archives
of thermodynamics
Vol. **40**(2019), No. 3, 195–210
DOI: 10.24425/ather.2019.130001

Identification of waste heat energy sources of a conventional steam propulsion plant of an LNG carrier

ANDRZEJ ADAMKIEWICZ*
SZYMON GRZESIAK

Maritime University of Szczecin, Faculty of Maritime Engineering,
Wały Chrobrego 1-2, 70-500 Szczecin, Poland

Abstract This paper presents the origins of marine steam turbine application on liquefied natural gas carriers. An analysis of alternative propulsion plant trends has been made. The more efficient ones with marine diesel engines gradually began to replace the less efficient plants. However, because of many advantages of the steam turbine, further development research is in progress in order to achieve comparable thermal efficiency. Research has been carried out in order to achieve higher thermal efficiency throughout increasing operational parameters of superheated steam before the turbine unit; improving its efficiency to bring it nearer to the ideal Carnot cycle by applying a reheating system of steam and multi stage regenerative boiler feed water heating. Furthermore, heat losses of the system are reduced by: improving the design of turbine blades, application of turbine casing and bearing cooling, as well as reduction in steam flow resistance in pipe work and maneuvering valves. The article identifies waste energy sources using the energy balance of a steam turbine propulsion plant applied on the liquefied natural gas carrier which was made out basing on results of a passive operation experiment, using the measured and calculated values from behavioral equations for the zero-dimensional model. Thermodynamic functions of state of waste heat fluxes have been identified in terms of their capability to be converted into usable energy fluxes. Thus, new ways of increasing the efficiency of energy conversion of a steam turbine propulsion plant have been addressed.

Keywords: Waste heat energy; Steam turbine; Efficiency; Propulsion plant

*Corresponding Author. Email: a.adamkiewicz@am.szczecin.pl

Nomenclature

b	–	physical exergy, kJ/kg
c_p	–	heat capacity, kJ/kgK
i	–	specific enthalpy, kJ/kg
$\dot{m}$	–	mass flow rate, kg/s
p	–	pressure, kPa
s	–	specific entropy, kJ/kgK
t	–	temperature, °C
u	–	internal energy, kJ/kg
V	–	specific volume, m^3/kg

Greek symbols

η	–	efficiency
λ	–	excess air coefficient
ψ_t	–	temperature coefficient of energy quality
$\psi_{\Delta i}$	–	exergy coefficient of energy quality

Subscripts and superscripts

alt	–	alternator
$b/\Delta i$	–	coefficient as function of exergy and enthalpy differential
gb	–	gearbox
exh	–	exhaust
i	–	internal
m	–	mechanical
T	–	temperature
0	–	reference state
1,2,...	–	number of control plane

Abbreviations

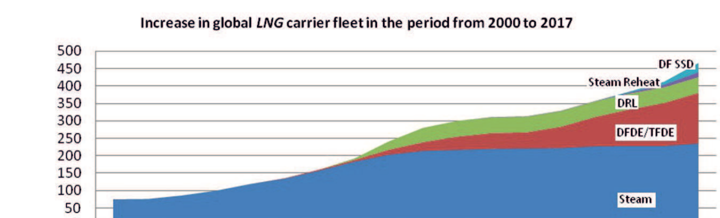
ADT	–	atmospheric drain tank
CST	–	conventional steam turbine
DFDE	–	dual fuel diesel electric
DF SSD	–	dual fuel slow speed diesel
DRL	–	diesel with re-liquefaction plant
FP	–	feed pump
FW	–	feed water
G/B	–	gear box
HFO	–	heavy fuel oil
HP, IP, LP	–	high-, intermediate-, low-pressure
IAS	–	integrated automation system

LNG	–	liquefied natural gas
LPSG	–	low pressure steam generator
MCR	–	maximum continuous rating
MT	–	main turbine
PMS	–	power management system
RPM	–	revolutions per minute
TA	–	turbo alternator
TFDE	–	triple fuel diesel electric

1 Introduction

Over the last decade marine transport of natural gas has become increasingly common due to technical difficulties of gas transmission over long distances which is confirmed by the number of newly built vessels and liquefied natural gas (LNG) terminals constructed around the world. Just in 2016 the maritime transport of natural gas has recorded a 5% increase compared to 2015 [5,6].

For LNG carriers, until the first decade of the 21st century steam power plants had been the leading solution due to the possibility of having boilers fed by both fuel oil and gas. That tendency, however, changed in the period from 2005 to 2010 (Fig. 1). An increase in LNG prices from 2001 to 2009 forced the ship industry to seek new, more economical solutions for power plants of LNG carriers [2,5].



	2000	2001	2002	2003	2004	2005	2006	2007	2008	2009	2010	2011	2012	2013	2014	2015	2016	2017
DF SSD																3	9	27
Steam Reheat												1	1	1	1	4	8	15
DRL								4	24	41	45	45	45	45	45	45	45	45
DFDE/TFDE						1	3	5	14	25	38	45	46	61	84	105	123	145
Steam	75	76	86	100	119	135	158	183	203	214	217	220	221	222	227	228	228	235

Figure 1: Increase in LNG carrier fleet based on propulsion plant type (2000–2017) [6].

From 2005 to 2017 ships with alternative propulsion plants were being launched into operations. They were equipped in power systems with higher and higher efficiencies, emitting in their exhaust smaller and smaller fluxes of toxic compounds and also characterized by higher operational flexibility [5]. From among alternative power systems the most commonly used at present are dual fuel diesel electric/triple fuel diesel electric (DFDE/TFDE) for the drive of which dual or triple fuel, medium speed, diesel engine are used. From 2007 to 2010, 45 ships with the diesel re-liquefaction plant (DRL) were launched. These propulsion plants were equipped with two-stroke low speed engines fueled by heavy fuel oil (HFO). In order to maintain constant pressure in cargo tanks, a re-liquefaction system had to be installed. Since 2015 systems equipped in DF SSD (dual fuel slow speed diesel) with highest heat efficiency of the power system have started coming into service [5].

The respond of the manufacturers of marine steam turbines to the dynamically changing market was the introduction of reheating plants operating according to the complex Clausius-Rankine cycle with increased parameters of superheated steam [1,7]. Efficiency of modern reheating steam plants are close to those of DFDE/TFDE, however lower than those with the highly heat efficient systems with low speed engines [2,8].

This paper attempts to identify the sources of waste energy of the widely applied conventional steam turbine (CST) plants in order to determine new directions in development for power systems aiming to increase heat efficiency of an LNG carrier power system.

2 Object and program of experimental studies

A thermodynamic analysis of a real cycle of a steam turbine power plant of an LNG carrier of 138 000 m^3 capacity was carried out basing on values obtained as a result of the operational experiment taking the measurement model as zero dimensional. The studied power plant whose main component particulars are listed in Tab. 1 consisted of a cross compound steam turbine, vacuum condenser, system of feed and condensate water including regenerative heaters and two steam boilers generating superheated steam of pressure $p_1 = 6100$ kPa and temperature $T_1 = 525\,^\circ$C.

2.1 Program and method of experimental research

The algorithm of carried out passive operational experiment is shown in Fig. 2. The procedure of the experiment consists of identification and analysis of machines and equipment realizing the thermal cycle of the power

Table 1: Particulars of conventional steam turbine (CST) plant main components.

CST plan of an LNG carrier capacity 138 000 m^3	
Main boilers	2 × KHI UME 65/50
Main turbine	KHI UA – 400 29080 kW at 90 rpm
Turbo alternators	2 × Shinko RG92-2 3450 kW 8145 rpm
Diesel generator	Wärtsilä 9R32LNE 3770 kW x 720 rpm
Feed water pumps	2 × Coffin Turbo DEB-16 180 m^3 at 8650 kPa

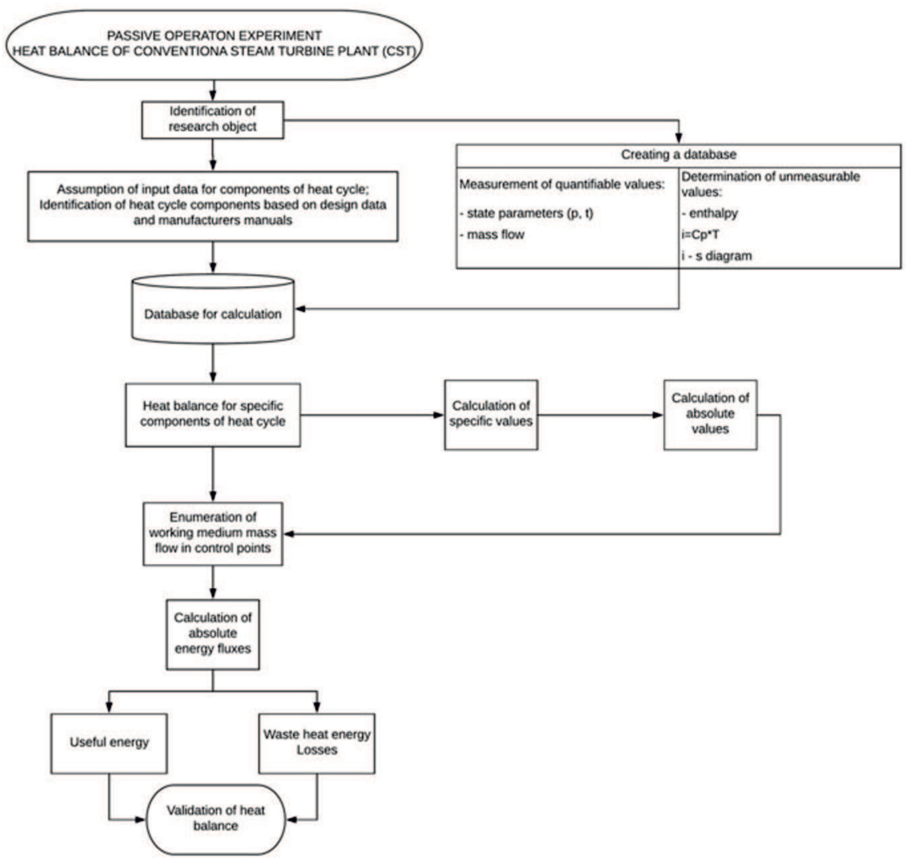


Figure 2: Algorithm of realization of the passive operational experiment.

system according to the thermal – flow diagram of the analyzed cycle shown in Fig. 3.

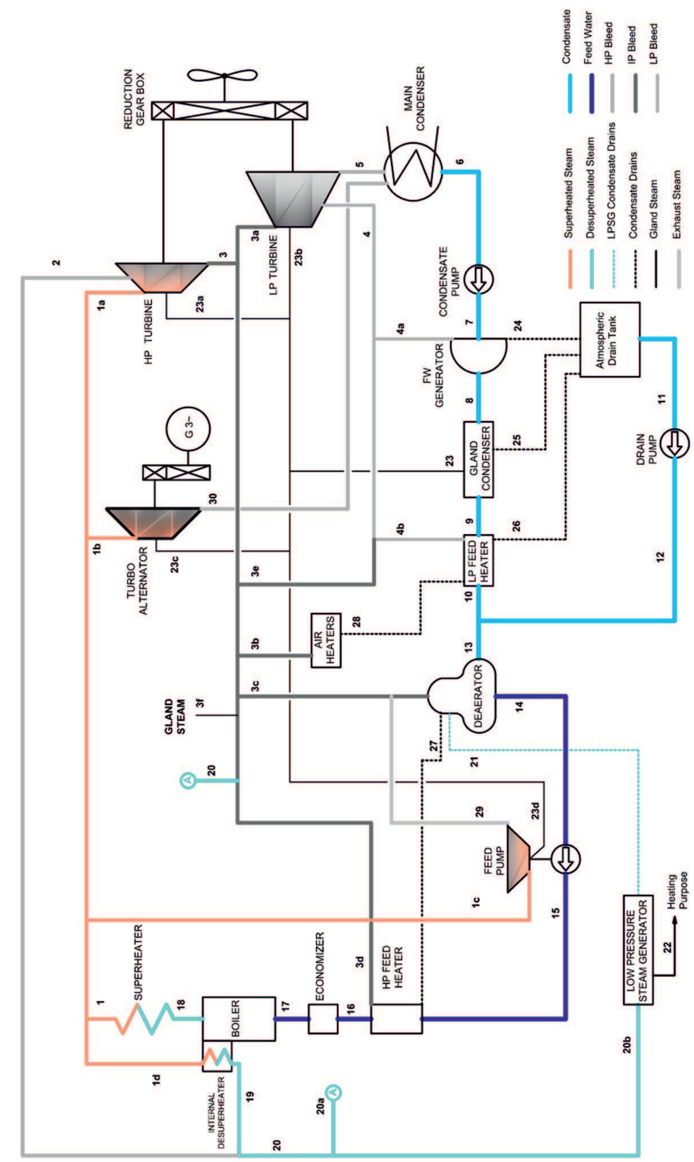


Figure 3: Thermal-flow diagram of the marine steam turbine propulsion plant of an LNG carrier.

An analysis was carried out basing on manufacturer's database, test results and delivery-acceptance reports of machines and equipment [11] as well as the values measured during operation for a chosen operational state maximum continuous rating (MCR). Measurable values were recorded using an integrated automation system (IAS), ship power management system (PMS) as well as additional equipment such as manometers, thermometers, flow meters, counters and exhaust analyzers. At the same time fluxes of enthalpy were determined as functions of the state of the working medium (Tabs. 2 and 3). These data were treated as variables in the heat balance in line with the algorithm shown in Fig. 2. As a result of calculations, values of fluxes of usable and waste energy were determined.

2.2 Thermal – flow diagram of the study object

On the basis of carried out identification of the power system and its technical documentation [11], a thermal-flow diagram of a complex thermal cycle of a real power turbine system was modeled which is shown in Fig. 3. The analyzed model consists of a steam boiler generating superheated steam with state parameters: pressure, temperature, and specific enthalpy (p_1, T_1, i_1 in Tab. 2). The steam supplies the main steam turbine, turbo generator and turbo feed water pump. There are three steam bleeds fitted on the main turbine. High pressure (HP) steam bleeds feed low pressure steam generator whose condensate is directed to the deaerator. Intermediate pressure (IP) bleed is located on the crossover pipe between the high pressure and low pressure turbine. The steam of the IP bleed is used to feed the high pressure feed heater, supplementing the system of exhaust steam from turbopumps of feed water, supplying steam air heaters, the system of gland steam and supplementing the shortage of steam in the low pressure bleed system. The low pressure bleed steam is used for feeding regenerative heaters of the first and third stage (condensate cooled fresh water generator and low pressure feed heater, respectively). The gland steam feeds the regenerative heater of the second stage (gland steam condenser). Condensates from the first three heaters are directed to atmospheric drain tank and are mixed with the main flux of the condensate before the deaerator. Condensate from the low pressure steam generator and from the high pressure heater are directed to the deaerator from which the feed water throughout the high pressure heater and economizer goes to the steam drum in the boiler. The exhaust steam from the low pressure turbine and turbo generator is condensed in the vacuum main condenser.

Table 2: Identification of main cycle components (input database for components).

Parameter	Symbol/Unit	Value	Temperature		Enthalpy	
			Symbol/Unit	Value	Symbol/Unit	Value
Boilers						
Superheated steam	p_{10} [kPa]	6100	T_1 [°C]	525	i_1 [kJ/kg]	3481
Desuperheated steam	p_{19} [kPa]	6080	T_{19} [°C]	288	i_{19} [kJ/kg]	2835
Efficiency on HFO	η [–]	0.879				
Calorific value	W_d [kJ/kg]	43040				
Air to fuel ratio O_2=2%	(λ=1.2) B	17.9				
Air heater						
Air temp in	T_{airin} [°C]	45				
Air temp out	T_{airout} [°C]	120				
Air heat capacity	c_p [kJ/kgK]	1				
Steam in	p_{3b} [kPa]	320	T_{3b} [°C]	245	i_{3b} [kJ/kg]	2943
Condensate out	p_{28} [kPa]	300	T_{28} [°C]	133	i_{28} [kJ/kg]	561
Main turbine						
Shaft power	P_e [kW]	29080				
Steam in	p_{1a} [kPa]	5950	T_{1a} [°C]	520	i_{1a} [kJ/kg]	3470
Exhaust steam	p_5 [kPa]	5	T_5 [°C]	33	i_5 [kJ/kg]	2294
HP bleed	p_2 [kPa]	1950	T_3 [°C]	372	i_2 [kJ/kg]	3186.2
IP bleed	p_3 [kPa]	660	T_3 [°C]	245	i_3 [kJ/kg]	2943
LP bleed	p_4 [kPa]	1.5	T_4 [°C]	131	i_4 [kJ/kg]	2734
Turbine eff. internal	λ_{iMT} [–]	0.874				
Turbine eff. mechanical	λ_{mMT} [–]	0.974				
G/B efficiency	λ_{GBMT} [–]	0.98				
Turbo generator						
Power	$Ne\ alt$ [kW]	1475				
Steam in	p_{1b} [kPa]	6100	T_{1b} [°C]	520	i_{1b} [kJ/kg]	3470
Exhaust steam	p_{30} [kPa]	5	T_{30} [°C]	35	i_{30} [kJ/kg]	2452
Turbine internal eff.	λ_{iTA} [–]	0.759				
Turbine mechanical eff.	λ_{mTA} [–]	0.97				
G/B efficiency	λ_{GBTA} [–]	0.98				
Alternator eff.	η_{altTA} [–]	0.96				
T/A eff. (overall)	η_{TA} [–]	0.266				
Feed pump						
Steam in	p_{1c} [kPa]	61	T_{1c} [°C]	520	i_{1c} [kJ/kg]	3470
Exhaust steam	p_{28} [kPa]	3	T_{29} [°C]	310	i_{29} [kJ/kg]	3100
Turbine internal eff.	η_{iFP} [–]	0.47				
Turbine mechanical eff.	η_{mFP} [–]	0.97				
Pump eff.	η_p [–]	0.6				
Head	H_z [kJ/kg]	7.47				
LPSG						
Capacity	M_{22} [kg/h]	2572				
LPSG steam	p_{22} [kPa]	900	T_{22} [°C]	175	i_{22} [kJ/kg]	2773
LPSG feed water	p_x [kPa]	1200	T_x [°C]	90	i_x [kJ/kg]	377
Heating steam in	p_{20b} [kPa]	1950	T_{20b} [°C]	372	i_{20b} [kJ/kg]	3186.2
LPSG condensate drain	p_{21} [kPa]	300	T_{21} [°C]	132	i_{21} [kJ/kg]	554
Air heater						
Air temp. In	T_{airin} [°C]	45				
Air temp out	T_{airout} [°C]	125				
Air heat capacity	c_p [kJ/kgK]	1				

Table 3: Overview of thermodynamic state parameters in the control planes.

Parameter State	Pressure A	Temp.	Enthalpy	Flow	Mass flow
	kPa	°C	kJ/kg	–	kg/h
1	2	3	4	5	6
1 – Superheated steam after boilers	6100	525	3481	1.0000	114901.40
1a – Superheated steam HP turbine in	5950	520	3470	0.9156	105199.73
1b – Superheated steam TA in	6100	520	3470	0.0497	5715.81
1c – Superheated steam feed pump in	6100	520	3470	0.0347	3985.85
1d – Superheated steam to internal desuperheater	6080	525	3481	0.0000	0.00
2 – HP bleed	1950	372	3186.2	0.0214	2457.38
3 – HP turbine exhaust	660	245	2943	0.8942	102742.35
3a – LP turbine steam in	660	245	2943	0.7921	91016.50
3b – IP bleed to air heaters	320	245	2943	0.0437	5020.21
3c – IP bleed to deaerator	300	245	2945	0.0188	2158.21
3d – IP bleed to HP heater	660	245	2943	0.0362	4157.78
3e – IP bleed make up to LP bleed	660	245	2943	0.0000	0.00
3f – IP bleed to gland steam	660	245	2943	0.0034	389.65
4 – LP blead	150	131	2734	0.0838	9627.88
4a - LP blead to FW generator	150	131	2734	0.0209	2403.38
4b – LP bleed to LP Heater	150	131	2734	0.0629	7224.50
5 – Exhaust steam from LP turbine	6.6	38	2294	0.7083	81388.62
6 – Condensate from main condenser	5	33	138	0.7581	87104.43
7 – Condensate FW gen. in	1000	33	138	0.7581	87104.43
8 – Condensate gland condenser in	1000	48.5	204	0.7581	87104.43
9 – Condensate LP heater in	1000	51	215	0.7581	87104.43
10 – Condensate LP heater out	1000	102	427	0.7581	87104.43
11 – Drains from ADT	100	90	376	0.1309	15037.75
12 – Drains from ADT pump out	1000	90	377	0.1309	15037.75
13 – Condensate deaerator in	1000	95	398	0.8890	102142.18
14 – Feed water deaerator out	300	131	550	1.0000	114901.40
15 – Feed water HP heater in	7500	131	555	1.0000	114901.40
16 – Feed water HP heater in	7500	151	640	1.0000	114901.40
17 – Feed water economizer out	7500	230	990	1.0000	114901.40
18 – Saturated steam boiler drum out	6500	280	2778	1.0000	114901.40
19 – Desuperheated steam boiler out	6080	288	2835	0.0000	0.00
20 – Desuperheated steam	1950	372	3186.2	0.0214	2457.38
20a – Desup. steam make up to IP bleed	1950	372	3186.2	0.0000	0.00
20b – LPSG heating steam in	1950	372	3186.2	0.0214	2457.38
21 – Drains from LPSG	300	132	554	0.0214	2457.38
22 – Aux. heating steam	900	175	2773	–	–
23a – HP turbine gland steam out	70	190	2857	–	–
23b – LP turbine gland steam out	70	190	2857	–	–
23c – TA gland steam out	70	190	2857	–	–
23d – Feed pump gland steam out	70	190	2857	–	–
23 – Gland steam to gland condenser	70	190	2857	0.0034	389.65
24 – FW generator drains	100	82	342	0.0209	2403.38
25 – Gland condenser drains	100	95	398	0.0034	389.65
26 – LP heater drains	100	80	335	0.1066	12244.71
27 – HP heater drains	380	142	594	0.0362	4157.78
28 – Air heaters drains	300	133	561	0.0437	5020.21
29 – Feed pumps exhaust steam	300	310	3100	0.0347	3985.85
30 – TA exhaust steam	7.5	40	2452	0.0497	5715.81

3 Heat balance of a complex steam cycle

A complex thermal-flow model of a power system (Fig. 3) together with the values listed in Tab. 2 and in columns 2, 3, and 4 of Tab. 3 were used to determine the fluxes of steam and condensate masses in particular control planes of the system. A calculation algorithm is shown in Fig. 4.

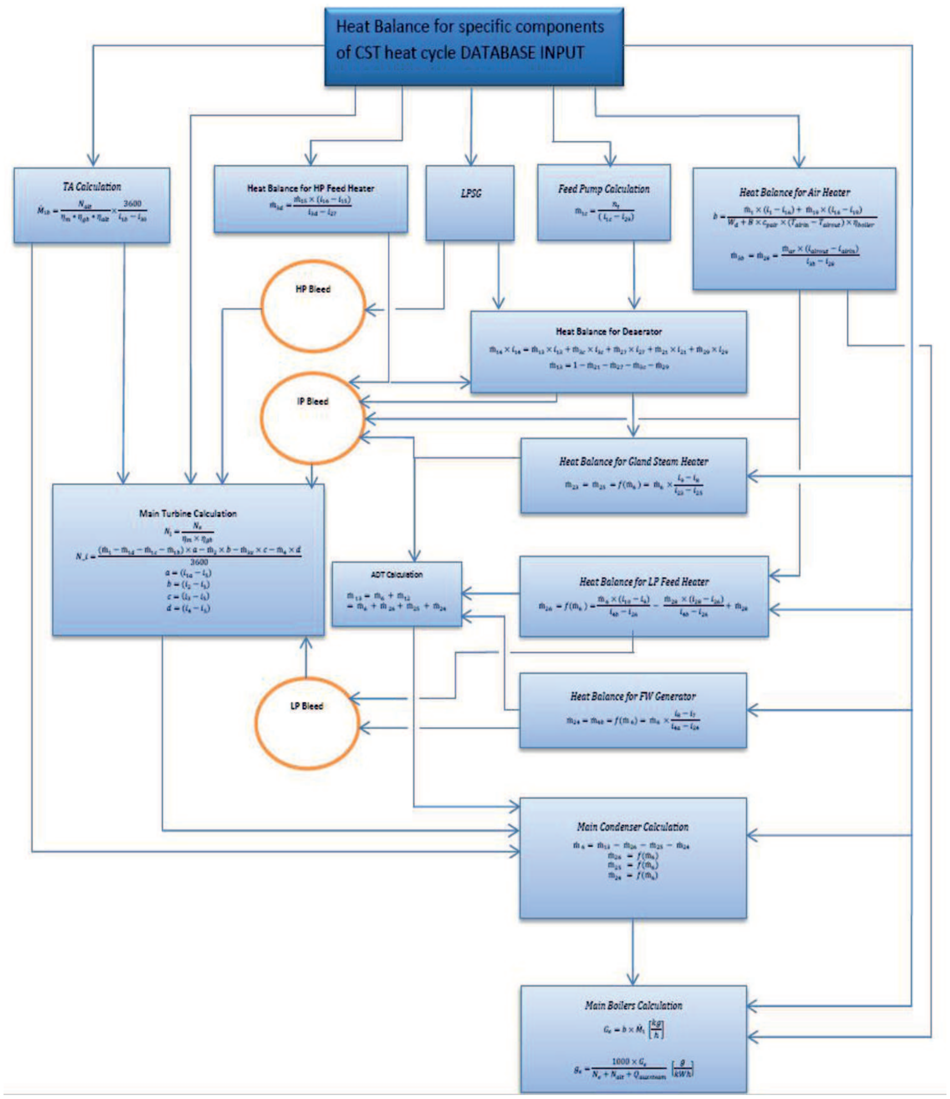


Figure 4: An algorithm of heat balance calculations.

Calculations of thermal balance were performed for specific values of mass flow rate ($\dot{m}_{15} = \dot{m}_1$), which enabled determination of steam required by the turbine of the feed water pump and the low pressure steam generator. Specific mass flow rate (Tab.3 column 5) referred to one kilogram of superheated steam generated in a boiler. Using the values determined as a result of the experiment, a set of conservation equations was solved for:

- deaerator (energy balance + balance of mass fluxes),
- air heater, determining the air flux required for $\lambda = 1.2$ and unitary requirement for heating steam.

Basing on the equations for the low pressure feed heater, gland steam condenser and fresh water generator versus condensate mass of the main condenser, a system of four equations was constructed, thanks to which mass fluxes of steam bleeds were determined. For the assumed requirement for electrical energy, hourly requirement for superheated steam of the turbo generator was calculated. Determination of requirement for steam for the main turbine and the whole power plant took place as a result of balancing the magnitudes of fluxes of steam bleeds, hourly requirement for live steam for the turbine generator and the assumed value of the power of the main turbine. The obtained results of calculated values of fluxes were listed in Tab. 3 in columns 5 and 6 which were the basis for determining the values of fluxes of usable and waste heat energy and heat losses of the plant. Results of these energy fluxes determined in the balance are listed in Tab. 4.

4 Quality assessment of waste heat energy

The identified values of energy fluxes in the context of their applicability as usable energy require evaluation of their usability for performing work and in consequence performing a transportation task of an LNG carrier. Only the quality of exhaust steam fluxes of the main power turbine and turbo generator and exhaust flux from the main steam boiler have been analyzed. Mechanical and flow resistance fluxes were considered technologically insignificant due to their small values as well as because of complexity of their identification and evaluation, whereas possibilities to lower them are closely connected with the design of the working elements such as geometry of profiles and blade rims, bearings of turbine, construction of their casing and their cooling methods together with installations, piping and fittings [1].

Table 4: Heat balance for CST plant at 100% MCR.

Heat balance for plant at 100% MCR (29080 kW at shaft speed 90 rpm)									
	Medium	Flow	Press. absolute	Temp.	Enthalpy	Energy		Specific heat cons.	Percent.
		kg/h	kPa	°C	kJ/kg	kJ/h	kW	kJ/kWh	%
MT useful energy	Mechanical energy	–	–	–	–	104688000.0	29080.0	3600.00	29.2
TA useful energy	Electrical energy	–	–	–	–	5310000.0	1475.0	182.60	1.5
Aux steam useful energy	Heat energy – Steam	2572.0	900.0	175	2773	6468314.2	1796.8	222.43	1.8
MT condenser losses	Heat energy – Steam	81388.6	6.6	38	2294	175473867.3	48742.7	6034.18	49.0
TA condenser losses	Heat energy – Steam	5715.8	7.5	40	2452	13226381.8	3674.0	454.83	3.7
Exhaust losses	Heat energy Exhaust gases	157827.5	> Atmos.	155	285	44935857.3	12482.2	1545.20	12.5
MT mechanical losses	Friction/Heat	–	–	–	–	2851577.8	792.1	98.06	0.8
TA mechanical losses	Friction/Heat	–	–	–	–	174560.8	48.5	6.00	0.05
FP mechanical losses	Friction/Heat	–	–	–	–	44243.0	12.29	1.52	0.01
MT gearbox losses	Friction/Heat	–	–	–	–	2136489.8	593.5	73.47	0.6
TA gearbox losses	Friction/Heat	–	–	–	–	112882.6	31.4	3.88	0.03
Flow losses in pipe lines	Heat/Flow restriction	–	–	–	–	1263915.3	351.1	43.46	0.4
FP pump losses		–	–	–	–	572208.9	158.9	19.68	0.16
TA alternator losses	Resistance/Heat	–	–	–	–	221250.0	61.5	7.61	0.06
SUB TOTAL						357479548.9	99299.9	12292.92	99.81

To evaluate the quality of waste energy fluxes the following state functions were used [3,10]:

- enthalpy:

$$i = c_p T \tag{1}$$

and

$$i = u + pV \ , \tag{2}$$

- physical exergy:

$$b_{steam} = i_{steam} - i_0 - T_0 \left(s_{steam} - s_0 \right) \tag{3}$$

or

$$b_{exh} = c_{pexh} \left(T_{exh} - T_0 \right) - T_0 c_{pexh} \ln \frac{T_{exh}}{T_0} \ , \tag{4}$$

- temperature coefficient of energy quality:

$$\psi_T = \frac{T_{source} - T_0}{T_{source}} \ , \tag{5}$$

- exergy coefficient of energy quality:

$$\psi_{b/\Delta i} = \frac{b}{\triangle i} \ , \tag{6}$$

- mass flow of waste energy sources.

Values of the defined state functions are listed in Tab. 5. To ensure comparability of the results, the obtained energy quality coefficients required defining parameters of reference states for each source of waste energy (related to environment conditions). For the reference state of the exhaust steam, parameters of the medium in the main condenser were taken (p_{steam0} = 5 kPa (absolute), T_{steam0} = 33 °C, and the determined condensate enthalpy i_6 = 138 kJ/kg). For the waste energy flux carried by exhaust gases from the boiler the reference state is determined by the following parameters: p_{exh0} = 102.5 kPa (atm.), T_{exh0} = 30 °C and enthalpy i_{exh0} = 30 kJ/kg.

For the assumed reference states and those determined during the operational experiment and calculated from propulsion plant heat balance, values of physical exergy (relations 3 and 4) were determined for particular carriers of waste heat energy as well as coefficients of energy quality (relations 5 and 6), Tab. 5.

Table 5: Determined functions of evaluation of the waste energy source quality.

Losses	Mass flow	Energy flux	Press. abs.	Temp.	Enthalpy	x	Exergy	Ψ_t	$\Psi_{b/\Delta i}$
	kg/h	kJ/h	kPa	°C	kJ/kg	–	kJ/kg	–	–
MT condenser losses	81388.60	175473867.3	06.6	38	2294	0.888	1926.40	0.132	0.8936
TA condenser losses	5715.81	13226381.8	7.5	40	2452	0.950	2069.70	0.175	0.8945
Exhaust losses	157827.50	44935857.3	10.5	155	285	–	139.25	0.806	0.5460

5 Summary and conclusions

The determined energy quality indicators, namely the temperature one ψ_T and the exergy one $\psi_{b/\Delta i}$ for exhaust gases point out to a high potential of this source. There are both a high temperature difference ($T_{exh} = 155\,^\circ$C, $T_0 = 30\,^\circ$C) as well as a considerable energy flux (about 12.5% of the energy introduced into the system). Usability of energy contained in the exhaust gases is decided by the maximum cooling temperature of exhaust on the outlet of economizer. That temperature due to condensation of sulphur compounds contained in the fuel has to be correlated with the acidic dew point, however, it cannot be lower than 140–150 °C, depending on sulphur content in the used fuel [4,9]. That criterion significantly limits the possibilities of using waste energy contained in exhaust gases. In that case it would be necessary to use fuel with lower sulphur content. Higher degree of exhaust cooling would be possible in the case of boilers fueled by LNG.

The determined values of physical exergy (b_{steam}) as well as the exergy coefficient of energy quality ($\psi_{b/\Delta i}$) for the exhaust steam from the main turbine as well as from the turbo generator unit point to a very high energy potential of these fluxes. However, due to low energy state, a small temperature difference and high dispersion of the exhaust steam heat, direct use of this heat in a classical ship heat exchanger (with partitions between the heating medium and a medium receiving the heat) is not possible.

The obtained and presented results are technical hints indicating rational utilization of the identified waste heat in the process of mixing fluxes, and therefore the technical possibilities of realizing regenerative steam-water cycle in which heating of the feed water would mainly recover the latent heat of exhaust steam form the main turbine could be considered. The use of a system using regeneration ejectors, recovering latent heat (condensation heat from the main condenser), would allow to reduce the demand for the bleed steam system, what will results in an increase of the toal available enthalpy drop. The power of the turbine power unit would also be increased or the demand for superheated steam and fuel would be reduced.

Received 13 December 2018

References

[1] Adamkiewicz A.: *Technological development possibilities of heat turbines in LNG carrier power systems.* In: Steam Turbines: Theory. Construction. Operation. Wydawn. Politechniki Śląskiej, Gliwice 2016, 13–23 (in Polish).

[2] Adamkiewicz A., Grzesiak S.: *Evolution of energy efficiency of modern lng carrier's steam turbine propulsion plant.* Rynek Energi 130/3(2017), 92–98 (in Polish).

[3] Adamkiewicz A., Michalski R., Zeńczak W.: *Selected Problems of Energy Conversion Technologies in Marine Power Engineering Systems.* Kaprint, Lublin 2012 (in Polish).

[4] Gawron M., Przybyła K., Lipnicki Z.: *Exhaust Gas Flow Conditions in the Chimney and Acid Dew Point.* Zeszyty Nnaukowe Uniwersytetu Zielonogórskiego **144**(2011), 48–59 (in Polish).

[5] Grzesiak S.: *Alternative propulsion plants for LNG Carriers.* New Trends in Production Engineering **1**(2018), 1, 399–407.

[6] IGU IGU World LNG Report (2017).: http://www.igu.org (accessed May 2017).

[7] Ohira H., Hiramatsu, S., Matsumoto S., Fujino Y.: *Key technologies for Mitsubishi LNG carrier.* MHI Tech. Rev. **44**(2007), 3, 1–4.

[8] Petel M., Nath N.: *Improve steam turbine efficiency.* Hydrocarb. Process. **79**(2000), 6, 85–86, 88, 90.

[9] Stechman A.: *Selected issues in acid corrosion of the flue pipes – calculation of the acid dew point of exhaust gases.* Prace Naukowe Instytutu Budownictwa Politechniki Wrocławskiej, Wrocław 2008 (in Polish).

[10] Szargut J. Ziębik A.: *Fundamentals of Thermal Engineering.* WNT, Warszawa 1998 (in Polish).

[11] Technical documentation of an 138k m^3 capacity LNG carrier:
– Piping diagram of engine room.
– Heat balance for cooling water.
– Steam balance for auxillart system.
– On board test procedure with results for engine room auxillary machinery.

– On board test procedure with results for main boiler.
– On board test procedure with results for engine room for main steam turbine.
– Main propultion turbine (Kawassaki UA-400) Vol. 1,2.
– Main propulsion turbine instruction manual Vol. 1-3.
– Steam turbine for electric generator (Shinko RG92-2).
– Feed water pump & steam turbine (Coffin Turbo DEB-16).
– Main Boiler machinery part (Kawasaki UME 65/50) Vol. 1-2.
– Main Boiler automation part (Kawasaki UME 65/50) Vol. 1-2.
– Instruction for operation & maintenace (Main boiler)
(Kawasaki UME 65/50) Vol. 1-3.
– Steam to steam generator (Dongwa BKU).
– Electric load analysis.
– Vessel particulars.