ISSN 1407-8015

TRANSPORT AND ENGINEERING
MAŠĪNZINĀTNE UN TRANSPORTS

2008-8015

INTELLIGENT TRANSPORT SYSTEMS
INTELEKTUĀLĀS TRANSPORTA SISTĒMAS

HEAT RECOVERY POSSIBILITIES FOR ADVANCED SLOW SPEED DIESEL ENGINE POWER PLANTS

A.Zāģeris, *Engineer-Ships' Surveyor*
Det Norske Veritas Latvia SIA
Kronvalda bulv. 10, Rīga, LV-1010, Latvia,
tel. 371-7325010, fax: 371-7356270,
e-mail: Arnis.Zageris@dnv.com

J.Cimanskis, *Dr.Hab.Sc Ing..,*
Latvian Maritime Academy,
Flotes st., 5B, Rīga, LV-1016, Latvia,
tel. 371-7432091, fax: 371-7432091,
LMA e-mail: j.cimanskis@livas.lvv

Key words: Slow speed diesel engines, exhaust boiler, Complex Waste Heat Recovery System (CWHRS), Ambient conditions

1. Introduction.

In this article we will investigate waste heat, e.g. exhaust gas, cooling water, scavenging air, recovery possibilities for advanced low speed diesel engines due to its wide application in marine industry. During the years of engine technical development diesel efficiency has been approaching its theoretical upper limit by reducing all possible heat losses, and first of all in exhaust gasses (see Fig.1). Nevertheless, still unused heat of exhaust gases is the main constituent [1, 2, 3], recovery of which may considerably contribute additional efficiency increase of the whole power plant despite of its low potential. Taking into consideration both fuel price continuous and high growth, and adoption of more stringent International Requirements for environment protection (MARPOL Annex VI), the recovery of

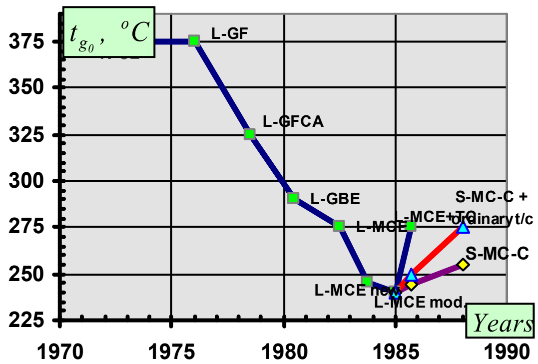


Figure 1. Tendency of exhaust gas temperature changes at boiler inlet with development of low speed engine modification for B&W Manufacturer at ISO ambient conditions.

cooling water heat is another source of further economically beneficiary savings.
Based on assumptions above, following main goals of our explorations might be declared:

1. to ensure ship demands in electricity and low-potential heat (by means of saturated steam consumption) via the deepest gas cooling in a steam exhaust boiler, ie. to get the lowest possible gas temperature at boiler outlet $t_{g_{exh}}$,
2. the effective use of cooling water heat for boiler feed water warm-up and low-potential heat supply, substituting and reducing saturated steam consumption;
3. with a consideration of said above, it is important to evaluate the rate of ship's provision in electricity and heat in dependence on main engine (advanced slow-speed diesel) load level and sailing regions (ambient conditions),
4. finally, based on research results it should be possible to estimate the effective ways of either compensation of electricity/heat shortage or its surplus use.

2. Main input conditions.

Exhaust gas heat recovery is ensured by means of Waste Heat Recovery System (WHRS) consisting of exhaust boiler (EB), where superheated steam is directed to steam turbine driving alternator and producing electricity for ships demands (Fig.2) [1, 2, 3]. Saturated steam is used
for low potential heat consumers, e.g. fuel oil preheat, bunker, accommodation, etc. The lowest level of steam pressure p_s in boiler is recommended to be not less than 7 bars [4, 5, 6] to ensure heavy fuel oil (IFO 380) preheating at main engine (ME) inlet as well.

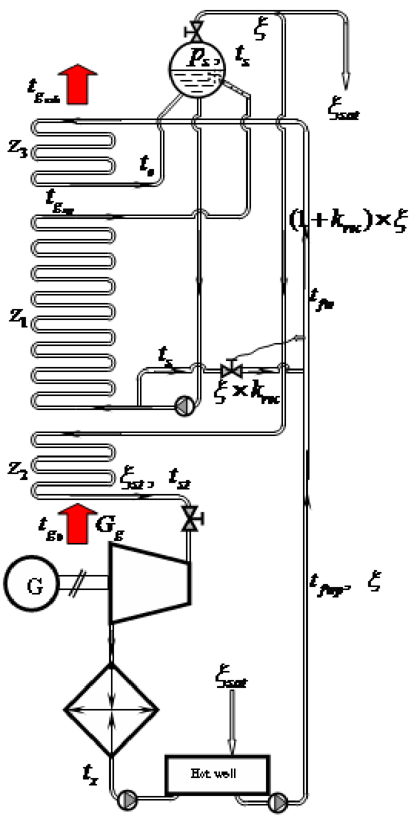


Figure 2. Waste Heat Recovery System with thermostatic mixing valve.

For advanced slow speed diesel engines exhaust gas heat recovery is problematic due to its low potential at reduced and pre-determined meanings of gas temperature at main engine outlet, ie. exhaust boiler inlet t_{g_0}, ^oC. Then the main way how to increase waste heat recovery system (WHRS) (see fig.2) efficiency is to ensure the deepest gas cooling in a steam exhaust boiler, id. to get the lowest possible gas temperature at boiler outlet $t_{g_{exh}}$, which in its turn has some limitations due to possible acid corrosion of tail surfaces and accepted thermodynamic parameters [1, 7]. On the other hand gas overcooling is also limited by required lowest level of steam pressure for chosen single stage heat recovery circuit. So based on recommendations, maintenance experience and design practice it is recommended that lowest flue gas temperature at exhaust boiler to be maintained at the level of $t_{g_{exh}} \geq 160^o C$ [4, 8]. Boiler feed water warm-up till around $t_{fw} \geq 120^o C$ [1, 7, 9] is carried out by means of recirculation, allowing to achieve effective gas heat recovery at safe operation conditions.

To ensure further increase of WHRS it is recommended that feed water pre-heat is carried out by means of cooling water of either jacket ($t_{cyl} \approx 80 \div 90^o C$) one or scavenging air ($t_{cyl} \approx 105 \div 110^o C$) or both of them till temperature $t_{fwp_{II}} \approx 100^o C$, thus reducing required amount for recirculation $k_{rec} \downarrow \approx 3{,}6 \times$ (see Fig.3) and increasing boiler steam output. In order to reduce saturated steam consumption $\xi_{sat}, \frac{kg \;\; steam}{1 \;\; kg \;\; gases} \downarrow$ by substituting of cooling water consumers of low-potential heat are to be defined and might be divided in two groups:

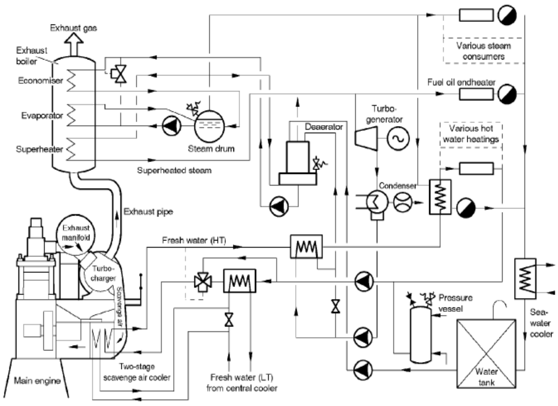


Figure 3. Heat Recovery system layout.

A. Consumers of heat being independent on ambient conditions with required warm-up temperature as follows (see Fig.4) [7, 8, 9]:
   A.1. Fuel oil preheat at ME inlet, which could be ensured only by means of saturated steam as required temperature might be up to $130 \div 150^o C$;
   A.2. Fuel oil preheat at separator inlet and could be covered by turbocharger scavenging air heat as required temperature is around $70 \div 90^o C$;
   A.3. Lubricating oil preheat at separator inlet and possible heat supply could be arranged as for consumer #2 or even by jacket cooling water heat could be a source of supply as required temperature level is lower, i.e. $\leq 70^o C$;

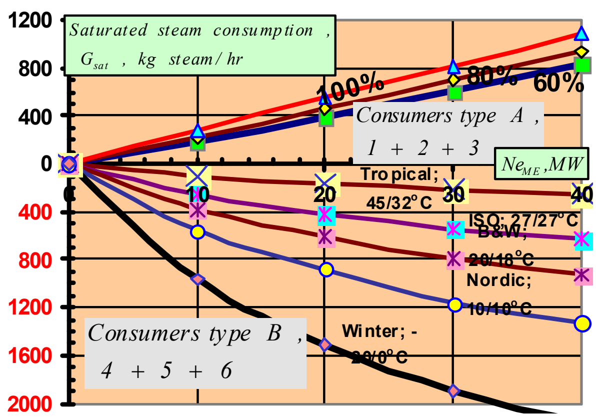


Figure 4. Saturated steam consumption on main engine output @ various ambient conditions for different consumers of low potential heat.

B. Consumers of heat being dependent on ambient conditions, id. outlet temperature as follows:
   B.4. Bunker (and settling) tanks, etc.
   B.5. Service tanks
   B.6. Accommodations, laundry/washing water and etc.

Heat consumers ##4, 5, 6 could be covered either by jacket cooling water heat or turbocharger scavenging air heat, when it is whether available and/or required. Definitely such extensive heat recovery comes to complexity in heat exchange system (see Fig.3), requiring additional investments and electrical supply either. Hence, briefly summarizing said above our plan of action targeted on ships' power plant efficiency increase via waste heat recovery (turbo-generator power output growth) consists of following steps:

1. flue gas heat recovery in exhaust boiler (see Fig.2), supplying electro-energy by steam turbo-generator and low potential heat;

2. feed water preheating from $50^o C$ till $\approx 100^o C$ by;
   2.1. jacket cooling water till $70^o C$ and ensuring additional growth in turbine output equally to value ΔNe^{3}_{TG} followed by
   2.2. turbocharger air cooling water till $100^o C$ thus reducing economizer part in boiler and coming to growth in both steam and turbine output $\Delta Ne^{t/c}_{TG_{\Delta t}}$;
3. Reduction in saturated steam consumption by it substituting with cooling water (jacket and turbocharger) heat for relevant consumers (see Fig.5), and as a result another additional turbo-generator output increase by value $\Delta Ne^{2_\xi}_{TG}$ or $\Delta Ne^{t/c}_{TG_\xi}$.

Such an extensively developed system is to be called as Complex Waste Heat Recovery System (CWHRS) and based on considerations above following main tasks should be explored in this article:

1. Efficiency outcome due to exhaust gas and cooling water heat recovery extent;
2. Ambient condition ($t_{s/w}$ -sea water and t_a - ambient air temperatures) impact on both main engine output dates (flue gas temperature t_{g_0} and amount G_{g_0}) and low potential heat consumers is significant (Fig.54);
3. Main engine (advanced slow speed diesel engines) type ($t_{g_0} = var.$) output ($G_{g_0} = var.$) and work load impact;
4. Electrical and heat shortage determination and compensation.

3. Effective utilization of electrical energy.

The main efficiency index of CWHRS is turbo-generator output; and especially its utilizing extent by ship consumers. What does it mean? Of course, first of all we are interested in gaining maximum output by CWHRS; and in case of heat/electrical energy surplus we will have safety margin enough to run unit in off-line operation regime at different load and ambient conditions with the consideration of equipment wear and tear impact. At the same time from economical point of view we are interested in utilizing all recovered energy, what depends on power plant arrangements, e.g. parallel operation of both main engine and steam turbo-generator either on shaft via reduction gear or main switchboard [5, 9]. But shortage in heat/electricity is more problematical event, which should be effectively compensated, as in case of absence such technical sources it might come to CWHRS disabling and considerable loss in effect of recovered energy.

Although the problem above is utmost important, nevertheless it is not the main core of our topic to analyze and compare the effective ways how to either compensate or utilize heat/electricity shortage and surplus, but without mentioning the problem and defining the possible solutions our investigations would not be considered as accomplished ones. Therefore different technical solutions will be briefly described only and proposed.

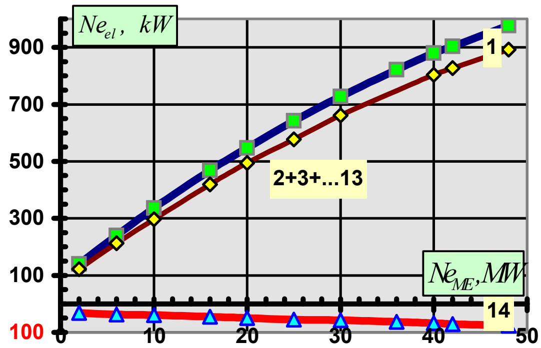


Figure 5. Electro-power station demands in electricity on main engine output for different consumers.

First of all, the level of electrical consumption is to be determined. All ship electrical consumers could be divided in two groups (see Fig.5):

A. Consumers being independent on ambient conditions, e.g. **#1-** cooling pumps for diesel generators and ancillaries for turbo-generator, **##2-13** - ME pumps (cooling, fuel, lubricating-oil), hydraulic pumps and equipment, navigation equipment, lighting, navigation equipment, etc.;

B. **#14 -** Consumers being dependent on ambient conditions, e.g. air conditioner of accommodations [2, 3].

Of course, presented values might significantly differ with the equipment saturation, being dependable on ships' type, but for most cases these figure are realistic. As a rule electricity shortage is the issue for our investigated power plants; and the most common way how to effectively solve the problem is introduce auxiliary boiler in the system and use it as steam drum during WHRS operation [3, 7]. When steam pressure is dropping below some lowest limit, burner will start to run in order to maintain required steam consumption by turbo-generator. There could be also other means of auxiliary boiler governing, what is not an issue of the topic, but such an arrangement have wide application; and it is also required during either ship's stay or operation at low load level; and it economical beneficence especially at low level of either electricity or heat shortage.

Taking into consideration the fact, that oxygen content in exhaust gases is around 16%, it would be possible to arrange additional fuel burning. Such arrangements were introduced on m/v "K.Ciolkovskis" (Latvian Shipping Co.) with some good experience.

With the growth of heat/electricity shortage level auxiliary boiler back up will be inefficient and parallel work with diesel generator should be considered as another alternative. There could be two main options:

A. Parallel operation with diesel of the same output. Such an arrangement simplifies the electrical power plant set up during new-building stage and future maintenance. But on another hand even at low level of shortage power distribution amongst diesel and turbine would be so, that it comes to un-complete recovered energy use, thus artificially reducing CWHRS efficiency.
B. Parallel operation with diesel of the reduced output, so-called peak-engine. The level of possible power turbo-generator shortage should be evaluated with the consideration of wide range of different factors, e.g. ME load level, ambient conditions; and the choice of peak-diesel type/modification should be so, that <u>safe</u> parallel operation is ensured during as long as possible the voyage time. In this case diesel engine governor line should be more astatic, when running together with steam turbine. When diesel-engine is running alone, governor is to be readjusted. At the same time in order to save both initial and maintenance costs for diesel electrical plant peak engine output should be enough to serve at some specific conditions of the ship, e.g. during either ship's stay without cargo operations or specific loading conditions in parallel with base diesel generator. It would be also benefit to have the engines of the same modification, as in this case we can reduce the amount of base diesels for electrical power plant by introducing smaller peak-engine.

Another alternative could be also shaft generator application and possible parallel operation with turbo-generator with different arrangement of engagement [2, 3].

Also gas by-pass around turbocharger could be considered as effective solution despite of SFOC (specific fuel oil consumption) drop down, thus increasing gas temperature at boiler inlet. During new building stage the *high efficiency turbocharger* could be substituted with a *conventional* one. Then the amount of air required for combustion purposes can, however, be adjusted to provide a higher flue gas temperature, if this is needed for the exhaust gas boiler. The matching of the engine and the turbo-charging system is then modified, thus increasing the exhaust gas temperature by 20°C. This modification will lead to a 7-8% reduction in the exhaust gas amount, and involve an SFOC penalty of up to 2 g/BHPh [1, 7].

Also it could be possible to find the ways how to reduce electricity consumption level, e.g.: different mounted on ME service pumps; central cooling system arrangement by introducing central scoop type sea-water heat-exchanger.

4. System options.

In our investigations it is explored complex waste heat recovery system (CWHRS) of exhaust gases and cooling water, ie. jacket and turbocharged air one by means of high temperature stage (HTS) (see Fig.4). Such a complex combination (CWHRS +HTS) is accompanied by both higher investment and possible maintenance costs, but it also ensures the best efficiency of power plant in total. In some cases this type of upgrading might be economically unattractive, especially for ships already in operation; therefore following options are considered for comparison:

Option # 1 - "CWHRS+HTS (+ t/c air)", when the highest steam turbo-generator output Ne^{1}_{TG} is delivered at certain ambient conditions and ME load level;

Option # 2 - "CWHRS", when only heat of exhaust gasses and jacket cooling water is recovered. The performance of heat consumers ##4, 5, 6 are ensured by jacket cooling water only, as well as feed water preheating till $70^{o}C$ after hot well. In a result steam turbo-generator output Ne^{2}_{TG} will be lower than as above;

Option # 3 - "WHRS+$70^{o}C$", when jacket cooling water heat is used only for feed water preheating till $70^{o}C$. Such a modification is easy for implementation even for ships already in operation and steam turbine output will be equal to value Ne^{3}_{TG};

Option # 4 - "WHRS", when only flue gas heat is recovered and turbo-generator output Ne^{0}_{TG} is the lowest one as well as investments.

Based on these considered options efficiency of system complexity, i.e. recovery rate of ME cooling water heat, could be presented by a value of additional relative increase in steam turbo-generator output $K_{\Delta\eta} = \dfrac{Ne^{1,2,3}_{TG} - Ne^{0}_{TG}}{Ne^{0}_{TG}} \times 100\%$. Such an optional division is based not only on different complexity of the system, but it is also dependent on the fact, that both cooling water heat availability and heat consumer demands are very much affected by both ambient conditions and ME load level. Therefore it might come that *CWHRS+HTS (+ t/c air)* will transform into *WHRS* only.

5. Ambient condition and ME load level influence.

Based on suggestions above it was considered three main ambient conditions, which would cover the whole range of both sailing area and period, in our investigations. Also main engine load level has a significant influence on gas dates, especially on flue gas amount, therefore 60% of Maximum Continuous Rating (MCR) is considered as the lowest reasonable level to be investigated, based on service experience. Main engine modifications are reflected via inlet gas temperature different level, being also considered in our investigations.

6. ISO Ambient Conditions.

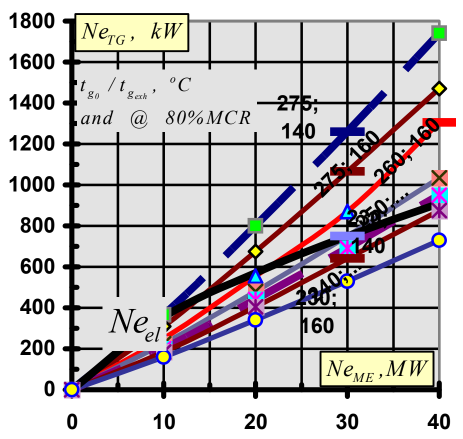
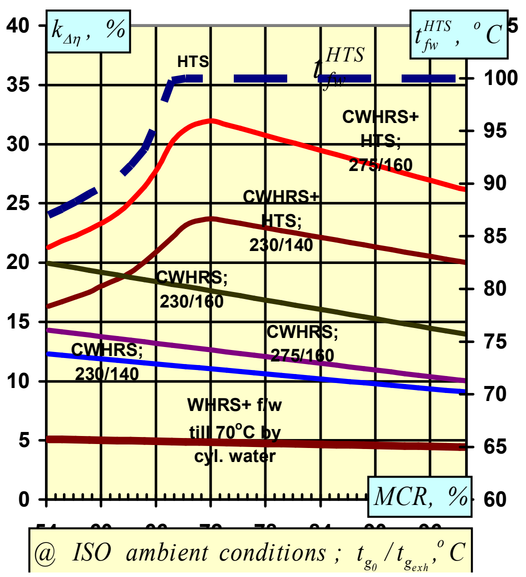


Figure 6. Turbo-generator output recovered by CWHRS in dependence on ME output/ modification @ ISO ambient conditions & MCR=80%.

In this sailing area sea water and ambient air temperatures are equal as follows: $t_{s/w} = 27^{o}C$ & $t_{a} = 27^{o}C$. Based on investigations, around $28 \div 37\%$ of sailing time lies in this weather zone, which characterizes with high gas temperatures at low demands in consumption of heat (saturated steam, cooling water heat). Low potential heat consumers type B ##4, 5, 6 could be provided by jacket cooling water heat only in the whole region of working load levels of ME, i.e. from 100% till 60% of MCR. Next-important is effective utilization of turbocharged air cooling water heat, which is much more dependent on ME load level, being described further. Despite the fact that ISO ambient conditions are most favorable ones in order to obtain the highest turbo-generator output, still the main engine output should be high enough to ensure autonomous "CWHRS + HTS" performance, when $Ne_{ME} \geq 15 \div 20MW$ @ $80\%MCR$ (see Fig.6) with the consideration of main engine modification and the fact, that electrical consumption level is the highest one due to necessity to run air-conditioner. As it was mentioned above main engine load level has direct impact on efficiency indices of the performance of CWHRS. If flue gas temperature has some slight increase, then gas amount reduces significantly, being close in direct ratio on MCR as well as recoverable heat amount from turbocharger air cooling water (see Fig.7a, b). At ≈72% of MCR required heat for consumers type A ##2, 3 exceeds available one thus gradually increasing saturated steam consumption At load level of $\approx 67{,}5\%MCR$ there is not available amount this heat even to warm up feed water till desired maximum temperature,



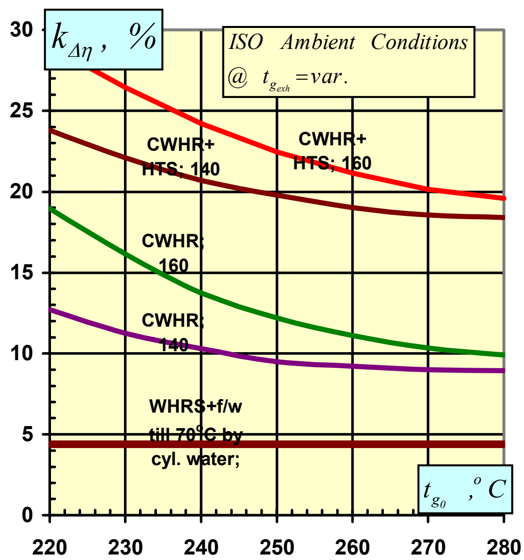


Figure 7a Figure 7b

WHRS Relative thermo efficiency increase rate dependence on **Figure 7a** ME load level and **Figure 7b** inlet gas temperature due to various complexity of ME cooling water heat recovery @ ISO ambient conditions for different ME modifications & gas cooling rate.

of thus increasing water amount for recirculation $k_{rec}\uparrow$,cooling water temperature starts drop down and at level 60% of MCR feed water temperature before thermostatic mixing valve will drop down 90^oC. So gradually with the lowering of ME load level "CWHRS + HTS (+ t/c air)" simplifies into CWHRS only. Ultimate recovery of available cooling water heat ensures the highest additional net gain $K_{\Delta\eta}$ around $25 \div 34\%$ for "CWHRS+HTS (+ t/c air)", while for "CWHRS" only this relative increase constitutes around $12 \div 18\%$ (see Fig.7). In most simplest case, when only boiler feed water is warmed up $70^o C$ before thermostatic mixing valve by jacket cooling water, this efficiency gain is equal to $\approx 5\%$. At reduced ME load levels value $K_{\Delta\eta}$ has a tendency to grow up, which could be interpreted as below:

- Specific weight of so called "gas factor" reduces $Ne^0_{TG} \downarrow$ due to direct dependence on MCR level,
- While low potential heat consumer demands remain without changes (see Fig.4), being fully met by cooling water heat, as well as power constituents $Ne^{1,2,3}_{TG}$
- Only recoverable heat amount from turbocharger air cooling water is almost in direct dependence on MCR, thus coming to the fact value $K_{\Delta\eta}$ will start to decline at some certain load level. This part of power net gain is difference between two values as follows - $Ne^1_{TG} - Ne^2_{TG} = \Delta Ne^{t/c}_{TG}$, which in its turn is sum of other two constituents as described above $\Delta Ne^{t/c}_{TG} = \Delta Ne^{t/c}_{TG_\xi} + \Delta Ne^{t/c}_{TG_{\Delta t}}$, i.e. due to saturated steam consumption substitution $\Delta Ne^{t/c}_{TG_\xi}$ (consumers ##2 and 3) and feed water warm up in HTS $\Delta Ne^{t/c}_{TG_{\Delta t}}$.

  With a reduction of inlet gas temperature, i.e. the choice of different ME modification, the effect of cooling water factor $K_{\Delta\eta}$ grows up (see Fig.7 b), while it comes to opposite result by ensuring more deep gas cooling rate. Nevertheless these are the most favorable ambient conditions for ultimate effective CWHRS operation.

7. B&W Ambient Conditions.

Almost at equally sea water and ambient air temperatures as follows- $t_{s/w} = 18^o C$ & $t_a = 20^o C$ these conditions correspond to summer one in temperate zone. Total sailing time for ships in general could reach up to $30 \div 38\%$ in average. As outlet from ME gas temperature is very much dependent on ambient one, then it $t_{g_0} \downarrow$ drops down considerably $\approx -15^o C$, while gas charge has a tendency to raise slightly $\approx +3\%$. Low potential heat consumers type B ## 4, 5, 6 are fully ensured by jacket cooling water heat in the whole range of ME load level, while effective and complete use of turbocharger air cooling water heat is limited with the MCR drop down below 76%. Despite of some adverse effect still these ambient conditions are quite favorable for effective performance by "CWHRS+HTS" also due to the fact that electrical consumption is the lowest possible one (see Fig.5) as air conditioning unit is switch off. Nevertheless the lowest main engine nominal output, at which autonomous turbo-generator performance is ensured, is higher and should be around $Ne_{ME} \geq 18 \div 25 MW$ @ $80\% MCR$, which will be inversely

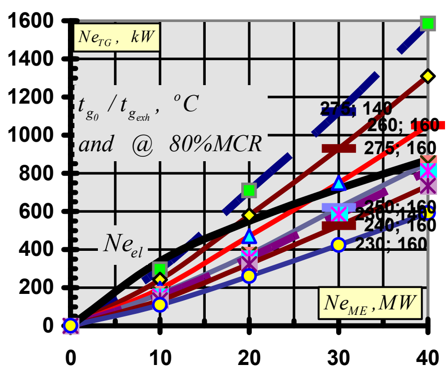


Figure 8. Turbo-generator output recovered by CWHRS in dependence on ME output/ modification @ B&W ambient conditions & MCR=80%.

proportional to reduction in both load levels and inlet gas temperatures (ME different modifications) (see Fig.8). Hence by choosing such a main engine nominal output level at specific MCR , when following precondition $Ne_{TG} > Ne_{el}$ is observed, we can expect in electrical power and heat supply solely by CWHRS only during around $\approx 50 \div 80\%$ of total sailing time. Due to lowered gas dates additional efficiency outcome $K_{\Delta\eta}$, as a result of WHRS complexity, is higher than for ISO ambient conditions (see Fig.9 a, b):

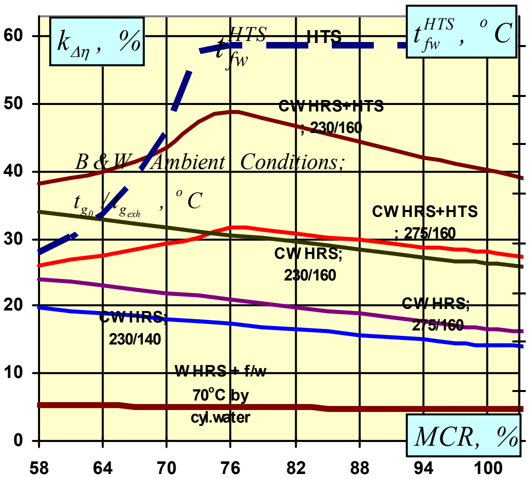


Figure 9a

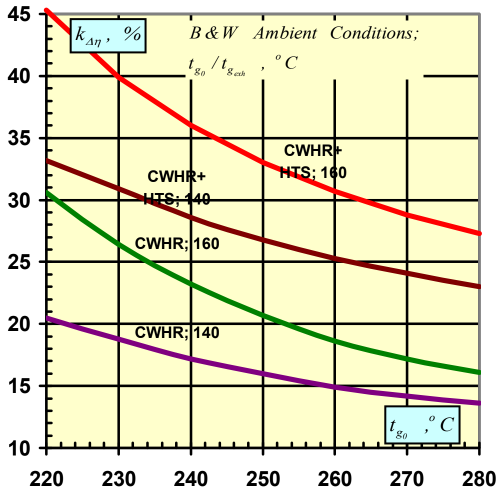


Figure 9b

WHRS Relative thermo efficiency increase rate dependence on **Fig.9 a** ME load level and **Fig.9b** inlet gas temperature due to various complexity of ME cooling water heat recovery @ B&W ambient conditions for different ME modifications & gas cooling rate.

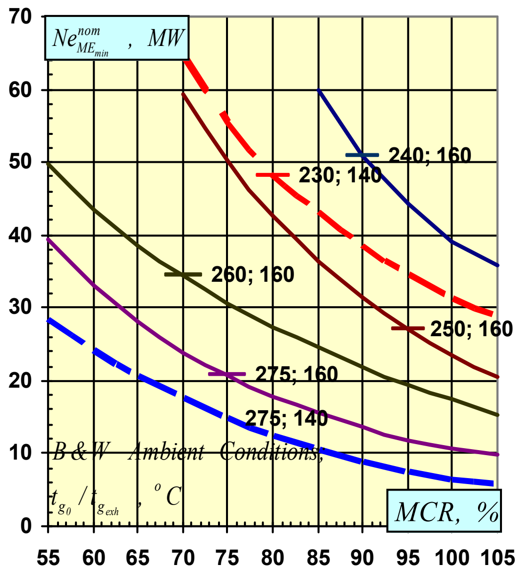


Figure 10a

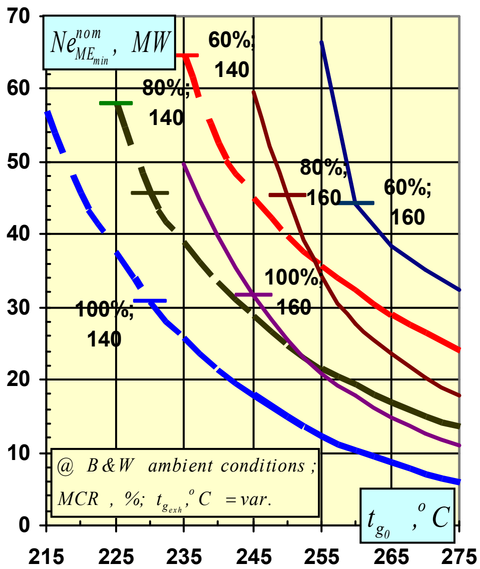


Figure 10b

The minimum required ME nominal output, when all ship demands in both electricity and heat are provided by CWHRS+HTS only, in dependence on **Fig.10a** ME load level and **Fig.10b** inlet gas temperature (ME modification) @ B&W ambient conditions for different ME modifications & gas cooling rate.

- But it $K_{\Delta\eta}$ is more sensitive to load level changes;
- So already below $\approx 76\%MCR$ High Temperature Stage cannot supply enough recovered cooling heat for consumers type A ##2, 3, thus slightly increasing saturated steam consumption and diminishing power gain $\Delta Ne^{t/c}_{TG_{\xi}} \downarrow$;
- But already at load level $\approx 73\%MCR$ cooling water temperature after HTS starts drop down till $\approx 80^{o}C$ at $60\%MCR$ as well as feed water preheating temperature and its relevant steam turbine power gain $\Delta Ne^{t/c}_{TG_{\Delta t}} \downarrow$;
- Jacket cooling water heat is in enough amount to warm up feed water till designed temperature and to substitute all determined low potential heat consumers in the whole range of ME load level;
  Based on presented dates required level of ME nominal output could be determined in dependence on load level and inlet gas temperature (ME modification) (see Fig.10a, b). These dates would be important during new-building project stage; and following interpretations are described as below:
- Till around $\approx 85\%MCR$ level there is slight increase in required ME nominal output $Ne^{nom}_{ME_{min}}$,
- What predetermines quite reliable and effective performance of "CWHRS+HTS";
- But at load levels below $\approx 75\%MCR$ "CWHRS + HTS" efficiency drop down is increasingly dominating;
- As well as inlet gas temperature reduction below $\approx 260 \div 255^{o}C$;
- Combination of both factor MCR and t_{g_0} reduction is the worst scenario for efficient CWHRS introduction and performance.

8. Conclusions

1. Due to reduced thermodynamic parameters of exhaust gasses after advanced slow speed diesel engines, further increase in WHRS efficiency could be obtained by extensive utilization of main engine cooling water heat.
2. Arrangements, how to recover cooling water heat, and the extent could be different in dependence on requirements, different conditions and economical usefulness, whether to recover jacket cooling water heat or turbocharged air one, or both. Such a system becomes more complex and in its ultimate version, id. "CWHRS+HTS", when both all possible low potential heat consumers and feed water pre-heating is ensured by recovery of cooling water heat only, efficiency increase is ensured as high as $K_{\Delta\eta} = 20 \div 40\%$ (see Fig. (see Fig.7, 9, 12) via reducing both saturated steam consumption and re-circulated boiler water amount.
3. Not only absolute figures of efficiency growth are important, but, especially, the fact, that all ship's demands in electricity and heat are met solely by CWHRS, is the main core point of the task. Therefore, possible efficient ways, how to compensate eventual electrical/heat shortage, are to be considered; but it could be done, when possible shortage, also surplus levels are found and studied in dependence on various factors affecting ships' service.
4. "CWHRS (+HTS)" efficiency rate strongly depends not only on ME nominal output and load level, but also ambient conditions are those, which has significant impact via flue gas dates (see Fig.6, 8, 11). In warmer regions CWHRS ability to supply ships' demands in electricity and heat are the most favorable ones (Fig. 6), while in cold sailing areas the

relative effect $K_{\Delta\eta}$ of recovered cooling water heat is the highest one (see Fig.12). Therefore both contribution and possible post-effects of WHRS complication should be carefully studied for each project. Nevertheless at possible MCR level $\approx 80\%$ ship demands in electricity are fully covered when ME nominal output is $\geq 15 \div 18MW$, which has a tendency to grow for colder regions as follows $\geq 28MW$.

5. With the reduction of design inlet gas temperature before EB t_{g_0}, id. different ME modification, turbocharger type, "CWHRS (+HTS)" ability to supply ships' demands in electricity and heat reduces significantly, especially when t_{g_0} reduces below $240^o C$.

References

1. Heinrich Schmid Winterthur Manager. Less Emissions Through Waste Heat Recovery. Application Technology, Ship Power Wärtsilä Switzerland Ltd, Wärtsilä Corporation, April 2004, p.10.

2. Heinrich Schmid and German Weisser. Marine Technologies for Reduced Emissions; Wärtsilä Switzerland Ltd, Winterthur, Wärtsilä Corporation, April 2005, p.12.

3. Маслов В.В. Утилизация теплоты судовых дизелей. Транспорт, 1990; 144 с.

4. J.Cimanskis, A.Zāģeris. Optimization method of thermodynamic analyzes of waste gas heat recovery circuit at fixed dimensions (height) of exhaust boiler. ICTS 2005, 9th International Conference on Transport Science, Portorož, Slovenia, p.15.

5. A.Zāģeris, J.Cimanskis. Some aspects of heat recovery possibilities at low gas temperature potential. 8th International Conference Maritime Transport and Infrastructure, 2006, p.11-18.

6. J.Cimanskis, A.Zāģeris. Dziļās gāzu utilizācijas sistēmu optimizācijas modeļi. RTU 45. Starptautiskā Zinātniskā Konference, 2004.gada 14.-16. oktobrī, Rīga.

7. Sulzer RTA48T and RTA58T Diesel Engines. Engine Selection and Project Manual. Issue June 1995. 172p.

8. Influence of Ambient Temperature Conditions on Main Engine Operation; http://www.manbw.com/article_001824.html, 16/03/2006, p.11

9. Thermo Efficiency System (TES) for Reduction of Fuel Consumption and CO2 Emission; http://www.manbw.com/article_001824.html, 19/07/2005, p.12.

A.Zāģeris, J.Cimanskis. Siltuma reģenerācijas iespējas modernizētajos zemapgrieziena dīzeļdzinēju energoiekārtās

Šo pētījumu objekts ir kuģu galveno dzinēju kompleksās izplūdes gāzu un dzesējamā ūdens siltuma utilizācijas iekārtas (KSUI – kompleksā siltuma utilizācijas iekārta) iespējas ar mērķi palielināt kuģa spēka iekārtas (KSI) kopējo termodinamisko lietderību. Pateicoties plašajam pielietojumam, kā kuģa galvenais dzinējs (GD) irparedzēts modernais mazapgriezienu divtaktu dīzeļdzinējs, kuram raksturīgs zems izplūdes gāzu termiskais potenciāls, tādējādi liekot detalizēti izvērtēt zaudējamā siltuma atgriešanas iespējas, nodrošinot gan papildu jaudas izstrādi ar turbo-ģeneratora palīdzību, gan apgādājot nepieciešamos siltuma patērētājus, atkarībā no vides, GD jaudas, noslodzes un citiem faktoriem. Iegūtie rezultāti dod iespēju gan novērtēt efektivitātes pieaugumu, gan prognozēt nepieciešamos nosacījumus elektrostacijas izvēles optimizācijai.

A.Zāģeris, J.Cimanskis. Возможности регенораций тепла в модернизированых низкооборотных дизельных энергоустановках

Обьетом исследований является система глубокой утилизации тепла (СГУТ) как уходящих газов, так и воды охлаждения судового главного двигателя (ГД) с целью максимально повысить термодинамическую эффективность судовой силовой установки (ССУ). Благодаря широкому применению в качества ГД выбран современный малооборотный двухтактный дизель, характеризуемый низкими термодинамическими параметрами утилизируемого тепла, что требует тщательно обосновать возможные способы возврата теряемого тепла, путем обеспечения как выработки дополнительной мощности турбогенератором, так и работы различных потребителей тепла в зависимости от внешних условий, мощности и загрузки ГД, и других факторов. Полученные результаты дают возможность оценить как прирост эффективности, так и прогнозировать необходимые условия для оптимального выбора электростанции.

A.Zāģeris, J.Cimanskis. Heat recovery possibilities for advanced slow speed diesel engine power plants

In order to ensure further growth in thermo efficiency of ships' power plant, complex recovery of both exhaust gas and cooling water heat from ships' main engine (ME) is the object of investigations. Advanced low speed diesel engines are considered as ships' ME due to its wide application in marine industry. At the same time thermo potential of exhaust gasses are the lowest ones for the engines, thus requiring detailed investigation of heat recovery in way of both additional power generation by steam turbine and ships' required heat supply in dependence on ambient, ME output and load level, and other factors. Obtained results give us a possibility to evaluate both eventual efficiency growth and forecast required conditions to carry out optimization of electric power station.