in Fig 5.18 under the computational thermodynamic models. Once the input and output heats and works are known using data from Tab. 5.18, a set of characteristic parameters are achieved with information from Tables 5.17.

Tab. 5.17 Data as cycle parameters achieved from the C4L computing task, which is required to perform the analysis of the performance cycle results.

State	T (K)	p (kPa)	v (m^3/ kg)	u (kJ/ kg)	s (kJ/ kg-K)	Cv (kJ/kg-K)	Cp (kJ/kg-K)
			WF: **Dry Air**				
1	360.9	100.0	1.036	258.1	5.896	1.009	0.722
2	700.0	194.0	1.036	512.7	6.380	1.075	0.787
3	581.9	100.0	1.670	421.3	6.380	1.047	0.759
4	300.0	51.6	1.670	214.3	5.896	1.005	0.717
			WF: **H_2**				
1	362.1	100.0	14.93	-573.2	67.63	14.42	362.1
2	700.0	193.3	14.93	2928.0	74.42	14.60	700.0
3	580.0	100.0	23.92	1678.0	74.42	14.50	580.0
4	300.0	51.7	23.92	-1211.0	67.63	14.35	300.0
			WF: **He**				
1	382.2	100.0	7.938	-357.5	32.83	5.193	3.116
2	700.0	183.2	7.938	632.7	34.71	5.193	3.116
3	549.5	100.0	11.41	163.8	34.71	5.193	3.116
4	300.0	54.6	11.41	-613.5	32.83	5.193	3.116

Using the data from Table 5.17, which is achieved from the analysis of the C4L cycle for ideal air, hydrogen, and helium as WFs, the results shown in Tab. 5.18 have been obtained, where we can observe the results of the ideal (reversible) C4L operating with dry air, hydrogen and helium as ideal gas WFs.

Tab. 5.18 The results of the ideal (reversible) C4L operating with dry air, hydrogen and helium as real gas WFs.

Process/parameter descriptions	**Process**	**Notation**	**Dry Air**	**H_2**	**He**	**Units**
Added heat to the cycle Δu_{21}	$Cv(T_2\text{-}T_1)$	q_{i12}	254.5	3502	990.2	kJ/kg
Extracted heat from the cycle Δu_{34}	$Cv(T_3\text{-}T_4)$	q_{o34}	207	2889	777.3	kJ/kg
Adiabatic output expansion work Δu_{23}	$Cv(T_2\text{-}T_3)$	w_{o_exp23}	91.34	1240	468.9	kJ/kg
Adiabatic output contraction work Δu_{41}	$Cv(T_4\text{-}T_1)$	$\lvert w_{o\_cont41}\rvert$	43.81	637.5	256	kJ/kg
Net output work $w_{o\_exp23}+\lvert w_{o\_cont41}\rvert$	$Cv(\Delta T_{23}+\Delta T_{41})$	w_n	135.1	1888	724.9	kJ/kg
Thermal efficiency. $(w_{o\_exp23}+\lvert w_{o\_cont41}\rvert)/q_{i12}$	w_n/q_{i12}	η_{th}	53.10	53.92	73.21	%
Specific stroke volume		ΔV	0.6344	9.9877	3.4753	m^3/kg
Ratio work/stroke vol.	$w_n/\Delta V$	$(w_n\text{-}V)_R$	213	210.1	208.6	$kJ/kg/m^3$
Adiabatic exp. conv. ratio (w_{exp}/q_{i12})	$(T_2\text{-}T_3)/(T_2\text{-}T_1)$	$\Delta T_{23}/\ \Delta T_{21}$	0.3589	0.3571	0.4735	
Adiabatic contr. conv. ratio (w_{cont}/q_{34})	$(T_4\text{-}T_1)/(T_3\text{-}T_4)$	$\Delta T_{41}/\ \Delta T_{34}$	0.2116	0.2207	0.3293	
Work out ratio $w_{o\_exp23}/\lvert w_{o\_cont41}\rvert$	$(T_2\text{-}T_3)/(T_4\text{-}T_1)$	w_R	2.085	1.961	1.832	
Cycle pressure ratio p_2/p_1	p_2/p_1	p_R	1.941	1.933	1.832	
High cycle temperature	$T_2=T_H$	T_H	700	700	700	K
Low cycle temperature	$T_4= T_L$	T_L	300	300	300	K

From the data achieved by means of the cycle computation and subsequent analysis, which is depicted in Tab. 5.18, some useful relations regarding to the performance behaviour are discussed:

1. Heat added to the cycle q_i. It is equivalent to the amount of heat absorbed by the WF. The amount of heat q_i added to the cycle under a closed isochoric process is proportional to the increment of temperatures and its specific heat capacity at constant volume.
2. Heat extracted from the cycle. The fact of adding heat involves the extraction of another amount of heat q_o sufficient to balance or equalize the low-temperature entropy T_4 to the entropy of the beginning of the cycle T_1. Consists of an amount of extracted heat q_o, from the cycle as the amount of heat extracted by cooling the WF. Likewise, the amount of heat to be extracted depends on the temperature difference between the one the end of the expansion T_3 and the one the end of the isochoric cooling process T_4, which depends on the entropy s_1 of the beginning of the cycle, as well as its specific heat capacity al constant volume. Since the top temperatures for both WFs is the same, which is fixed at 700 K, then the differences in added and extracted heats with respect to the used WFs is only due to the specific heat capacity at constant volume, that is, in the case of adding heat for air, hydrogen and helium approaches 254.5 kJ/kg, 3502 kJ/kg, 990.2 kJ/kg respectively. Likewise, in the case of extracting heat for air, hydrogen and helium respectively approaches 207 kJ/kg, 2889 kJ/kg and 777.3 kJ/kg.
3. Adiabatic output expansion work w_{oexp}. It is assumed as the output useful work delivered by an adiabatic expansion of the WF, which undergoes the increment of internal energy achieved by previous isochoric heat addition to the WF. Although the temperature increments due to adding heat is the same for both WFs, the amount of output expansion work is different for each WF since it depends on the specific heat capacity at constant volume and on the pressure with regards to the reference pressure. Thus, the output expansion work for air, hydrogen and helium respectively approaches 91.34 kJ/kg, 1240 kJ/kg and 468.9 kJ/kg.
4. Adiabatic output contraction work $w_{o\ cont}$. It is assumed as the output useful work delivered by adiabatic contraction of the WF, achieved by a previous process of heat

extraction from the WF. The output contraction-based work $w_{o\ cont}$ obeys a special heat-work interaction characterized by doing output mechanical work while the internal energy of the contracted WF increases. This rare physical fact responds to the first law in terms of energy balance since the previous isochoric process of extracted heat causes the adiabatic contraction of the WF while increasing its internal energy to keep the energy balance. Thus, the contraction-based work $w_{o\ cont}$ for the air, hydrogen and helium approaches 43.81 kJ/kg, 637.5 kJ/kg and 256 kJ/kg. It means an important amount of useful work without energetic cost.

5. Net output work w_n, as the output useful works due to adiabatic output expansion work plus adiabatic output contraction work. Although the thermal cycles assume as the net work the total output and input work which includes the work done on the system to drive the working feed compressor, in these preliminary studies such feed compressor work has not been included, so that it will be taken into account. Nevertheless, neglecting the work done on the system to drive the working feed compressor, the net work will approach respectively for air, hydrogen and helium 135.1 kJ/kg, 1888 kJ/kg and 724.9 kJ/kg.
6. Thermal efficiency η_{th}, as the ratio of the net output work w_n, to the added heat q_i, considering a reversible cycle. Due to the irrelevance of the feed compressor work, this thermal efficiency exhibits a higher value than that resulting from the fact of including the feed compressor work. Thus, without considering such restriction, the thermal efficiency reaches respectively for air, hydrogen and helium 53.10 %, 53.92 % and 73.21 %. As can be observed a great difference between WFs exists, so that this characteristic is one of the most relevant as criteria to select them including disadvantages (cost, availability, hazardless, maintenance, consumption due to fluid leaks) among the most important drawbacks.
7. Specific stroke volume. It is assumed as the specific change of volume along the piston stroke displacement (the change of volume of a kg of the WF) into the corresponding cylinder chamber due to expansion and contraction. This data is interesting from the design perspectives. Thus, in terms of design purposes for the specific cylinder dimensions of each cylinder should consider the values respectively for air, hydrogen and helium 0.6344 m^3/kg, 9.9877 m^3/kg and 3.4753 m^3/kg.

8. Ratio (work/stroke vol.), as the ratio of the net output useful work to the specific stroke volume. The results of air, hydrogen and helium yields 213 kJ/kg/m^3, 210.1 kJ/kg/m^3 and 208.6 kJ/kg/m$^{3.}$ As it can be observed by these results, these quantities are very similar to one another regardless the WF type.
9. Adiabatic expansion conversion ratio, as the ratio of the output useful work achieved when expanding the WF by adding heat to the cycle, to the amount of added heat to the cycle. The results of air, hydrogen and helium yields 0.3589, 0.3571 and 0.4735.
10. Adiabatic contraction conversion ratio, as the ratio of the output useful work achieved when contracting the WF by extracting heat from to the amount of extracted heat from the cycle. The results of air, hydrogen and helium yields 0.2116, 0.2207 and 0.3293.
11. Work out ratio. It consists in the ratio of the adiabatic expansion-based work to the adiabatic contraction-based work gotten by the cycle. The results for air, hydrogen and helium are 2.085, 1.961 and 1.832. This means that for helium as WF, the adiabatic contraction-based work carried out by cooling the WF, is a bit more efficient.
12. Cycle pressure ratio. It is the ratio of the highest to the lowest pressure of the cycle. The results for air, hydrogen and helium are 1.941, 1.933 and 1.832. These results suggest to us that there is certain relation between the pressure ratio and the work ratio, so that when the pressure ratio tends to 1 (equal pressure), this implies that the work done when extracting heat tends to equal the work by adding heat. With regarding to the pressure ratio or relation and the works ratio or relation, it can be seen that both are correlated at a certain extent: this means that if the pressure relation is high, the relation between the work obtained by adding heat and the work gained by heat extraction it is also high. Therefore, to achieve a low work ratio, a low-pressure ratio is necessary, which implies that the minimum pressure ratio is the unit with which both pressures (high and low) are equal. The consequence is that a desired work ratio is obtained when the pressure ratio tends to unity, which means that the work obtained by heat extraction is approximately equal to the work obtained by adding heat. This is equivalent to obtaining half the mechanical work by heat extraction, which implies that it has no energy cost.

13. The C4L cycle is not sensitive to variations in extreme cycle temperatures (high T_2 and low T_4). So, considering the Carnot factor as a benchmark of thermal efficiency, it turns out that at low differences between extreme temperatures (T_2- T_4) the Carnot factor is low while the reversible C4L cycle can operate with acceptable efficiency. However, taking into account the effects of real irreversibilities the performance drops dramatically to unacceptable levels.

Analysis of the results derived from the implementation of WHRS by means of case studies

In order to situate the analysis of the results concerning the study of cases that allow us to conclude the validity of the solutions given in an appropriate context, two structures of both conventional and proposed heat recovery systems are shown in Fig. 5.19 in order of technological capabilities. Therefore Fig. 5.19 depicts a comparison of heat balances MAN® 12S90ME-C9.2. [40]. (a), without heat recovery. (b), with the commercial WHRS plant showing the 5.9 % gain in HUF.

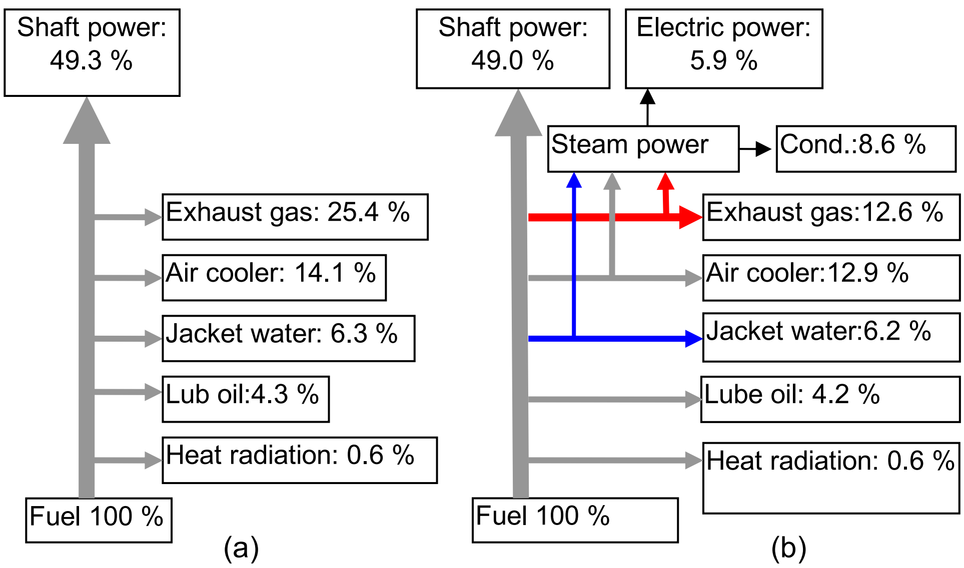

Fig. 5.19 Comparison of heat balances for balances MAN® 12S90ME-C9.2. [40]. (a), without WHRS. (b), with the conventional WHRS showing the 5.9 % gain in HUF.

The next figures (See Fig. 5.20-22) asses the performance of the proposed machine for the three WFs used previously. Comparing to the performances with the commercial one shown in Fig. 5.19, all the three methods outperform the commercial system. However, air, which has inherent low performances for this purpose, almost equals the Rankine power plant proposed by MAN® (See Fig. 5.22). In contrast, the two other WFs can achieve performances considerably higher, between 16.2 % (9 % of heating-based power + 7.2 % cooling-based power) and 18.3 % (10.2 % of heating-based power + 8.1 % cooling-based power) of HUF increase as shown in Fig 5.20 and 5.21. These values differ considerably from the 5.9 % obtained by the commercial system in Fig 5.19.

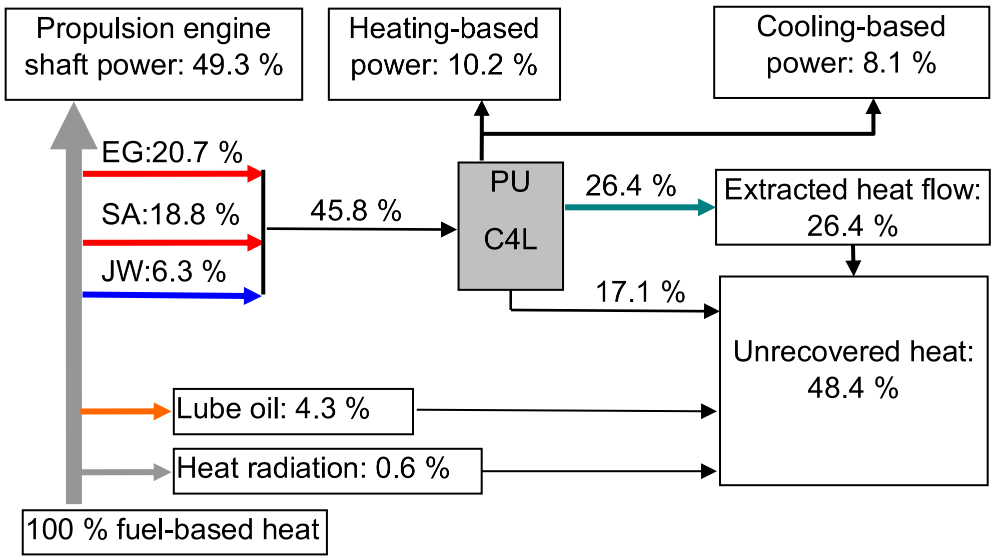

Fig. 5.20 Heat balances for balances MAN® 12S90ME-C9.2. [40] using Helium as WF. The HUF obtained is 18.3 %.

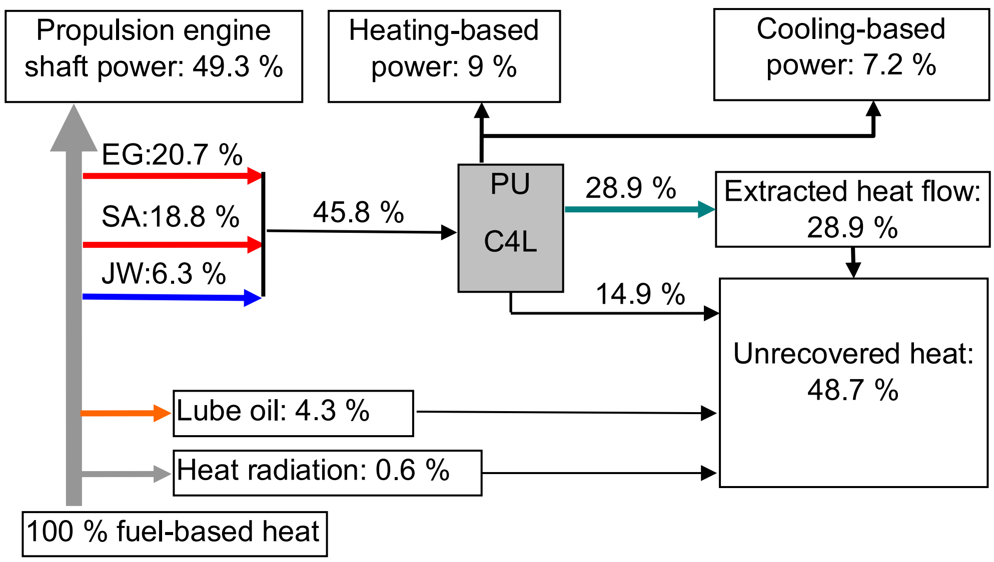

Fig. 5.21 Heat balances for balances MAN® 12S90ME-C9.2. [40] using Hydrogen as WF. The HUF obtained is 16.2 %.

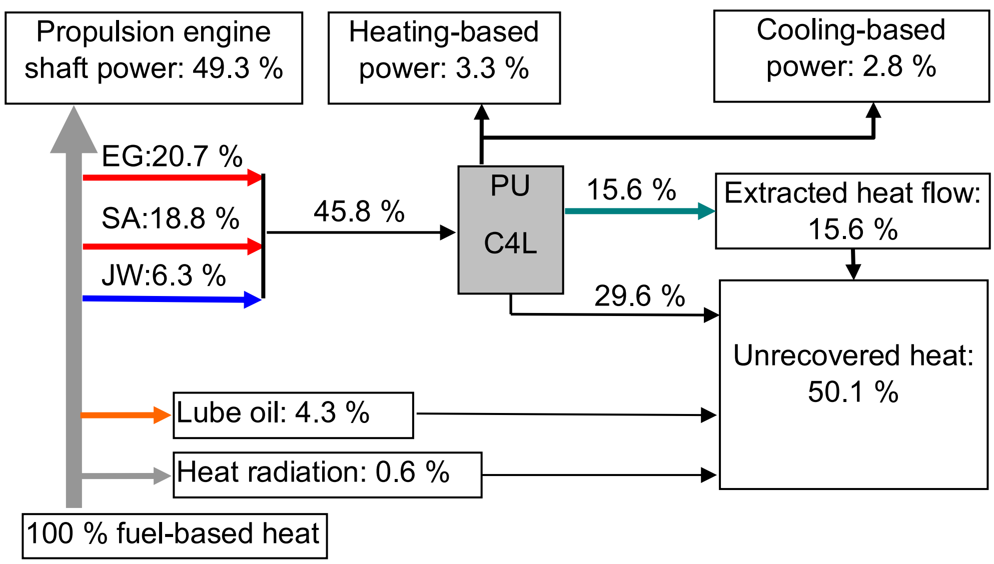

Fig. 5.22 Heat balances for balances MAN® 12S90ME-C9.2. [40] using Air as WF. The HUF obtained is 6.1 %.

In the energy flow diagram of Fig. 5.20-22 in which the energy balance of a power unit equipped with the C4L cycle is described, it can be seen that a fraction of the useful work has been obtained based on the use of heat from cooling of the WF by means of heat extraction at zero cost.

This fact means that the sum of the engine output, the work obtained through the addition of heat, the work gained through the extraction of heat and the wasted heat (the heat not recovered) rejected to the environment is greater than 100 % of the added heat. The difference corresponds to the unique characteristics in the operation of this type of cycle. As illustrated in Fig. 5.20-22, the outcome of the machine is not dictated directly by the heat input and the heat output. Thus, the subtraction of the heat input and output does not give the work output in cycles that combine both expansion and contraction. This is the reason why a flow diagram including machines that involve contraction-expansion interactions cannot be directly analysed using classical principles, such as, the logic behind Carnot-Rankine-Stirling-Ericson cycles. In these cycles, work is performed using exclusively expansions. Therefore, their output will always be dictated by the difference of heat input and output. When dealing with cycles with both contractions and expansions the net work will be the positive sum of the interactions that deliver work, and their performance can be assessed dividing the net work by the heat input.

The common characteristic of the three architectures is their amount of heat wasted when extracting heat to deliver work is considerable. However as seen in Fig. 3.2 heat is degraded and it tends to become not possible to be uses or converted due to its degradation or temperature drop.

Summary of the analysis of results derived from the implementation of WHRS by means of case studies

As a consequence of the analysis of results carried out on the study of heat recovery cases using power units characterized by doing useful mechanical work as a result of the extraction of heat from the WF (air, hydrogen, and helium), it is worth noting those results concerning the residual heat recovery factor achieved from the available residual heat before being released to the environment. In Table 5.19 there are shown five propulsion plants that use the same 2 Strokes industrial engine, where the first one doesn't include any WHRS, so that its shaft output power (PSP) approaches a 49.3 % of the total added heat energy, while for the four cases that include a WHRS its output PSP approaches 49.1 % of the total added heat energy, as observed in the second row of the Table 5.17.

With reference to the recovered heat it is observed that the propulsion plant equipped with a single propulsion engine without any WHRS its HUF approaches only 49,3 % which implies an amount of WH that approaches 51.7 %. Over 50 % of the available heat of the propulsion system is lost.

Table 5.19 Comparison results of the performance analysis. PE, (propulsion engine). PSP, (propulsion shaft power). RC, (RC as WHRS). WH, (WH). RH, (heat recovered). HWF, (heat-work flow) HUF, (heat utilization factor).

Parametric subject	Propulsion engine (PE), no WHRS	PE+WHRS (RC)	PE+WHRS: (PU-Air)	PE+WHRS: (PU-H_2)	PE+WHRS: (PU-He)
PSP (%)	49.3	49.1	49.3	49.3	49.3
WH (%)	51.7	45.8	51.7	51.7	51.7
HR (%)	0	5.1	6.1	16.2	18.3
HWF (%)	100	100	105.5	114.2	116.0
HUF %	**49.3**	**54.2**	**55.4**	**65.5**	**67.6**

Thus, the existing state of the art technologies based on RC contribute to the heat recovery in about 5.1 % which supposes a HUF of 54.2 %. However, the remaining three propulsion plants are equipped with innovative heat recovery systems, whereby the heat utilization factor is significantly increased.

Therefore, the technology proposed by this thesis based on power units driven by the residual heat rejected from the propulsion engine cooling system can contribute to the WH recovery in about 6.1 %, 16.2 % 18.3 % of the total supplied heat energy, which means a HUF of 55.2 %, 65.3 %, and 67.5 % respectively for power units operating with air, hydrogen, and helium as WFs.

6 General conclusions

6.1 Conclusion

Throughout the study carried out in this thesis, it is concluded that:

1º, the hypothesis formulated on the fact that one of the multiple possible heat-work interactions, and specifically the heat-work interaction characterized by the realization of useful mechanical work as a consequence of the previous isochoric heat extraction and subsequent compression-based contraction of the WF has been validated, in such a way that the realization of useful mechanical work by extracting heat from the WF is possible and efficient.

2º, to carry out the task of doing mechanical work by means of the aforementioned heat-work interaction, an unconventional thermal cycle is required, which has been devised on the basis of an experimental qualitative proof of concept and subsequently developed and successfully tested.

3rd, to provide physical support to the developed thermal cycle, mechanisms capable of converting heat into mechanical work have been devised and based on the proofs of concept, have led to the study of cases through which the proposed thermal cycle has been validated.

Thus, throughout the thesis chapters, studies leading to the validation of results have been carried out according to the following scheme of development of the research work:

Chapter 2 reviews the existing technology and analyses the performance of the existing techniques to convert low-grade heat energy into mechanical energy and recover waste low-grade heat mainly from ships propulsion plants. The review included the following systems:

- Organic Rankine Cycle.
- RC and its diverse variants including single and dual pressure exhaust gas boilers with water as WF.
- Commercial systems based on Brayton and RC with dual pressure exhaust gas boilers operating with water.

Rankine based cycles are one of the most common thermodynamic cycles for WH recovery. Compared to the 4- closed process-based cycle established in this thesis, RC have three main disadvantages in terms of thermodynamic: Firstly, Rankine-based cycles do not have closed process-based heat transfer; secondly, they do not use an adiabatic contraction–based compression process in the cycle using cool to deliver effective work; finally its WF experiences a change of phase.

Nevertheless, Rankine based systems have some advantages compared to the presented architectures. For instance, they do not require to transfer heat to two TFs in cascade. This translates directly into considerable power losses. A Rankine power system typically demands to heat directly its WF. Thus, only one heat transfer process is urged. Most of the presented architectures require to heat a TF, which works with the heat pump and heats the machine WF,

Along the revision of the state-of-the-art technology carried out, some relevant findings conduct us towards the research path worth the aim of developing more efficient WHRS. Since the known types of WHRS applicable to marine propulsion plants are limited or restricted by the range of available temperatures. The marine propulsion plants deliver roughly 50 % of residual heat in different grades. This residual heat comes mainly from:

- The exhaust gases of the propulsion engine, which approach 500 K at turbo outlet, which represent roughly over 25 % of the primary energy using a commercial WHRS.
- The cooling heat of the scavenge air, approaching 500 K, which represents typically more than the 15 % of the primary energy using a commercial WHRS.
- The jacket cooling water, which is close to 350 K, which represents approximately 6 % of the primary energy.
- The cooling heat of the lubricating oil at temperatures of approximately 335 K, which represents roughly 3 % of the primary energy.

Using a selective strategy, from these data it is deduced that there are two categories of residual heat based on quantitative and qualitative criteria, in such a way that taking into account the residual heats of the exhaust gases captured at the outlet of the turbochargers and the scavenge air cooler outlet water, it obtains around the 40 % of primary energy as residual heat at a temperature close to 500 K, which supposes an important amount of recoverable medium-grade residual heat.

Furthermore, the rest of residual low-grade heat due to jacket water and lube oil cooling can be partially recovered as useful heat in both distilled water production plants and air-conditioned heating, among other services.

The revision of the state of the art related to low and medium grade residual heat recovery technologies deals with:

- Organic Rankine Cycle of any kind.
- RC operating with single pressure exhaust gas boiler with water, recovered from the propulsion.
- RC operating with dual pressure exhaust gases boiler with water recovered from the propulsion engines.
- Brayton cycles using a partial bypass of the exhaust gas before the inlet of the turbochargers.

However, the available mentioned residual heats from scavenge air, jacket cooling and lube oil cooling heats, despite the advances carried out during the two last decades mainly with the advantages of the dual pressure exhaust gas boilers, cannot be efficiently recovered by means of the mentioned Rankine and Brayton cycles in comparison with the closed processes based cycles characterized for doing useful work by adding and extracting heat. Thus, the evidence of experimental facts, demonstrate that a combination between Brayton and RC operating with single and dual pressure exhaust gases boiler from the propulsion engine exhaust gases and scavenge air is the most efficient technology taking the residual heat from these sources

In all reviewed ORCs they exhibit a performance lower than the techniques based in the use of the double pressure exhaust gas boiler, since the efficiency of an ORC can operate under efficiency close to 5-6 % while the combined techniques with double pressure exhaust gas boiler can approach an efficiency close to 9 %. Thus, the heat utilization factor (HUF) can be increased in more than 5 %, when combined with partial direct expansion of the exhaust gas, approaching 54 % for most of the actual conventional WHRS applied to marine power propulsion plants.

From the performance comparison between ORCs, conventional or single pressure Rankine and double pressure RC operating between the temperature ranges corresponding to the marine propulsion plants equipped with WHRSs, the fact that despite the increased cost associated with the structural complexity of the heat recovery system, the most effective technology corresponds to the Rankine dual pressure cycle, although a significant amount of WH from jacket cooling water, scavenge air cooling water and lubricating oil cooling is wasted.

With this technique, an important fraction of the residual heat available from the scavenge air, and the jacket cooler is wasted due to the lack of a sophisticated technology capable of recovering efficiently such wasted residual heats.

The fact that an important amount of residual heat from the propulsion engine is being wasted encourages us to change the paradigm associated with heat-to-work conversion by taking advantage of the heat extraction from a WF to obtain an adiabatic closed contraction-based compression process which undergoes output useful work at relative low cost. The task and pretended achievement capable of fulfilling such above requirements demands imagining, designing, and developing a physical thermal engine structure able to overcome such challenges which are treated in chapter 3, and the next chapters.

Chapter 3 introduces and analyses a thermal cycle operating with isochoric closed processes on transfer heat to / from the cycle. It is composed by 4-processes, which is used in a patented machine for the simulation of chapter 5. As described, it is the combination of two C3L, a expansion based cycle and a contraction based cycle. The contraction works produced along with the closed-processes used are the main difference compared to current technology.

This section also validates the thermodynamic cycle. It demonstrates that the cycle fulfils the entropy balance for a thermodynamic cycle. In addition, it describes an experimental validation carried out in a laboratory, which concludes that an adiabatic compression can deliver useful work.

The background associated with these achievements concerns to the following concepts: Among a set of possible heat-work interactions (HWI) described in Table 3.2, the interaction characterized by the contraction of a WF achieved by isochoric heat extraction or isochoric cooling has been demonstrated by proof of concept implemented under empirical evidence and / or experimental to be of extraordinary utility in the task of converting heat to useful mechanical work.

With this achievement, the possibility of developing thermal cycles superior to the conventional ones is based on the use of HWIs, which entails the realization of useful mechanical work at no cost. Thus, this chapter is focused on the development of a thermal cycle that operates by isochoric addition and extraction of heat. Therefore, there has been no use of the concept of heat rejection to a heat sink as in conventional cycles. Hence the difference between the added heat and the isochoric extracted heat is different from the heat added and rejected in a conventional thermal cycle.

Based on Equation 3.2, it is observed that a fraction of the net work is due to the addition of heat, while the complementary fraction of the net work is due to the extraction of heat. Hence, it follows that the heat fraction obtained by adding heat has a certain energy cost while the other working fraction achieved by heat extraction has no cost, assuming that the task of cooling the WF is energetic costless. This leads to the fact that the thermal efficiency described by equation 3.33 turns out to be higher than the thermal efficiency of any conventional thermal cycle that performs work by adding and rejecting heat.

Therefore, given the fact that performing useful mechanical work by heat extraction or cooling has no energy cost, it is worth asking whether it is possible to carry out a thermal cycle only with cooling-based HWIs. The conclusion is that throughout the chapter it is verified that the capacity to extract heat from the WF is restricted by the initial entropy of the closed isochoric process of heat addition (because entropy s4 must be equal to entropy s1). Extracting more heat to the heat sink implies to reduce the entropy s4, so that s4 < s1. Trying to achieve s4 < s1 by overcooling the WF, undergoes the impossibility of closing the cycle, which cannot happen because the process 4-1 of the 4LC must be isentropic, that is s4 = s1 as well as s2= s3. This condition limits our capacity to extract heat from the WF so that the thermal efficiency is also restricted by the condition given by s4 = s1: This means an inevitable closed isentropic HWI in this closed processes-based cycle.

Furthermore, as consequence of the ideal cycle analysis, the above deductions are stated based on the fact that the ratio of the contraction-based work to the expansion-based work denoted as $w_R = \frac{w_{o_{Contraction}}}{w_{o_{Expansion}}}$ is associated with the pressure ratio denotes as $p_R = \frac{p_2}{p_1}$ according to the following relations:

$\lim\limits_{p_R \to 1}(w_R) = 1$, and $\lim\limits_{p_R \to \infty}(w_R) = 0$

This means that there exists a pressure ratio for which the performance of the thermal cycle is from an operational perspective, acceptable or indeed may be optimum.

In order to be prepared to analyse the case studied in the next chapter (chapter 5), chapter 4 has addressed the problem of having physical structures of thermal machines capable of implementing the different versions of the thermal cycle proposed in Chapter 3, Various architectures characterized by their abilities to withstand such thermal cycles have been described.

Thus, chapter 4 shifts the focus to mechanical systems that can materialize the described thermodynamic cycle, which includes several patents. These inventions are the solution for two problems. P [1] is an engine architecture that can make use of the thermodynamic cycle. P [2] and P [3] are devices, which can convert a linear alternative movement into a continuous rotational device.

The engine architecture associated with the patent P [1] is characterised by four heat exchangers connected to a reciprocating double-acting cylinder actuator responsible for performing mechanical work. Such heat exchangers have been conceived to operate transferring heat by forced convection, needing two per cylinder chamber: one of them is a heater and the other is a cooler, and eventually must be isolated from its respective cylinder chambers to ensure the isochoric heat transfer by means of a valve timing control system.

The engine architecture associated with the patents P [2] and P [3] is characterized by the realization of the isochoric heat transfer processes by forced convection in the reciprocating double-acting cylinder chambers, responsible for performing mechanical work, and it has been conceived to operate under such a mode that while the piston remains motionless during the isochoric heat transfer process, the piston rod remains uncoupled from the mechanism responsible for converting the thermal energy into mechanical work (connecting rod-crankshaft or a mangle rack device).

Therefore, the implementation of both proposed techniques as part of the innovations carried out in the thesis requires different techniques dealing with cycle timings, operating strategies, methodologies as function of its mechanical structures, which have been described along the chapter.

Chapter 5:

Chapter 5 evaluates the implementation of architectures based on P [1]. The marine industry is used to study its implementation. Different MAN® marine Diesel engines are reviewed. Diesel engines waste most of their heat in their lubrication oil, scavenge air, exhaust and cooling water. Three recovery systems are suggested to collect heat. The analysis concludes that these engines WH between 370 K and 500 K. Therefore, this range of temperatures is used to simulate the suggested architectures.

Three different WFs are proposed to extract mechanical power from the mentioned temperature range. The criteria to evaluate them are the utilization factor, which depends directly on the WF. Air, Hydrogen and Helium are chosen to simulate the described power units. As expected, helium achieves significantly better results mainly, due to its unique properties. However, this section also demonstrates that it is possible to extract efficiently power using air as WF. The analysis of results concludes that the established systems can surpass considerably the performance exhibited by state-of-the-art technology and are suitable solutions to reduce the inherent problem of the low-grade heat energy conversion: Its low efficiency.

Thus, based on the results of the case studies carried out for air, hydrogen and helium as WFs it is worth highlighting that the characteristics of the used WFs play an important role in the cycle performance with independence of the thermal engine structure (which also exerts a strong negative influence on the performance such as the heat transfer effectiveness provoking the increment of entropy transfer by inherent irreversibilities of the real gases as components of the air, favouring the rapid degradation of the heat), which also obviously can exert some influence in the performance. Therefore, the conclusions related with the analysis of the case studies using the mentioned WFs let us to highlight the following comments regarding the used WFs:

Air as WF exhibits the lowest thermal efficiency as well as low specific work among the mentioned TWFs.

Since the dominant component of the air is a diatomic molecule, the N_2, ideally, its properties are similar to that of the H_2; however, the fact that the real operation undergoes the characteristics of a real gas its performance is lower than that of hydrogen.

However, it exhibits certain advantages versus hydrogen and helium, such as a healthy secure fluid, abundant, zero acquisition and maintenance costs, and non-degradable, non-pollutant as the most relevant advantages.

An important drawback associated with the proposed thermal engine is that each power unit must be designed for a unique WF since the characteristics of every TWF requires a specific re-dimensioning of the power unit components mainly those related to heat transfer and heat-work interactions, such as the dimensions of the reciprocating double-acting cylinders, as well as heat exchangers and forced convection equipment.

It has been proven by experimental evidence that effective heat transfer depends strongly on forced convection-based heat transfer techniques. This characteristic is applicable to all used WFs.

Hydrogen as WF exhibits interesting properties, such as heat transfer characteristics, high specific heat at constant pressure, which undergoes high specific work capacity. Although like the air it is a diatomic element, as real gas its heat transfer characteristics supposes an advantage improving the thermal efficiency comparing to the air.

As the most relevant drawback derived from the use of hydrogen, this WF exhibits undesired characteristics such as its highly explosive properties, so that its manipulation care is delicate.

As a consequence of its properties as real gas, such as its heat transfer properties and its unique high specific heat capacity undergoes much better performance results than air.

The helium as WF exhibits the best properties: it is an inert gas, also non-explosive, which exhibits good heat transfer properties and good heat capacity although less than hydrogen.

However, mono atomic structure associated with its high adiabatic expansion coefficient favours its high thermal performance.

Drawback: It is not abundant as a natural resource, must be manufactured at a high economic cost. It is dissipated and lost by diffusion through confinement walls, which undergoes a daily flow of fluid losses.

Its high thermal efficiency in thermal cycles with closed processes as a WF, its superiority compared to air or hydrogen is beyond doubt. It is far superior to the others.

Finally, it is worth noting that the great difference between the type of thermal machine studied in the thesis and conventional thermal machines, lies in the fact that at the same time as conventional machines operate by adding heat to convert into useful work while an important fraction of the heat is rejected and sent to a heat sink, the thermal machine studied in the thesis operates by adding heat, part of which is converted into useful mechanical work whereas the other part of the heat is extracted from the WF while performing useful mechanical work by adiabatic contraction of the WF, without economic cost which contributes to the enhancement of the thermal efficiency.

6.2 Suggestions for Further Work

Based on the analytical results provided by the thesis, which could only be reliable after enduring experimental viability tests, a great gap is opened in the experimental field. Thus, it is possible to know accurately which machine architectures are subject to implementation based on criteria of technical and economic feasibility that take into account both thermal efficiency and design, manufacturing or implementation and operating costs during their life cycle.

Then, starting from the fact that the proof of concept on which the research studies carried out to conclude the results of this thesis are based are validated only by conceptual results, it is possible to propose future lines of research in the field of thermal energy conversion to mechanical and / or electrical taking into account the qualitative aspects of the different design strategies and consequent architectures of conversion of thermal energy to mechanical.

Therefore, in order to increase both the thermal efficiency of the thermal engines based on the addition and extraction of heat at constant volume (instead of rejecting heat as in conventional thermal cycles, where useful work fulfill the energy balance w=qi-qo), as well as the heat utilisation factor, the following relevant topics should be explored by means of proposing new designs, developing and improving:

- Double-acting cylinders capable of operating discontinuously and intermittently so that it is possible to carry out isochoric heat transfer at inactive piston rod of the actuator cylinder.
- Effective linear to rotary conversion techniques to allow double-acting cylinders capable of operating discontinuously and intermittently.
- WFs of specific thermal capacity and high adiabatic expansion coefficient.
- Systems for WF retention and / or effectively recovery to minimize inherent losses.
- Cold exergy storage techniques to increase and take advantage of cooling capacities based on the combination with thermal energy conversion techniques based on heat extraction, which allows greater heat utilisation factor.

7 References

[1] S. Iglesias Garcia, R. Ferreiro Garcia, J. Carbia Carril, D. Iglesias Garcia, A review of thermodynamic cycles used in low temperature recovery systems over the last two years, Renew. Sustain. Energy Rev. 81 (2018) 760–767. https://doi.org/10.1016/j.rser.2017.08.049.

[2] S. Iglesias Garcia, R. Ferreiro Garcia, J. Carbia Carril, D. Iglesias Garcia, Critical review of the first-law efficiency in different power combined cycle architectures, Energy Convers. Manag. 148 (2017) 844–859. https://doi.org/10.1016/j.enconman.2017.06.037.

[3] C. Forman, I.K. Muritala, R. Pardemann, B. Meyer, Estimating the global waste heat potential, Renew. Sustain. Energy Rev. 57 (2016) 1568–1579. https://doi.org/10.1016/j.rser.2015.12.192.

[4] S. Lecompte, H. Huisseune, M. Van Den Broek, B. Vanslambrouck, M. De Paepe, Review of organic Rankine cycle (ORC) architectures for waste heat recovery, Renew. Sustain. Energy Rev. 47 (2015) 448–461. https://doi.org/10.1016/j.rser.2015.03.089.

[5] R. Ebara, F. Tanaka, M. Kawasaki, Sulfuric acid dew point corrosion in waste heat boiler tube for copper smelting furnace, Eng. Fail. Anal. 33 (2013) 29–36. https://doi.org/10.1016/j.engfailanal.2013.04.007.

[6] J. Sarkar, Review and future trends of supercritical CO2 Rankine cycle for low-grade heat conversion, Renew. Sustain. Energy Rev. 48 (2015) 434–451. https://doi.org/10.1016/j.rser.2015.04.039.

[7] S. Suárez De La Fuente, A.R. Greig, Making shipping greener: ORC modelling under realistic operative conditions, (2013) 1–17.

[8] F. Baldi, C. Gabrielii, A feasibility analysis of waste heat recovery systems for marine applications, Energy. 80 (2015) 654–665. https://doi.org/10.1016/j.energy.2014.12.020.

[9] S.E. Thermodynamics, Absorption Resorption Cycle for Heat Recovery of, in: Refrig. Air

Cond., 2008: pp. 1–8.

[10] J.M. Cullen, J.M. Allwood, Theoretical efficiency limits for energy conversion devices, Energy. 35 (2010) 2059–2069. https://doi.org/10.1016/j.energy.2010.01.024.

[11] MAN Marine, Serie G - Project Guides, (n.d.). https://marine.man-es.com/two-stroke/project-guides (accessed June 1, 2020).

[12] F. Di Natale, C. Carotenuto, Particulate matter in marine diesel engines exhausts: Emissions and control strategies, Transp. Res. Part D Transp. Environ. 40 (2015) 166–191. https://doi.org/10.1016/j.trd.2015.08.011.

[13] Z. Fotourehchi, Health effects of air pollution: An empirical analysis for developing countries, Atmos. Pollut. Res. 7 (2016) 201–206. https://doi.org/10.1016/j.apr.2015.08.011.

[14] HREII DEMO Observatory, Eu Paper : Orc Waste Heat Recovery in European Energy Intensive Industries, 2013. http://www.hreii.eu/public/Annex 4.2.II EU paper def.pdf.

[15] B. Kölsch, J. Radulovic, Utilisation of diesel engine waste heat by Organic Rankine Cycle, Appl. Therm. Eng. 78 (2015) 437–448. https://doi.org/10.1016/j.applthermaleng.2015.01.004.

[16] J. Song, Y. Song, C. wei Gu, Thermodynamic analysis and performance optimization of an Organic Rankine Cycle (ORC) waste heat recovery system for marine diesel engines, Energy. 82 (2015) 976–985. https://doi.org/10.1016/j.energy.2015.01.108.

[17] T. Deethayat, A. Asanakham, T. Kiatsiriroat, Performance analysis of low temperature organic Rankine cycle with zeotropic refrigerant by Figure of Merit (FOM), Energy. 96 (2016) 96–102. https://doi.org/10.1016/j.energy.2015.12.047.

[18] W. Pu, C. Yue, D. Han, W. He, X. Liu, Q. Zhang, Y. Chen, Experimental study on Organic Rankine cycle for low grade thermal energy recovery, Appl. Therm. Eng. 94 (2016) 221–227. https://doi.org/10.1016/j.applthermaleng.2015.09.120.

[19] M. Usman, M. Imran, Y. Yang, B.S. Park, Impact of organic Rankine cycle system installation on light duty vehicle considering both positive and negative aspects,

Energy Convers. Manag. 112 (2016) 382–394. https://doi.org/10.1016/j.enconman.2016.01.044.

[20] R. Shi, T. He, J. Peng, Y. Zhang, W. Zhuge, System design and control for waste heat recovery of automotive engines based on Organic Rankine Cycle, Energy. 102 (2016) 276–286. https://doi.org/10.1016/j.energy.2016.02.065.

[21] H. Tian, G. Shu, H. Wei, X. Liang, L. Liu, Fluids and parameters optimization for the organic Rankine cycles (ORCs) used in exhaust heat recovery of Internal Combustion Engine (ICE), Energy. 47 (2012) 125–136. https://doi.org/10.1016/j.energy.2012.09.021.

[22] E.H. Wang, H.G. Zhang, B.Y. Fan, M.G. Ouyang, Y. Zhao, Q.H. Mu, Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery, Energy. 36 (2011) 3406–3418. https://doi.org/10.1016/j.energy.2011.03.041.

[23] F. Yang, H. Zhang, C. Bei, S. Song, E. Wang, Parametric optimization and performance analysis of ORC (organic Rankine cycle) for diesel engine waste heat recovery with a fin-and-tube evaporator, Energy. 91 (2015) 128–141. https://doi.org/10.1016/j.energy.2015.08.034.

[24] M. Wang, J. Wang, Y. Zhao, P. Zhao, Y. Dai, Thermodynamic analysis and optimization of a solar-driven regenerative organic Rankine cycle (ORC) based on flat-plate solar collectors, Appl. Therm. Eng. 50 (2013) 816–825. https://doi.org/10.1016/j.applthermaleng.2012.08.013.

[25] Nasruddin, M. Idrus Alhamid, Y. Daud, A. Surachman, A. Sugiyono, H.B. Aditya, T.M.I. Mahlia, Potential of geothermal energy for electricity generation in Indonesia: A review, Renew. Sustain. Energy Rev. 53 (2016) 733–740. https://doi.org/10.1016/j.rser.2015.09.032.

[26] F. Molés, J. Navarro-Esbrí, B. Peris, A. Mota-Babiloni, Experimental evaluation of HCFO-1233zd-E as HFC-245fa replacement in an Organic Rankine Cycle system for low temperature heat sources, Appl. Therm. Eng. 98 (2016) 954–961. https://doi.org/10.1016/j.applthermaleng.2016.01.011.

[27] A. Desideri, S. Gusev, M. van den Broek, V. Lemort, S. Quoilin, Experimental comparison of organic fluids for low temperature ORC (organic Rankine cycle) systems for waste heat recovery applications, Energy. 97 (2016) 460–469. https://doi.org/10.1016/j.energy.2015.12.012.

[28] Y.M. Kim, D.G. Shin, C.G. Kim, G.B. Cho, Single-loop organic Rankine cycles for engine waste heat recovery using both low- and high-temperature heat sources, Energy. 96 (2016) 482–494. https://doi.org/10.1016/j.energy.2015.12.092.

[29] M.E. Mondejar, F. Ahlgren, M. Thern, M. Genrup, Quasi-steady state simulation of an organic Rankine cycle for waste heat recovery in a passenger vessel, Appl. Energy. 185 (2017) 1324–1335. https://doi.org/10.1016/j.apenergy.2016.03.024.

[30] P. Li, J. Li, G. Pei, J. Ji, A cascade organic Rankine cycle power generation system using hybrid solar energy and liquefied natural gas, Sol. Energy. 127 (2016) 136–146. https://doi.org/10.1016/j.solener.2016.01.029.

[31] M. Romero Gómez, R. Ferreiro Garcia, J. Romero Gómez, J. Carbia Carril, Review of thermal cycles exploiting the exergy of liquefied natural gas in the regasification process, Renew. Sustain. Energy Rev. 38 (2014) 781–795. https://doi.org/10.1016/j.rser.2014.07.029.

[32] M. Romero Gómez, J. Romero Gómez, L.M. López-González, L.M. López-Ochoa, Thermodynamic analysis of a novel power plant with LNG (liquefied natural gas) cold exergy exploitation and CO2capture, Energy. 105 (2016) 32–44. https://doi.org/10.1016/j.energy.2015.09.011.

[33] A.S. Panesar, An innovative Organic Rankine Cycle system for integrated cooling and heat recovery, Appl. Energy. 186 (2017) 396–407. https://doi.org/10.1016/j.apenergy.2016.03.011.

[34] T. Sung, K.C. Kim, Thermodynamic analysis of a novel dual-loop organic Rankine cycle for engine waste heat and LNG cold, Appl. Therm. Eng. 100 (2016) 1031–1041. https://doi.org/10.1016/j.applthermaleng.2016.02.102.

[35] R. Ferreiro García, J. Carbia Carril, J. Romero Gomez, M. Romero Gomez, Power plant

based on three series Rankine cycles combined with a direct expander using LNG cold as heat sink, Energy Convers. Manag. 101 (2015) 285–294. https://doi.org/10.1016/j.enconman.2015.05.051.

[36] T. Ho, S.S. Mao, R. Greif, Increased power production through enhancements to the Organic Flash Cycle (OFC), Energy. 45 (2012) 686–695. https://doi.org/10.1016/j.energy.2012.07.023.

[37] J.A.R. Sarr, F. Mathieu-Potvin, Improvement of Double-Flash geothermal power plant design: A comparison of six interstage heating processes, Geothermics. 54 (2015) 82–95. https://doi.org/10.1016/j.geothermics.2014.12.002.

[38] W.Y. Lee, M. Kim, Y.J. Sohn, S.G. Kim, Power optimization of a combined power system consisting of a high-temperature polymer electrolyte fuel cell and an organic Rankine cycle system, Energy. 113 (2016) 1062–1070. https://doi.org/10.1016/j.energy.2016.07.093.

[39] T. Wang, Y. Zhang, Z. Peng, G. Shu, A review of researches on thermal exhaust heat recovery with Rankine cycle, Renew. Sustain. Energy Rev. 15 (2011) 2862–2871. https://doi.org/10.1016/j.rser.2011.03.015.

[40] MAN Turbocharger, Waste Heat Recovery MAN, (n.d.). http://turbocharger.man.eu/technologies/waste-heat-recovery (accessed January 1, 2019).

[41] Man Diesel-Turbo, Waste Heat recovery, (n.d.). http://dieselturbo.man.eu/docs/default-source/shopwaredocumentsarchive/waste-heat-recovery.pdf?sfvrsn=2 (accessed January 1, 2017).

[42] P. Tremuli, Developments and Perspectives of Marine Engines, in: Wärtsilä EuDA Presentation Brussel (Ed.), Clean Combust. Greenh. Gases, 2008.

[43] M.D. Mirolli, Waste heat recovery, World Cem. 38 (2007) 85–88. http://www.123seminarsonly.com/Seminar-Reports/2016-01/171875692-Wartsila-Dual-Fuel-Engine.pdf.

[44] A.A. Kornhauser, Internal combustion engines, Fourth rep, Tata McGraw-Hill

Publishing Company Limited, 7 West Patel Nagar, New Delhi 110008, 2004. https://doi.org/10.4324/9780080560892-14.

[45] R. F. Garcia; J. C. Carril, Closed Processes Based Heat-Work Interactions Doing Useful Work by Adding and Realising Heat, Sryahwa. 6 (2018) 8–23.

[46] R. Garcia, B. Sanz, Isothermal and Adiabatic Expansion Based Trilateral Cycles, Br. J. Appl. Sci. Technol. 8 (2015) 448–460. https://doi.org/10.9734/bjast/2015/17350.

[47] R. Garcia, Contributions on Closed System Transformations Based Thermal Cycles, Br. J. Appl. Sci. Technol. 4 (2014) 2821–2836. https://doi.org/10.9734/bjast/2014/10074.

[48] R.F. Garcia, J.C. Carril, J.R. Gomez, M.R. Gomez, Energy and entropy analysis of closed adiabatic expansion based trilateral cycles, Energy Convers. Manag. 119 (2016) 49–59. https://doi.org/10.1016/j.enconman.2016.04.031.

[49] R.F. Garcia, J.C. Carril, J. Romero Gomez, M. Romero Gomez, Preliminary thermodynamic study of regenerative Otto based cycles with zero NOx emissions operating with adiabatic and polytropic expansion, Energy Convers. Manag. 113 (2016) 252–263. https://doi.org/10.1016/j.enconman.2016.01.066.

[50] R. García, C. Carril, Combined Cycle Consisting of Closed Processes Based Cycle Powered by A Reversible Heat Pump that Exceed Carnot Factor, J. Adv. Phys. 15 (2019) 6078–6100. https://doi.org/10.24297/jap.v15i0.8034.

[51] R. Garcia, Preliminary Study of an Efficient OTEC Using a Thermal Cycle with Closed Thermodynamic Transformations, Br. J. Appl. Sci. Technol. 4 (2014) 3840–3855. https://doi.org/10.9734/bjast/2014/11152.

[52] R. Garcia, B. Sanz, C. Sanz, Power Plant Driven by Residual Heat Rejected by the Bottoming Low Pressure Steam Turbines, Br. J. Appl. Sci. Technol. 6 (2015) 216–228. https://doi.org/10.9734/bjast/2015/12111.

[53] R. F. Garcia; J. C. Carril, Analysis and Comparison of Closed System Based Thermal Cycles Undergoing Different Polytropic Transformations, Sryahwa. 7 (2019) 26–37.

[54] M. Meng, K. Jing, S. Mander, Scenario analysis of CO2 emissions from China's electric

power industry, J. Clean. Prod. 142 (2017) 3101–3108. https://doi.org/10.1016/j.jclepro.2016.10.157.

[55] K. Zaman, M.A. el Moemen, Energy consumption, carbon dioxide emissions and economic development: Evaluating alternative and plausible environmental hypothesis for sustainable growth, Renew. Sustain. Energy Rev. 74 (2017) 1119–1130. https://doi.org/10.1016/j.rser.2017.02.072.

8 Appendices

Appendix 1: Basic EES code for a C4L

```
"Energy input"
m_o=313,9
T_o=431,4
cp_o=1,67
"Mass distribution"
m_hA=m_h/2
m_hB=m_h/2
m_cA=m_c/2
m_cB=m_c/2
"Temperature difference"
dT_w.w=10
dT_o.w=10
dT_o.a=40
dT_w.a=35
dT_he.w=10
dT_o.he=15
dT_e=5
"Initial pressure"
p0=100
"Pressure difference WF
dp=2/100
"Initial water pressure"
p_w=100
"Pressure difference Water
dp_w=0,2
p_w1=p_w+dp_w
"Coolant"
```

```
T_w=305
T_w1=T_w
"Output coolant
T_w2=T_w+dT_e
T_w3=T_w+dT_e
cp_w=CP(WATER;T=T_w;P=P_w)
"WF. For Air F1$=Air, for Helium F1$=He and for Hydrogen F1$=H2"
CP_1=CP(F1$;T=Tm)
CV_1=CV(F1$;T=Tm)
gamma=cp_1/cv_1
R=cp_1-cv_1
Tm=(T_A[4]+T_A[2])/2

"PU Simulation"

"Known PU temperatures"
T_A[2]=T_o-dT_o.a
T_A[4]=T_w+dT_w.a
T_B[2]=T_A[4]
T_B[4]=T_A[2]
"Known PU pressures"
p_A[1]=p0
p_A[3]=p0
p_B[3]=p0
p_B[1]=p0


"Side A"
"Isochoric heating A12"
p_A[1]*v_A[1]/T_A[1]=p_A[2]*v_A[2]/T_A[2]
v_A[2]=v_A[1]
```

```
v_A[1]=VOLUME(F1$;p=p_A[1];T=T_A[1])
"Adiabatic expansion A23"
T_A[2]/T_A[3]=(p_A[2]/p_A[3])^((gamma-1)/gamma)
"Isochoric cooling_A34"
(p_A[3]*V_A[3])/T_A[3]=(p_A[4]*V_A[4])/T_A[4]
V_A[4]=V_A[3]
v_A[3]=VOLUME(F1$;p=p_A[3];T=T_A[3])
"Adiabatic contraction_A41"
T_A[4]/T_A[1]=(p_A[4]/p_A[1])^((gamma-1)/gamma)
"Internal Energy, Enthalpy and Entropy state 1A"
u_A[1]=INTENERGY(F1$;T=T_A[1])
h_A[1]=ENTHALPY(F1$;T=T_A[1])
s_A[1]=ENTROPY(F1$;p=p_A[1];T=T_A[1])
"Internal Energy, Enthalpy and Entropy state 2A"
u_A[2]=INTENERGY(F1$;T=T_A[2])
h_A[2]=ENTHALPY(F1$;T=T_A[2])
s_A[2]=ENTROPY(F1$;p=p_A[2];T=T_A[2])
"Internal Energy, Enthalpy and Entropy state 3A"
u_A[3]=INTENERGY(F1$;T=T_A[3])
h_A[3]=ENTHALPY(F1$;T=T_A[3])
s_A[3]=ENTROPY(F1$;T=T_A[3];P=p_A[3])
"Internal Energy, Enthalpy and Entropy state 4A"
u_A[4]=INTENERGY(F1$;T=T_A[4])
h_A[4]=ENTHALPY(F1$;T=T_A[4])
s_A[4]=ENTROPY(F1$;p=p_A[4];T=T_A[4])


"Side B"
"Isochoric heating _B34"
(p_B[3]*V_B[3])/T_B[3]=(p_B[4]*V_B[4])/T_B[4]
V_B[3]=V_B[4]
```

```
v_B[3]=VOLUME(F1$;T=T_B[3];P=P_B[3])
"Adiabatic expansion_B41"
T_B[4]/T_B[1]=(p_B[4]/p_B[1])^((gamma-1)/gamma)
"Isochoric cooling_B12"
(p_B[1]*V_B[1])/T_B[1]=(p_B[2]*V_B[2])/T_B[2]
V_B[2]=V_B[1]
V_B[1]=VOLUME(F1$;T=T_B[1];P=p_B[1])
"Adiabatic contraction_B23"
T_B[3]/T_B[2]=(p_B[3]/p_B[2])^((gamma-1)/gamma)

"Internal Energy, Enthalpy and Entropy state 1B"
u_B[1]=INTENERGY(F1$;T=T_B[1])
h_B[1]=ENTHALPY(F1$;T=T_B[1])
s_B[1]=ENTROPY(F1$;p=p_B[1];T=T_B[1])
"Internal Energy, Enthalpy and Entropy state 2B"
u_B[2]=INTENERGY(F1$;T=T_B[2])
h_B[2]=ENTHALPY(F1$;T=T_B[2])
s_B[2]=ENTROPY(F1$;p=p_B[2];T=T_B[2])
"Internal Energy, Enthalpy and Entropy state 3B"
u_B[3]=INTENERGY(F1$;T=T_B[3])
h_B[3]=ENTHALPY(F1$;T=T_B[3])
s_B[3]=ENTROPY(F1$;T=T_B[3];P=p_B[3])
"Internal Energy, Enthalpy and Entropy state 4B"
u_B[4]=INTENERGY(F1$;T=T_B[4])
h_B[4]=ENTHALPY(F1$;T=T_B[4])
s_B[4]=ENTROPY(F1$;p=p_B[4];T=T_B[4])

"Mass balance"
T_o2=T_A[1]+dT_o.a
m_o*cp_o*(T_o-T_o2)=m_hA*(h_A[2]-h_A[1])+m_hB*(h_B[4]-h_B[3])
m_h=m_c
```

```
m_w1*cp_w*(T_w2-T_w)=m_cA*(h_A[3]-h_A[4])+m_cB*(h_B[1]-h_B[2])

"Transfer fan"
Vol_A[1]=m_hA*v_A[1]
Vol_A[2]=Vol_A[1]
Vol_A[3]=m_hA*v_A[3]
Vol_A[4]=Vol_A[3]
Vol_B[1]=m_cB*v_B[1]
Vol_B[2]=Vol_B[1]
Vol_B[3]=m_cB*v_B[3]
Vol_B[4]=Vol_B[3]
dV=Vol_A[3]-Vol_A[2]
eff_vt=80/100
m_t=dV/v_A[1]
Pot_vt=m_t*(h_B[1]-h_A[1])/eff_vt
"Convective fans"
dpv=2
eff_vc=80/100
Pot_vc=(m_h/2*dpv*v_A[1])/eff_vc
Pot_vcnet=4*Pot_vc

"Heat and work"
"w_A_12"
w_A12=0
"qi_A_12"
qi_A12-w_A12=u_A[2]-u_A[1]
"w_B_12"
w_B12=0
"qo_B_12"
qo_B12-w_B12=u_B[1]-u_B[2]
"qi_A_23"
```

```
qi_A23=0
"w_A_23"
qi_A23-w_A23=u_A[3]-u_A[2]
w_A23´=(1/(gamma-1))*(p_A[2]*V_A[2]-p_A[3]*V_A[3])
"qo_B_23"
qo_B23=0
"w_B_23"
qo_B23+w_B23=u_B[3]-u_B[2]
w_B23´=(1/(gamma-1))*(p_B[3]*V_B[3]-p_B[2]*V_B[2])
"w_A_34"
w_A34=0
"qo_A_34"
qo_A34+w_A34=u_A[3]-u_A[4]
"w_B_34"
w_B34=0
"qi_B_34"
qi_B34-w_B34=u_B[4]-u_B[3]
"qo_A_41"
qo_A41=0
"w_A_41"
qo_A41-w_A41=u_A[4]-u_A[1]
w_A41´=(1/(gamma-1))*(p_A[1]*V_A[1]-p_A[4]*V_A[4])
"qo_B_41"
qo_B41=0
"w_B_41"
qo_B41-w_B41=u_B[1]-u_B[4]
w_B41´=(1/(gamma-1))*(p_B[4]*V_B[4]-p_B[1]*V_B[1])
"Work_mechanical cycle"
w_c=w_A23+w_B23+w_A41+w_B41
eff_c=85/100
Pot_c=(m_cB*w_B23+m_cA*w_A41+m_hA*w_A23+m_hB*w_B41)*eff_c
```

```
"Absorbed heat"
q_i=qi_A12+qi_B34
Q_itot=qi_A12*m_hA+qi_B34*m_hB
"Extracted heat"
q_o=qo_A34+qo_B12
Q_otot=qo_A34*m_cA+qo_B12*m_cB
"Water Pump"
T_o3=T_w1+dT_e
m_w2*cp_w*(T_w3-T_w1)=m_o*cp_o*(T_o2-T_o3)
m_w=m_w1+m_w2
v_w=VOLUME(WATER;T=T_w;P=P_w)
Pot_bbw=m_w*v_w*(dp_w)
"Net power"
Pot_n=(Pot_c-Pot_vcnet-Pot_vt-Pot_bbw)
"Heat in the termal oil"
Q_oil=m_o*cp_o*(T_o-T_o2)
"Non-absorbed heat of the termal oil"
Q_op=m_o*cp_o*(T_o2-T_o3)
"efficiency"
eff=Pot_n/Q_itot

CR=Pot_n/Q_oil
```

Appendix 2: Resumen castellano

Esta tesis explora el potencial del calor de grado bajo y medio en diferentes ciclos termodinámicos utilizados para convertir el calor residual en trabajo mecánico. El objetivo de esta tesis es estudiar el estado del arte de los ciclos termodinámicos utilizados para recuperar calor de bajo grado y proponer mejoras.

La relevancia de investigar aplicaciones de calor de baja calidad o calor residual es que hay disponible una gran cantidad de energía térmica a un costo insignificante dentro del rango de temperaturas medias y bajas, con el inconveniente de que los ciclos térmicos existentes no pueden hacer un uso eficiente de dicho calor disponible a baja temperatura, debido a su baja eficiencia.

La presente tesis ofrece un enfoque diferente y analiza la recuperación de calor de bajo grado desde un punto de vista termodinámico, comparando su eficiencia de conversión. Los Ciclos Rankine Orgánico (ORCs) revisados muestran eficiencias similares. Por el contrario, los ciclos presentados en esta tesis, cerrados, sin condensación, que intercambian calor a volumen constante y convierten la energía térmica en trabajo mecánico adiabáticamente, tienen una configuración que permite la explotación eficiente del calor de bajo grado. También se analizan las estrategias de recuperación de calor y se proponen varios sistemas no convencionales para explotar el calor de bajo grado en la máquina propuesta.

Dos factores relevantes ejercen una fuerte influencia en el criterio para decidir la estructura de las plantas de propulsión marina:

- La reducción del consumo de combustible que experimenta al aumentar la eficiencia térmica de los motores térmicos asociados.
- La emisión de dióxido de carbono debido al consumo de combustibles fósiles

Por un lado, las emisiones de dióxido de carbono de la combustión de combustibles fósiles utilizados en plantas de propulsión marina y sus plantas de potencia asociadas son una preocupación debido a los impactos ambientales originados por tales emisiones de gases de efecto invernadero, lo que contribuye a modificar la composición de la atmósfera en un sentido muy negativo. Por lo tanto, estos sistemas nocivos pueden reemplazarse por estrategias de diseño alternativas, o al menos podrían mitigarse mediante la reducción de las emisiones de dióxido de carbono, lo que podría lograrse cambiando las estrategias de diseño tecnológico y metodologías operativas. La tecnología de recuperación de calor que se puede utilizar en sistemas de recuperación de calor residual (WHRS) de grado medio o bajo es uno de los medios capaces de disminuir las emisiones de dióxido de carbono y de reducir significativamente el consumo de combustible. Esto obedece al hecho de que es posible una mejora significativa en la eficiencia de los sistemas de propulsión marina, ya que contribuye a la reducción del consumo de combustibles fósiles del sector del transporte marítimo en general. La mejora mencionada se puede lograr aprovechando la energía calorífica residual agotada y / o rechazada por el sistema de propulsión y sus motores auxiliares asociados. La tecnología convencional hasta ahora optó por el aprovechamiento parcial de los gases de escape, ya que no se pudo utilizar calor a temperaturas más bajas.

Convencionalmente, la tecnología de vanguardia en este campo utiliza ORCs avanzados bajo varias estructuras de ciclo combinado como una entre algunos de los sistemas de recuperación de calor residual (WHRS) diferentes que se están utilizando para el aprovechamiento del calor de grado medio y / o bajo que se desperdicia por el sistema de propulsión que no puede ser utilizado por los motores de propulsión.

Por otro lado, la reducción del consumo de combustible, lo que significa aumentar la eficiencia térmica, provoca la reducción de las emisiones de dióxido de carbono. Además de una reducción del requerimiento energético que supone consumir menos al necesitar menos combustible para realizar el mismo cometido. Esto debe mejorarse tanto como sea posible, ya que este criterio es lo suficientemente relevante como para emprender el desarrollo de tecnologías destinadas a mejorar la eficiencia térmica de los motores térmicos utilizados hasta ahora.

Sin embargo, en esta tesis se investiga un nuevo paradigma de WHRS con el objetivo de aumentar la eficiencia térmica lo cual hace que se reduzca el consumo de combustibles fósiles y las emisiones de dióxido de carbono.

Estas mejoras están asociadas con las siguientes contribuciones propuestas y realizadas en esta tesis:

- Recuperación de calor de más baja temperatura mediante el diseño de una estructura de recuperación capaz de recuperar el calor residual de las temperaturas medias y bajas rechazadas hasta ahora, provenientes de gases de escape, enfriadores de aire de admisión, enfriadores de agua de camisas y enfriadores de aceite lubricante.
- Motores térmicos más eficientes mediante el diseño de análisis y realización de una prueba de concepto de ciclos térmicos basados en procesos cerrados que funcionan con interacciones que expanden y contraen el fluido de trabajo generando trabajo útil.

Además, los motores térmicos propuestos se caracterizan por un ciclo térmico físicamente realizable compuesto por dos procesos isocóricos cerrados que consisten en agregar y extraer calor hacia / desde el fluido de trabajo y dos procesos adiabáticos cerrados que convierten la energía térmica en trabajo mecánico útil, que consiste respectivamente en expansión adiabática cerrada y contracción adiabática cerrada.

Los principales resultados son:

- Un sistema energético capaz de convertir calor de bajo grado en trabajo mecánico con un factor de utilización más alto que cualquier tecnología disponible tanto comercialmente como en el campo teórico o académico.
- Un ciclo térmico que supera cualquier otro ciclo usado para calor de bajo grado en términos de eficiencia y factor de utilización.

En cuanto a la estructura de la tesis:

Capítulo 1 Introducción

El estudio realizado para la realización de esta tesis está organizado de tal manera que a partir de la motivación y la estrategia que justifica una revisión del estado del arte, se enfoca en la definición de objetivos y el establecimiento de estrategias de búsqueda de soluciones alternativas, que se lleva a cabo en el capítulo 1.

Capítulo 2 - Revisión de la tecnología existente relacionada con las técnicas de recuperación de calor residual en las plantas de propulsión de buques.

El Capítulo 2 resume el estado del arte del calor residual de bajo grado (LGWH). También revisa, desde un punto de vista termodinámico, la tecnología de los sistemas comerciales y publicados. También revisa la tecnología existente y analiza el rendimiento de las técnicas existentes para convertir la energía térmica en energía mecánica. Se revisaron los siguientes sistemas:

• Ciclos de Rankine y sus diversas variantes.

• Sistemas comerciales

Los ciclos basados en Rankine son uno de los ciclos termodinámicos más comunes para la recuperación de calor residual. En comparación con el ciclo de 4 procesos presentado en esta tesis, los ciclos de Rankine tienen dos desventajas principales en términos de termodinámica: en primer lugar, los ciclos basados en Rankine no tienen intercambios de calor cerrados, en segundo lugar, no usan una contracción adiabática en el ciclo de enfriamiento para entregar un trabajo efectivo, por lo tanto, la tecnología disponible tiene serias limitaciones termodinámicas.

Sin embargo, los sistemas basados en Rankine tienen algunas ventajas en comparación con las arquitecturas presentadas. Por ejemplo, no requieren transferir calor a dos fluidos en cascada. Esto se traduce directamente en considerables pérdidas de potencia. Un sistema de energía basado en ciclo Rankine generalmente requiere calentar directamente su fluido de trabajo. Por lo tanto, solo se requiere un proceso de transferencia de calor. La mayoría de las arquitecturas presentadas requieren calentar un fluido, que circula con bombas y calienta o enfría el fluido de trabajo de las diferentes unidades de potencia.

Capítulo 3 - Soluciones propuestas

El Capítulo 3 presenta y analiza un ciclo térmico que funciona con procesos cerrados isocóricos. Está compuesto por 4 procesos, que se utilizan en una máquina patentada para la simulación del capítulo 5. Como se describe, es la combinación de dos C3L, un ciclo basado solo en enfriamiento y un ciclo basado solo en calentamiento. El trabajo de contracción producido junto con los procesos cerrados utilizados son la principal diferencia en comparación con la tecnología actual.

Esta sección también valida el ciclo termodinámico. Demuestra que el ciclo cumple el equilibrio de entropía para un ciclo termodinámico. Además, describe una validación experimental realizada en un laboratorio, que concluye que una contracción adiabática puede proporcionar un trabajo útil. Finalmente, la sección explora las potenciales aplicaciones de este ciclo termodinámico.

Capítulo 4 - Estructuras de motor alternativas propuestas e implementación

El Capítulo 4 enumera y describe las patentes obtenidas durante la investigación y el desarrollo realizado al investigar ciclos térmicos basados en la expansión-contracción de procesos cerrados sin condensación. Hay tres patentes: un ciclo combinado y dos dispositivos mecánicos. Estos inventos están diseñados para poner en práctica los ciclos térmicos propuestos en esta tesis. Es un enfoque a los sistemas mecánicos que pueden materializar el ciclo termodinámico descrito, que incluye tres patentes. Estos inventos son la solución para dos problemas. P [1] es una arquitectura de motor que puede hacer uso del ciclo termodinámico. P [2] y P [3] son dispositivos que pueden convertir un movimiento alternativo lineal en un dispositivo rotatorio continuo.

P [1] Motor térmico alternativo regenerativo de doble efecto

La tecnología denominada motor térmico alternativo regenerativo de doble efecto, de procesos cerrados y abiertos y su procedimiento operativo, tiene como objetivo la conversión de energía térmica en energía mecánica y / o eléctrica mediante un ciclo térmico regenerativo de doble efecto, implementado en una máquina térmica alternativa de doble efecto regenerativo que funciona con helio o aire o hidrógeno entre dos fuentes de calor (fuente caliente que produce calor al ciclo y fuente fría que absorbe el calor del ciclo) destinado a la conversión de energía térmica en trabajo mecánico y / o energía eléctrica.

P [2] Convertidor de fuerza alternativo discontinuo a par rotativo continuo

El objeto de la tecnología actual llamada convertidor de fuerza alternativa discontinuo a par rotativo continuo, es el uso de energía térmica convertida en energía mecánica por un cilindro termomecánico convencional que opera con un ciclo térmico que incluye procesos cerrados a volumen constante durante el cual el pistón permanece en reposo, mientras que el eje de la fuente de alimentación gira a una velocidad continua. Entre el cilindro termomecánico y el eje de cigüeñal, se instalan dos cilindros hidráulicos, el primero para bloquear el movimiento del pistón y el vástago del cilindro termomecánico para lograr procesos termomecánicos a volumen constante, mientras que el segundo está habilitado para conectar la fuerza provista por el primer cilindro hidráulico con biela-cigüeñal para lograr un par de rotación.

P [3] Convertidor de movimiento lineal alternativo discontinuo a continuo

El objetivo de la tecnología actual llamada movimiento rotativo alternativo discontinuo a continuo, es la conversión eficiente de energía térmica a energía mecánica y / o eléctrica por medio de cilindros alternativos convencionales que operan con un ciclo térmico que incluye procesos cerrados a volumen constante durante el cual el vástago del pistón permanece en reposo, mientras que el eje de la fuente de alimentación gira a una velocidad continua.

Capítulo 5 - Estudio de casos: la implementación de sistemas de recuperación de calor residual en las plantas de propulsión de buques

El Capítulo 5 analiza las fuentes de calor residual de motores industriales o marinos y propone un sistema basado en unidades de potencia, que pueden realizar el trabajo siguiendo las interacciones termodinámicas descritas en esta tesis. Las eficiencias para un caso específico se calculan con diferentes configuraciones utilizando hidrogeno, aire y helio como fluido de trabajo. Además, el factor de utilización (HUF) se analiza y compara al usar las unidades de potencia propuestas. Por lo tanto, los resultados se analizan teniendo en cuenta el factor de utilización como variable.

El Capítulo 5 también evalúa la implementación de arquitecturas basadas en P [1]. Se utiliza un motor marino para estudiar su implementación. Se revisan diferentes motores marinos MAN. Los motores marinos o industriales desperdician la mayor parte de su calor en su aceite lubricante, aire de admisión, escape y agua de refrigeración. Se proponen tres sistemas de recuperación para recolectar calor. El análisis concluye que estos motores desperdician calor entre 400 K y 500 K. Por lo tanto, este rango de temperaturas se utiliza para simular las arquitecturas propuestas.

La máquina propuesta utiliza componentes industriales básicos. Está diseñado para trabajar con todo tipo de calor residual de bajo grado. Por lo tanto, no hay necesidad de implementar sistemas complejos, como inyectores de alta presión para sistemas de combustión. Las unidades de potencia (PU) constituyen una materialización de los ciclos C4L. La PU materializa de la mejor manera posible el ciclo termodinámico que caracteriza al C4L.

El potencial de las arquitecturas de PU se explora utilizando el calor residual de un motor Diesel industrial comercial. El funcionamiento de la PU se describe en esta sección. Finalmente, el parámetro utilizado para evaluar y comparar las diferentes arquitecturas basadas en unidades de potencia C4L es la eficiencia térmica. La eficiencia térmica se utiliza a su vez para evaluar la HUF de una planta de energía marina. Esta variable indica la cantidad de energía utilizada y desperdiciada en una planta de potencia de acuerdo con la descripción en la sección 2.4. Los resultados muestran que el uso de ciclos C4L puede mejorar considerablemente la cantidad de energía convertida. Esto se debe a su capacidad de convertir calor a baja temperatura, que es el principal obstáculo de la tecnología disponible.

Se proponen tres fluidos de trabajo diferentes para extraer la potencia mecánica del rango de temperatura mencionado. El criterio para evaluarlos es el factor de utilización (HUF), que depende directamente del fluido de trabajo. Helio, Hidrógeno y Aire son elegidos para simular los sistemas de energía. Estos fluidos tienen propiedades térmicas diferentes, uno con características menos convenientes: aire, otro con características favorables, pero no tan abundante: helio y entre ambos el hidrógeno que posee como característica resaltable su calor especifico a volumen constante. Como se esperaba, el helio logra resultados significativamente mejores, principalmente, debido a sus propiedades únicas. Sin embargo, esta sección también demuestra que es posible extraer energía eficientemente usando aire como fluido de trabajo. El análisis de los resultados concluye que los sistemas propuestos pueden superar considerablemente la tecnología punta y son soluciones adecuadas para reducir el problema inherente de la conversión de energía de bajo grado: su baja eficiencia.

Con respecto a la HUF, el sistema propuesto supera a los sistemas comerciales. La eficiencia en sí del WHRS equipado con el PU descrito es mayor que la eficiencia de las soluciones de Rankine de los sistemas comerciales. Además, el calor residual recuperado es considerablemente mayor. Por ejemplo, el calor de escape se recupera hasta una temperatura más baja, la tecnología actual no puede enfriar los gases de escape porque, el costo de tener materiales para resistir el punto de rocío de los gases de escape es mayor a lo que la tecnología actual puede suministrar con este calor adicional. Por esta razón, el WHRS propuesto casi duplica los sistemas comerciales en términos de capacidad de conversión térmica. Esto demuestra la importante diferencia entre eficiencia térmica y HUF. Sistemas como proponen los fabricantes tienen cierta eficiencia térmica, pero se aplican a una fracción del calor residual, lo que hace que esos sistemas tengan un HUF bajo.

La PU propuesta se puede aplicar a una amplia gama de calor residual. Por lo tanto, el análisis que se hace en esta tesis no es un caso aislado. El método de conversión propuesto en esta tesis es un claro paso adelante en el campo de la conversión de energía. Como se exploró anteriormente, la principal diferencia entre la tecnología actual y el sistema propuesto es el enfoque termodinámico. Los sistemas actuales no utilizan eficientemente el calor de baja temperatura debido a restricciones termodinámicas, principalmente, intercambiando calor usando procesos abiertos, usando fluidos de trabajo poco eficientes y no usando ciclos que puedan explotar el trabajo de contracción.

Capítulo 6 - Conclusiones generales

El Capítulo 6 discute el trabajo realizado con la tecnología presentada a partir del capítulo 1 y hace una conclusión sobre la investigación y el desarrollo realizado en esta tesis. Esta sección también sugiere más investigaciones.

Con base en los resultados analíticos proporcionados por la tesis, que solo se podría confirmar después de ser soportadas por más pruebas experimentales, se abre una gran brecha en el campo experimental. Por lo tanto, sería posible saber con precisión qué arquitecturas de máquinas están sujetas a implementación en función de criterios de viabilidad técnica y económica que tienen en cuenta tanto la eficiencia térmica como el diseño, la fabricación o la implementación y los costos operativos durante su ciclo de vida.

Luego, a partir del hecho de que la prueba de concepto en la que se basan los estudios de investigación realizados para concluir los resultados de esta tesis se validan solo por resultados conceptuales, es posible proponer futuras líneas de investigación en el campo de la conversión de energía térmica a mecánica y / o eléctrica teniendo en cuenta los aspectos cualitativos de las diferentes estrategias de diseño y las consiguientes arquitecturas de conversión de energía térmica a mecánica.

Por lo tanto, para aumentar tanto la eficiencia térmica de los motores térmicos en función de la adición y extracción de calor a volumen constante (en lugar de rechazar el calor como en los ciclos térmicos convencionales, donde el trabajo útil cumple el balance de energía $w = qi-qo$), así como el factor de utilización del calor, se deben explorar los siguientes temas relevantes mediante la propuesta de nuevos diseños, desarrollo y mejora:

• Cilindros de doble efecto capaces de operar de manera discontinua e intermitente para que sea posible realizar una transferencia de calor isócora

• Técnicas efectivas de conversión lineal a rotativo para permitir cilindros de doble efecto capaces de operar de manera discontinua e intermitente.

• Fluidos de trabajo de alta capacidad térmica específica y alto coeficiente de expansión adiabática.

• Sistemas para la retención y / o recuperación de fluidos de trabajo térmico para minimizar las pérdidas inherentes.

• Técnicas de almacenamiento de frío para aumentar y aprovechar las capacidades de enfriamiento basadas en la combinación con técnicas de conversión de energía térmica basadas en la extracción de calor, lo que permite un mayor factor de utilización del calor.

Appendix 2: Resumo galego

Esta tese explora o potencial de calor de baixo e medio grao en diferentes ciclos termodinámicos empregados para converter a calor residual en traballo mecánico. O obxectivo desta tese é estudar o estado da técnica dos ciclos termodinámicos empregados para recuperar calor de baixo grao e propoñer melloras.

A relevancia de investigar aplicacións de calor de baixa calidade ou de calor residual é que unha enorme cantidade de enerxía térmica está dispoñible a un custo insignificante no intervalo de temperaturas medias e baixas, co inconveniente de que os ciclos térmicos existentes non poden facer un uso eficiente de tal calor dispoñible a baixa temperatura, debido á súa baixa eficiencia. O motivo de esa baixa eficiencia estudase nesta tese e consiste basicamente en un problema termodinámico nunca resolto ata agora. O cal consiste en usar transformacións termodinámicas de baixa eficiencia que se asumiu indiscutiblemente ao largo da historia como idóneas para a conversión de calor en traballo. Esta tese enfocase en investigar e propoñer técnicas de vangarda que como resultado obteñen unha eficiencia moi superior ao que se fixo historicamente e se fai aínda actualmente.

A presente tese ofrece un enfoque diferente, analiza a recuperación de calor de baixo grao desde o punto de vista termodinámico e compara a súa eficiencia térmica. Os Ciclos Rankine Organico (ORCs) revisados mostran eficiencias baixas similares. En contraste, os ciclos sen condensación baseados en pechados, que intercambian calor a volume constante e converten a enerxía térmica en traballo mecánico adiabáticamente, teñen unha configuración, que permite unha explotación eficiente de calor de baixo grao. Tamén se analizan as estratexias de recuperación de calor e proponse varios sistemas non convencionais para explotar calor de baixo grao na máquina proposta.

Dous factores relevantes exercen unha forte influencia no criterio para decidir a estrutura das plantas de propulsión mariña:

- A redución do consumo de combustible que sofre ao aumentar a eficiencia térmica dos motores térmicos asociados.
- A emisión de dióxido de carbono debido ao consumo de combustibles fósiles.

Por unha banda, as emisións de dióxido de carbono procedentes da combustión de combustibles fósiles empregados nas plantas de propulsión mariña e as súas plantas de potencia asociadas son unha preocupación e débese aos impactos ambientais orixinados por estas emisións de gases de efecto invernadoiro, o que contribúe a modificar a composición da atmosfera nun sentido moi negativo. Así, estes sistemas nocivos pódense substituír por estratexias de deseño alternativo ou polo menos poderíanse paliar mediante a redución das emisións de dióxido de carbono, que se poderían conseguir cambiando as estratexias de deseño de tecnoloxía e metodoloxías operativas. A tecnoloxía de recuperación de calor dispoñible que se pode empregar para sistemas de recuperación de calor residual (WHRS) de grao medio ou baixo é un tal medio que é tecnoloxicamente capaz de diminuír as emisións de dióxido de carbono, así como de reducir significativamente o consumo de combustible. Isto obedece a que é posible unha mellora significativa da eficiencia dos sistemas de propulsión mariña xa que contribúe á redución do consumo de combustibles fósiles do sector do transporte marítimo en xeral.

A mellora mencionada pódese aproveitar usando a calor residual esgotada e / ou rexeitada polo sistema de propulsión e os seus motores auxiliares asociados. A tecnoloxía convencional ata o de agora optaba polo aproveitamento dos gases de combustión do escape, xa que non se podía usar calor a temperatura máis baixa.

Convencionalmente, a tecnoloxía de punta neste campo usa ORC avanzados en varias estruturas combinativas como unha das diferentes tecnoloxías WHRS que se están empregando para o aproveitamento de calor de medio e / ou baixo grao residual polo sistema de propulsión que non pode ser empregado polos propulsores.

Por outra banda, a redución do consumo de combustible, que significa aumentar a eficiencia térmica sometida á redución das emisións de dióxido de carbono, debe mellorarse na medida do posible, baseado en que este criterio é o suficientemente relevante como para emprender o desenvolvemento de tecnoloxías dirixidas a mellorando a eficiencia térmica dos motores térmicos empregados ata o de agora. Ademais da redución de carga que supón consumir menos ao necesitar menos combustible para realizar o mesmo cometido De esta maneira, diminúese o requirimento enerxético da planta propulsora e mellorase aínda a súa eficiencia.

Non obstante, nesta tese investígase un novo paradigma de WHR co obxectivo de aumentar a eficiencia térmica que se reduce o consumo de combustibles fósiles e as emisións de dióxido de carbono.

Estas melloras están asociadas ás seguintes contribucións propostas e realizadas nesta tese:

- Recuperación de calor de baixa temperatura mediante o deseño dunha estrutura de recuperación capaz de recuperar a calor residual de temperaturas medias e baixas rexeitadas ata o momento, procedentes de gases de escape, refrixerador de aire de admisión, refrixerador de auga de camisas, e refrixerador de aceite de lubricación.

- Motores térmicos máis eficientes mediante o deseño de análise e realización dunha proba de concepto de ciclos térmicos baseados en procesos pechados que accionan engadindo e extraendo calor sometido a un modo de interacción calor-traballo innovador.

Ademais, os motores térmicos propostos caracterízanse por un ciclo térmico realizable fisicamente composto por dous procesos isocóricos pechados consistentes na adición e extracción de calor cara / dende o fluído de traballo e dous procesos adiabáticos pechados que sofren un traballo mecánico útil, consistente respectivamente por unha expansión adiabática pechada e compresión adiabática pechada a base de contracción do fluído de traballo.

Os principais resultados son:

- Unha arquitectura de potencia capaz de converter a calor de baixo grao en traballo mecánico cun factor de utilización superior a calquera tecnoloxía dispoñible ou teórica.
- Un ciclo térmico que supera calquera outro ciclo para calor de baixo grao en termos de eficiencia e factor de utilización.

En canto á estrutura da tese:

Capítulo 1 Introdución

O estudo realizado para a realización desta tese está organizado de tal maneira que a partir da motivación e a estratexia que xustifica unha revisión da estado da arte, enfócase na definición de obxectivos e o establecemento de estratexias de procura de solucións alternativas, que leva a cabo no capítulo 1.

Capítulo 2 - Revisión da tecnoloxía existente relacionada coas técnicas de recuperación de calor residual nas plantas de propulsión de buques.

O Capítulo 2 resume a estado da arte da calor residual de baixo grao (LGWH). Tamén revisa, desde un punto de vista termodinámico, a tecnoloxía dos sistemas comerciais e publicados. Tamén revisa a tecnoloxía existente e analiza o rendemento das técnicas existentes para converter a enerxía térmica en enerxía mecánica. Revisáronse os seguintes sistemas:

• Ciclos de Rankine e as súas diversas variantes.

• Sistemas comerciais

Os ciclos baseados en Rankine son un dos ciclos termodinámicos máis comúns para a recuperación de calor residual. En comparación co ciclo de 4 procesos presentado nesta tese, os ciclos de Rankine teñen dúas desvantaxes principais en termos termodinámicos: en primeiro lugar, os ciclos baseados en Rankine non teñen intercambios de calor cerrados, en segundo lugar, non usan unha contracción adiabática no ciclo de arrefriado para entregar un traballo efectivo, por tanto, a tecnoloxía dispoñible ten serias limitacións termodinámicas.

Con todo, os sistemas baseados en Rankine teñen algunhas vantaxes en comparación coas arquitecturas presentadas. Por exemplo, non requiren transferir calor a dous fluídos en fervenza. Isto tradúcese directamente en considerables perdas de potencia. Un sistema de enerxía de Rankine xeralmente require quentar directamente o seu fluído de traballo. Por tanto, só requírese un proceso de transferencia de calor. A maioría das arquitecturas presentadas requiren quentar un fluído, que circula con bombas e quentar ou arrefriar o fluído de traballo das diferentes unidades de potencia.

Capítulo 3 - Solucións propostas

O Capítulo 3 presenta e analiza un ciclo térmico que funciona con procesos cerrados isocóricos. Está composto por 4 procesos, que se utilizan nunha máquina patentada para a simulación do capítulo 5. Como se describe, é a combinación de dúas C3L, un ciclo baseado só en arrefriado e un ciclo baseado só en quecemento. O traballo de contracción producido xunto cos procesos cerrados utilizados son a principal diferenza en comparación coa tecnoloxía actual.

Esta sección tamén valida o ciclo termodinámico. Demostra que o ciclo cumpre o equilibrio de entropía para un ciclo termodinámico. Ademais, describe unha validación experimental realizada nun laboratorio, que conclúe que unha contracción adiabática pode proporcionar un traballo útil. Finalmente, a sección explora as potenciais aplicacións deste ciclo termodinámico.

Capítulo 4 - Estruturas de motor alternativas propostas e implementación

O Capítulo 4 enumera e describe as patentes obtidas durante a investigación e o desenvolvemento realizado ao investigar ciclos térmicos baseados na expansión-contracción de procesos cerrados sen condensación. Hai tres patentes: un ciclo combinado e dous dispositivos mecánicos. Estes inventos están deseñados para poñer en práctica os ciclos térmicos propostos nesta tese. É un enfoque aos sistemas mecánicos que poden materializar o ciclo termodinámico descrito, que inclúe tres patentes. Estes inventos son a solución para dous problemas. P [1] é unha arquitectura de motor que pode facer uso do ciclo termodinámico. P [2] e P [3] son dispositivos que poden converter un movemento alternativo lineal nun dispositivo rotatorio continuo.

P [1] Motor térmico alternativo rexenerativo de dobre efecto

A tecnoloxía denominada motor térmico alternativo rexenerativo de dobre efecto, de procesos cerrados e abertos e o seu procedemento operativo, ten como obxectivo a conversión de enerxía térmica en enerxía mecánica e / ou eléctrica mediante un ciclo térmico rexenerativo de dobre efecto, implementado nunha máquina térmica alternativa de dobre efecto rexenerativo que funciona con helio ou aire ou hidróxeno entre dúas fontes de calor (fonte quente que produce calor ao ciclo e fonte fría que absorbe a calor do ciclo) destinado á conversión de enerxía térmica en traballo mecánico e / ou enerxía eléctrica.

P [2] Convertedor de forza alternativo descontinuo a par rotativo continuo

O obxecto da tecnoloxía actual chamada convertedor de forza alternativa descontinuo a par rotativo continuo, é o uso de enerxía térmica convertida en enerxía mecánica por un cilindro termomecánico convencional que opera cun ciclo térmico que inclúe procesos cerrados a volume constante durante o cal o pistón permanece en repouso, mentres que o eixo da fonte de alimentación vira a unha velocidade continua. Entre o cilindro termomecánico e o eixo de cigüeñal, instálanse dous cilindros hidráulicos, o primeiro para bloquear o movemento do pistón e o vástago do cilindro termomecánico para lograr procesos termomecánicos a volume constante, mentres que o segundo está habilitado para conectar a forza provista polo primeiro cilindro hidráulico con biela-cigüeñal para lograr un par de rotación.

P [3] Convertedor de movemento linear alternativo descontinuo a continuo

O obxectivo da tecnoloxía actual chamada movemento rotativo alternativo descontinuo a continuo, é a conversión eficiente de enerxía térmica a enerxía mecánica e / ou eléctrica por medio de cilindros alternativos convencionais que operan cun ciclo térmico que inclúe procesos cerrados a volume constante durante o cal o vástago do pistón permanece en repouso, mentres que o eixo da fonte de alimentación vira a unha velocidade continua.

Capítulo 5 - Estudo de casos: a implementación de sistemas de recuperación de calor residual nas plantas de propulsión de buques

O Capítulo 5 analiza as fontes de calor residual de motores industriais ou mariños e propón un sistema baseado en unidades de potencia, que poden realizar o traballo seguindo as interaccións termodinámicas mencionadas en esta tese. As eficiencias para un caso específico calcúlanse con diferentes configuracións utilizando hidroxeno aire e helio como fluído de traballo. Ademais, o factor de utilización (HUF) analízase e compara ao usar as unidades de potencia propostas. Por tanto, os resultados analízanse utilizando o factor de utilización como variable.

O Capítulo 5 tamén avalía a implementación de arquitecturas baseadas en P [1]. Utilízase un motor mariño para estudar a súa implementación. Revísanse diferentes motores mariños MAN. Os motores mariños ou industriais desperdician a maior parte da súa calor no seu aceite lubricante, aire de admisión, escape e auga de refrixeración. Proponse tres sistemas de recuperación para colleitar calor. A análise conclúe que estes motores desperdician calor entre 400 K e 500 K. Por tanto, este rango de temperaturas utilízase para simular as arquitecturas propostas.

A máquina proposta utiliza compoñentes industriais básicos. Está deseñado para traballar con todo tipo de calor residual de baixo grao. Por tanto, non hai necesidade de implementar sistemas complexos, como inxectores de alta presión para sistemas de combustión. As unidades de potencia (PU) constitúen unha materialización dos ciclos C4L. A PU materializa da mellor maneira posible o ciclo termodinámico que caracteriza ao C4L.

O potencial das arquitecturas de PU explórase utilizando a calor residual dun motor Diesel industrial comercial. O funcionamento da unidade de potencia descríbese nesta sección. Finalmente, o parámetro utilizado para avaliar e comparar as diferentes arquitecturas baseadas en unidades de potencia C4L é a eficiencia térmica. A eficiencia térmica utilízase á súa vez para avaliar a HUF dunha planta de enerxía mariña. Esta variable indica a cantidade de enerxía utilizada e desperdiciada nunha planta de potencia de acordo coa descrición na sección 2.4. Os resultados mostran que o uso de ciclos C4L pode mellorar considerablemente a cantidade de enerxía convertida. Isto débese á súa capacidade de converter calor a baixa temperatura, que é o principal obstáculo da tecnoloxía dispoñible.

Proponse tres fluídos de traballo diferentes para extraer a potencia mecánica do rango de temperatura mencionado. O criterio para avalialos é o factor de utilización (HUF), que depende directamente do fluído de traballo. Helio, Hidróxeno e Aire son elixidos para simular os sistemas de enerxía. Estes fluídos teñen propiedades térmicas diferentes, un con características menos convenientes: aire, outro con características favorables, pero non tan abundante: helio e entre ambos o hidróxeno que posúe como característica resaltable a súa calor especifico a volume constante. Como se esperaba, o helio logra resultados significativamente mellores, principalmente, debido ás súas propiedades únicas. Con todo, esta sección tamén demostra que é posible extraer enerxía eficientemente usando aire como fluído de traballo. A análise dos resultados conclúe que os sistemas propostos poden superar considerablemente a tecnoloxía punta e son solucións adecuadas para reducir o problema inherente da conversión de enerxía de baixo grao: a súa baixa eficiencia.

Con respecto á HUF, o sistema proposto supera aos sistemas comerciais. A eficiencia en si do WHRS equipado co PU descrito é maior que a eficiencia das solucións de Rankine dos sistemas comerciais. Isto pódese lograr incluso con un WF común e a baixas presións, como é o caso do aire. Isto convela que o sistema ademais de usar un WF non contaminante, é un WF abóndante. Traballar a baixas presións tamén leva a que o risco de fugas é menor. Ademais, a calor residual recuperado é considerablemente maior. Por exemplo, a calor de escape recupérase ata unha temperatura máis baixa, a tecnoloxía actual non pode arrefriar os gases de escape porque, o custo de ter materiais para resistir o punto de rocío dos gases de escape é maior ao que a tecnoloxía actual pode fornecer con esta calor adicional. Por esta razón, o WHRS proposto case duplica os sistemas comerciais en termos de capacidade de conversión térmica. Isto demostra a importante diferenza entre eficiencia térmica e HUF. Sistemas como propoñen os fabricantes teñen certa eficiencia térmica, pero aplícanse a unha fracción da calor residual, o que fai que eses sistemas teñan un HUF baixo.

A PU proposta pódese aplicar a unha ampla gama de calor residual. Por tanto, a análise que se fai nesta tese non é un caso illado. Sen embargo, cabe recoñecer que non se trata da solución máis axeitada para calores de moi alta temperatura, o cal non é obxecto de estudo nesta tese doutoral. O método de conversión proposto nesta tese é un claro paso adiante no campo da conversión de enerxía. Como se explorou anteriormente, a principal diferenza entre a tecnoloxía actual e o sistema proposto é o enfoque termodinámico. Os sistemas actuais non utilizan eficientemente a calor de baixa temperatura debido a restricións termodinámicas, principalmente, intercambiando calor usando procesos abertos, usando fluídos de traballo pouco eficientes e non usando ciclos que poidan explotar o traballo de contracción.

O Capítulo 6 discute o traballo realizado coa tecnoloxía presentada a partir do capítulo 1 e fai unha conclusión sobre a investigación e o desenvolvemento realizado nesta tese. Esta sección tamén suxire máis investigacións.

Con base nos resultados analíticos proporcionados pola tese, que só se podería confirmar despois de ser soportadas por máis probas experimentais, ábrese unha gran brecha no campo experimental. Por tanto, é posible saber con precisión que arquitecturas de máquinas están suxeitas a implementación en función de criterios de viabilidade técnica e económica que teñen en conta tanto a eficiencia térmica como o deseño, a fabricación ou a implementación e os custos operativos durante o seu ciclo de vida.

Logo, a partir do feito de que a proba de concepto na que se basean os estudos de investigación realizados para concluír os resultados desta tese valídanse só por resultados conceptuais, é posible propor futuras liñas de investigación no campo da conversión de enerxía térmica a mecánica e / ou eléctrica tendo en conta os aspectos cualitativos das diferentes estratexias de deseño e as conseguintes arquitecturas de conversión de enerxía térmica a mecánica.

Por tanto, para aumentar tanto a eficiencia térmica dos motores térmicos en función da adición e extracción de calor a volume constante (en lugar de rexeitar a calor como nos ciclos térmicos convencionais, onde o traballo útil cumpre o balance de enerxía w =qi-qo), así como o factor de utilización da calor, débense explorar os seguintes temas relevantes mediante a proposta de novos deseños, desenvolvemento e mellora:

• Cilindros de dobre efecto capaces de operar de maneira descontinua e intermitente para que sexa posible realizar unha transferencia de calor isócora

• Técnicas efectivas de conversión lineal a rotativo para permitir cilindros de dobre efecto capaces de operar de maneira descontinua e intermitente.

• Fluídos de traballo de alta capacidade térmica específica e alto coeficiente de expansión adiabática.

• Sistemas para a retención e / ou recuperación de fluídos de traballo térmico para minimizar as perdas inherentes.

• Técnicas de almacenamento de frío para aumentar e aproveitar as capacidades de arrefriado baseadas na combinación con técnicas de conversión de enerxía térmica baseadas na extracción de calor, o que permite un maior factor de utilización da calor.