**Scientific Journals**
**of the Maritime University of Szczecin**

**Zeszyty Naukowe**
**Akademii Morskiej w Szczecinie**

**2019, 60 (132), 171–176**
**ISSN 1733-8670 (Printed)**
**ISSN 2392-0378 (Online)**
**DOI: 10.17402/386**

*Received: 25.09.2019*
*Accepted: 02.12.2019*
*Published: 18.12.2019*

# Determination of the operating parameters of steam jet injectors for a main boiler's regenerative feedwater system

**Andrzej Adamkiewicz**[1], **Szymon Grzesiak**[2]

Maritime University of Szczecin, Faculty of Maritime Engineering
1–2 Wały Chrobrego St., 70-500 Szczecin, Poland
e-mail: [1]a.adamkiewicz@am.szczecin.pl, [2]grzesiak87@gmail.com
corresponding author

**Key words:** steam jet injector, steam cycle, steam turbine, propulsion plant, thermal efficiency, determination

**Abstract**

Due to the development of alternative propulsion systems, there is a need for LNG tanker turbine propulsion plants to regain their competitiveness. Previous research revealed effective methods to increase the thermal efficiency of the steam cycle based on quality assessment, and it was proposed that the latent heat of the main turbine exhaust steam could be recovered. Research was carried out for the steam cycle using regenerative heat exchangers fed by steam jet injectors. In this paper, an algorithm to determine the operating parameters of steam jet injectors, and the calculation results for different drive steam parameters are presented. The obtained results will be used as input parameters for further heat balance calculations of the proposed regenerative steam cycles.

## Introduction

An analysis of modern steam propulsion systems of LNG tankers (Dzida & Mucharski, 2009; Adamkiewicz & Grzesiak, 2017) indicates that they have insufficient thermal efficiencies. Despite the advantages of these propulsion plants such as their reliability, low maintenance costs (OPEX OPerational EXpenditure), low emissions ($NO_X$, $SO_X$, HC), and the simplicity of energy conversion, they are being displaced from the market by highly efficient plants equipped with diesel engines (Grzesiak, 2018; IGU, 2018). At the same time, steam turbine manufacturers are pursuing research and development to increase their energy efficiency (Hirdaris et al., 2014; Kowalczyk, Głuch & Ziółkowski, 2016; Adamkiewicz & Grzesiak, 2017; Grzesiak, 2018).

In order to determine the possibility of increasing the efficiency of steam turbine plants, the identification of waste heat energy sources and a quality assessment were carried out for two of the main waste heat energy fluxes: exhaust gas from main boilers and condensation heat released in the main condenser (Adamkiewicz & Grzesiak, 2019). The analysis showed (Adamkiewicz & Grzesiak, 2018; 2019; Grzesiak, 2018; Grzesiak & Adamkiewicz, 2018) an unsatisfactory efficiency of turbine propulsion plants compared with other systems, and the need to analyse waste heat energy fluxes in order to research feasible technologies for its effective use. This analysis showed that (Adamkiewicz & Grzesiak, 2019) the exhaust steam flux has a high energy potential, but its energy level is too low to be useful for regenerative feed water heating due to its low temperature and pressure. Additionally, there is a need to identify solutions to increase the energy level of exhaust steam so that it is useful.

The use of steam injectors in which the turbine exhaust steam mixes with the turbine bleed steam offers a possible solution. The results of calculations made for simple systems according to the Clausius-Rankine cycle, whose heat-flow diagrams are presented in Figure 1 (Adamkiewicz & Grzesiak,

**Table 1. Determined functions of evaluation of the waste energy source quality**

| | Mass Flow | Energy flux | Press. Abs. | Temp. | Enthalpy | x | Exergy | ψ temp | ψ f(b,i) |
|---|---|---|---|---|---|---|---|---|---|
| | [kg/s] | [kJ/s] | [bar] | [°C] | [kJ/kg] | [–] | [kJ/kg] | [–] | [–] |
| MT condenser losses | 22.61 | 48742.7 | 0.066 | 38 | 2294 | 0.888 | 1926.4 | 0.132 | 0.8936 |
| TA condenser losses | 1.587 | 3673.99 | 0.075 | 40 | 2452 | 0.95 | 2069.7 | 0.175 | 0.8945 |
| Exhaust losses | 43.84 | 12482.2 | 1.05 | 155 | 285 | xx | 139.2 | 0.806 | 0.5460 |

2018; Grzesiak & Adamkiewicz, 2018) indicate the validity of using a steam jet injector. Such a use proposes a modification while maintaining the same steam cycle parameters (superheated steam pressure and temperature). This increases the thermal efficiency of the plant due to a decrease in the bleed steam demand, which increases the available enthalpy drop in the turbine. At the same time, less heat is removed from the cycle in the condenser. Increasing the ejection level and using the bleed steam from the lowest possible energy level increases the regeneration degree of the plant (Adamkiewicz & Grzesiak, 2018; Grzesiak & Adamkiewicz, 2018). However, using the injector to obtain the desired pressure at the device outlet requires a relatively high supply steam pressure.

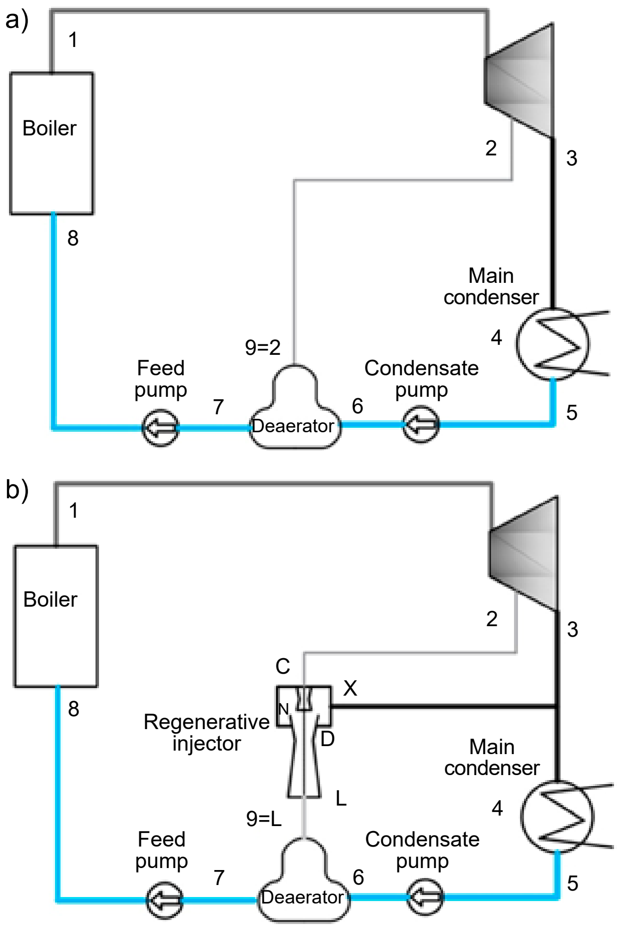


**Figure 1. Thermal – flow diagram of proposed model a) Clausius-Rankine cycle with regenerative heater (deaerator) feed on steam bleed; b) Clausius-Rankine cycle with regenerative heater (deaerator) feed by regenerative injector**

This makes it necessary to determine the bleed steam parameters to effectively supply the regenerative steam injector, which is the aim of this article (Hirdaris et al., 2014). The bleed steam parameters were determined from an exemplary steam-powered system (CST – conventional steam turbine) of an LNG tanker with a capacity of 138,000 $m^3$ from 2003.

For the calculations, the obtainable parameters of the state of the considered propulsion plant steam cycle (CST) were selected. For selected energy fluxes, the operating parameters of steam injectors in individual control planes were determined. The calculation results will serve as inputs for additional calculations of the heat balance of the cycles by applying steam jet injectors (in the regenerative main boiler feedwater systems of vessels).

## Determination of parameters of driving steam for steam jet injectors

The parameters of the bleed steam supplying the regenerative steam jet injectors were determined. Table 2 shows the steam parameters of the cycle implemented by a conventional steam system of an LNG tanker. The last two fluxes in Table 2 are the steam parameters determined based on the expansion curve, which was based on the state parameters at the measuring points available for the UA-400 turbine (Figure 2).

## Determination of operating parameters of steam jet injectors

The outlet stream parameters of the injectors were determined for the feed injector bleed steam at 19.5 bar, 10 bar, 6.6 bar, 3.1 bar, 3 bar, and 1.5 bar (Table 3). The outlet steam from steam jet injectors is a mixture of feed steam (bleed steam from the main propulsion turbine) and sucked steam (exhaust steam from the main propulsion turbine).

**Table 2. Parameters of the steam for the CST plant of an LNG carrier**

| State | Pressure | Temp | Enthalpy | | x |
|---|---|---|---|---|---|
| [–] | [bar] | [°C] | [kcal/kg] | [kJ/kg] | [–] |
| Superheated Steam after boilers | 61 | 525 | 831.3 | 3481 | 1 |
| Superheated Steam HP Turbine In | 59.5 | 520 | 828.7 | 3470 | 1 |
| HP Bleed | 19.5 | 372 | 761 | 3186.2 | 1 |
| HP Turbine Exhaust | 6.6 | 245 | 703 | 2943 | 1 |
| IP Bleed to HP Heater | 6.6 | 245 | 703 | 2943 | 1 |
| Feed Pumps Exhaust Steam | 3.1 | 310 | 742 | 3100 | 1 |
| LP Blead | 1.5 | 131 | 653 | 2734 | 1 |
| TA Exhaust Steam | 0.075 | 40 | 587 | 2452 | 0.95 |
| Exhaust Steam from LP Turbine | 0.06 | 38 | 551 | 2294 | 0.89 |
| 10 bar from expansion curve | 10 | 287 | 722.5 | 3025 | 1 |
| 3 bar from expansion curve | 3 | 170 | 669.5 | 2803 | 1 |

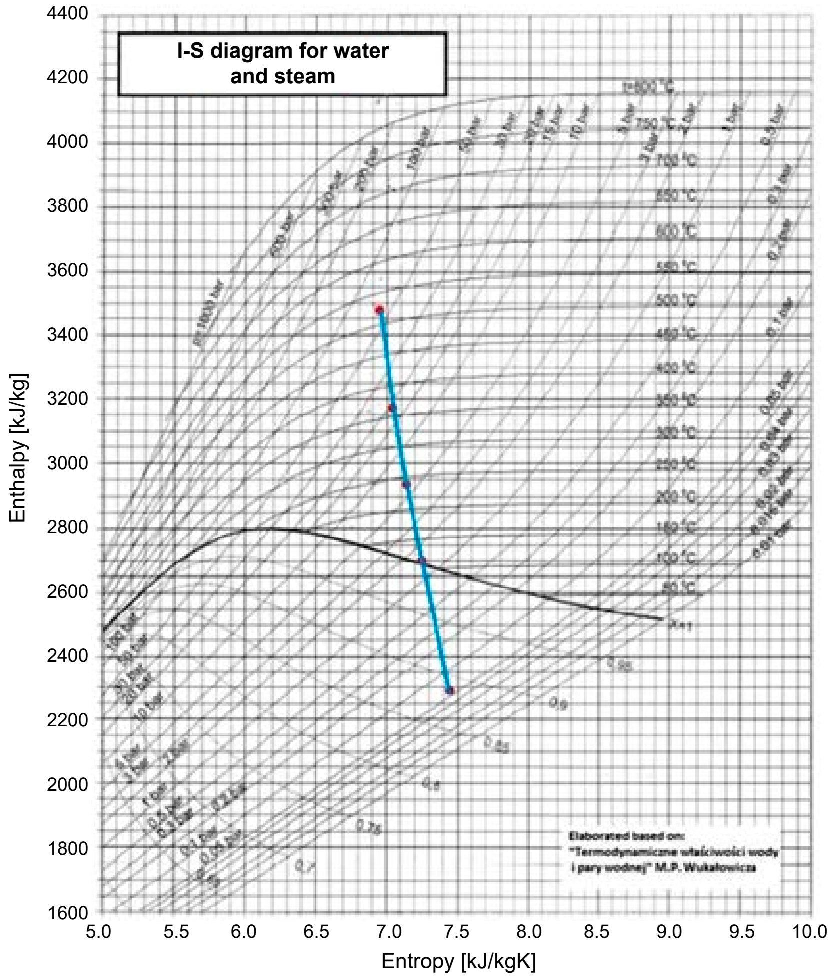


**Figure 2. Steam expansion curve in a Kawasaki UA-400 Turbine on an I-S diagram**

Calculations were made in accordance with the algorithm presented in Figure 3 (Gryboś, 1956; Goliński & Troskolański, 1979; Hegazy, 2007).

Figure 4 presents the correlation of steam pressure leaving the injector as a function of the degree of ejection (defined as the ratio of the steam sucked in by the injector to the drive steam of the injector). The equation describing the steam pressure after the injector depending on the assumed degree of ejection was also determined.

The calculated exhaust steam pressures for a steam injector fed by 19.5 and 10 bar allow the

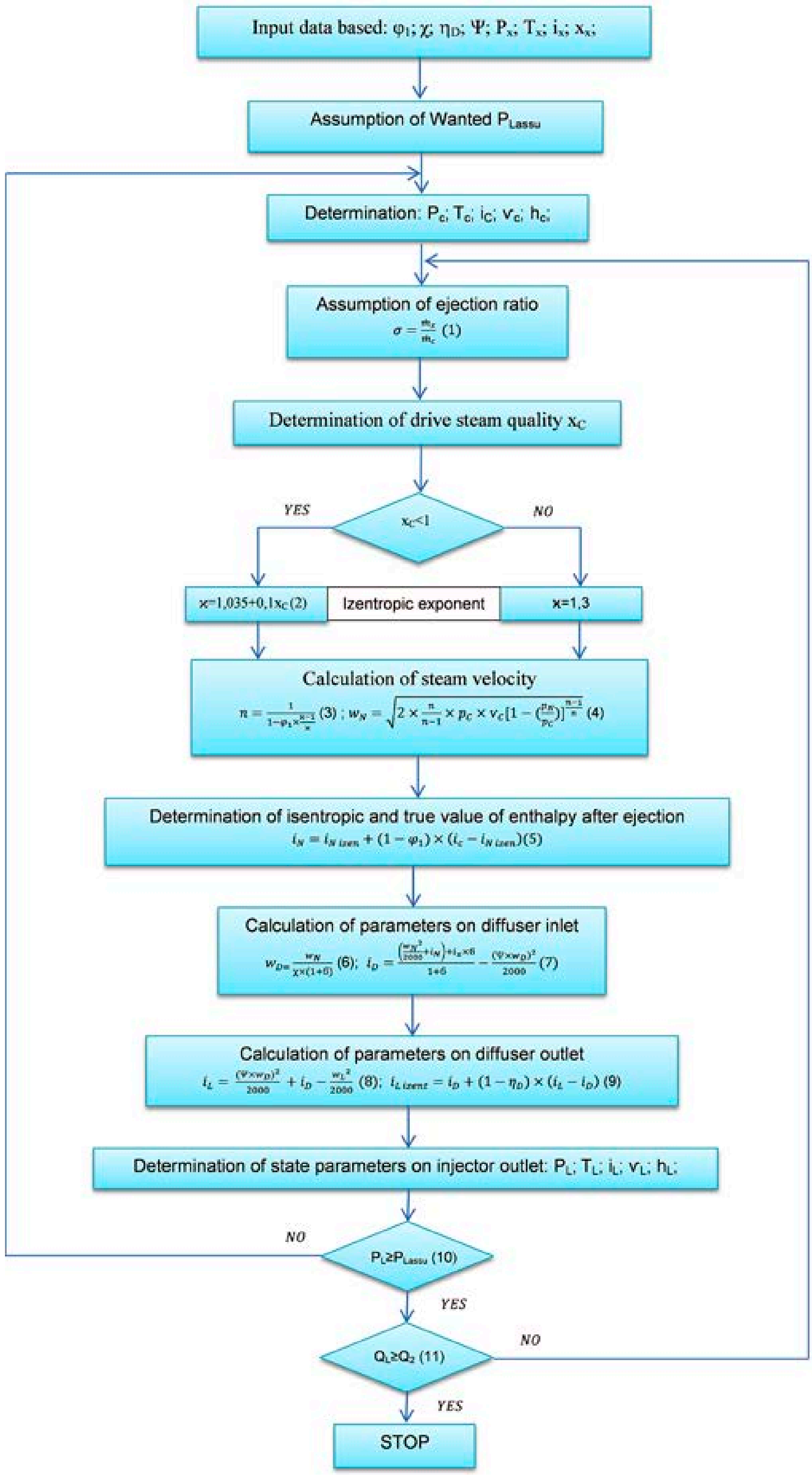


**Figure 3. Regenerative injector calculation algorithm**

**Table 3. Calculation results for steam injectors at different drive steam pressures**

| | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| HP Bleed 19.5 bar | | | | | | | | | | | |
| σ | [–] | Ejection Ratio – assumed | 0.667 | 0.500 | 0.429 | 0.376 | 0.333 | 0.250 | 0.200 | 0.167 | 0.143 |
| wd | [m/s] | Velocity of steam on diffusor inlet | 883.2 | 981.3 | 1029.9 | 1069.8 | 1103.9 | 1177,5 | 1226.6 | 1261.6 | 1287.9 |
| iD | [kJ/kg] | Enthalpy of steam on diffusor inlet | 2521.4 | 2507.0 | 2496.9 | 2487.3 | 2478.0 | 2454.7 | 2436.8 | 2422.8 | 2411.7 |
| iLizen | [kJ/kg] | Enthalpy of steam mixture after isentropic compression | 2804.0 | 2856.3 | 2881.9 | 2902.8 | 2920.6 | 2958.5 | 2983.6 | 3001.4 | 30147 |
| iL= | [kJ/kg] | Enthalpy of steam after diffusor | 2835.4 | 2895.1 | 2924.7 | 2949.0 | 2969.8 | 3014.5 | 3044.4 | 3065.7 | 3081.7 |
| tl | [°C] | Temperature of steam after diffusor | 178.0 | 208.9 | 224.2 | 236.7 | 247.4 | 270.3 | 285.6 | 2966 | 3048 |
| pl | [bar] | Pressure of steam after diffusor | 0.368 | 0.525 | 0.633 | 0.745 | 0.86 | 1.19 | 1.5 | 1.835 | 2.025 |
| Bleed 10 bar – determined from the expansion curve | | | | | | | | | | | |
| σ | [–] | Ejection Ratio – assumed | 0.667 | 0.500 | 0.429 | 0.376 | 0.333 | 0.250 | 0.200 | 0.167 | 0.143 |
| wd | [m/s] | Velocity of steam on diffusor inlet | 797.3 | 885.9 | 929.8 | 965.7 | 996.6 | 1063.0 | 1107.3 | 1139.0 | 1162.7 |
| iD | [kJ/kg] | Enthalpy of steam on diffusor inlet | 2472.7 | 2460.1 | 2451.5 | 2443.2 | 2435.4 | 2415.8 | 2400.7 | 2389.0 | 2379.7 |
| iLizen | [kJ/kg] | Enthalpy of steam mixture after isentropic compression | 2702.7 | 2744.5 | 2764.9 | 2781.6 | 2795.8 | 2826.1 | 2846.1 | 2860,3 | 2870,8 |
| iL= | [kJ/kg] | Enthalpy of steam after diffusor | 2728.3 | 2776.1 | 2799.8 | 2819.2 | 2835.8 | 2871.7 | 2895.6 | 2912.6 | 2925.4 |
| tl | [°C] | Temperature of steam after diffusor | 122.7 | 147.905 | 160.3 | 170.5 | 179.3 | 198.1 | 210 | 219.7 | 226.4 |
| pl | [bar] | Pressure of steam after diffusor | 0.302 | 0.42 | 0.492 | 0.565 | 0.637 | 0.86 | 1.05 | 1.225 | 1.375 |
| IP Bleed 6.6 bar | | | | | | | | | | | |
| σ | [–] | Ejection Ratio – assumed | 0.667 | 0.500 | 0.429 | 0.376 | 0.333 | 0.250 | 0.200 | 0.167 | 0.143 |
| wd | [m/s] | Velocity of steam on diffusor inlet | 746.8 | 829.8 | 870.9 | 904.6 | 933.5 | 995.7 | 1037.2 | 1066.9 | 1089.1 |
| iD | [kJ/kg] | Enthalpy of steam on diffusor inlet | 2448.6 | 2437.2 | 2429.5 | 2422.1 | 2415.1 | 2397.7 | 2384.3 | 2373.9 | 2365.7 |
| iLizen | [kJ/kg] | Enthalpy of steam mixture after isentropic compression | 2650.2 | 2686.5 | 2704.3 | 2718.8 | 2731.1 | 2757.5 | 2774.9 | 2787.2 | 2796.4 |
| iL= | [kJ/kg] | Enthalpy of steam after diffusor | 2672.6 | 2714.2 | 2734.9 | 2751.8 | 2766.2 | 2797.4 | 2818.2 | 2833.1 | 2844.3 |
| tl | [°C] | Temperature of steam after diffusor | 93.4 | 115.4 | 126.3 | 135.3 | 142.9 | 159.5 | 170.5 | 178.4 | 184.3 |
| pl | [bar] | Pressure of steam after diffusor | 0.262 | 0.35 | 0.41 | 0.47 | 0.53 | 0.69 | 0.825 | 0.95 | 1.05 |
| Feed pumps exhaust 3.1 bar | | | | | | | | | | | |
| σ | [–] | Ejection Ratio – assumed | 0.667 | 0.500 | 0.429 | 0.376 | 0.333 | 0.250 | 0.200 | 0.167 | 0.143 |
| wd | [m/s] | Velocity of steam on diffusor inlet | xxx | xxx | 883.1 | 917.3 | 946.6 | 1009.7 | 1051.8 | 1081.8 | 1104.4 |
| iD | [kJ/kg] | Enthalpy of steam on diffusor inlet | xxx | xxx | 2569.0 | 2566.7 | 2564.0 | 2555.8 | 2548.5 | 2542.5 | 2537.5 |
| iLizen | [kJ/kg] | Enthalpy of steam mixture after isentropic compression | xxx | xxx | 2851.7 | 2871.8 | 2889.0 | 2925.8 | 2950.0 | 2967.5 | 2980.4 |
| iL= | [kJ/kg] | Enthalpy of steam after diffusor | xxx | xxx | 2883.1 | 2905.7 | 2925.1 | 2966.9 | 2994.8 | 3014.7 | 3029.6 |
| tl | [°C] | Temperature of steam after diffusor | xxx | xxx | 202.3 | 214.1 | 223.9 | 245.4 | 259.7 | 269.7 | 277.2 |
| pl | bar] | Pressure of steam after diffusor | xxx | xxx | 0.333 | 0.371 | 0.41 | 0.51 | 0.598 | 0.67 | 0.732 |
| Bleed 3 bar – determined based on expansion curve | | | | | | | | | | | |
| σ | [–] | Ejection Ratio – assumed | 0.667 | 0.500 | 0.429 | 0.376 | 0.333 | 0.250 | 0.200 | 0.167 | 0.143 |
| wd | [m/s] | Velocity of steam on diffusor inlet | xxx | xxx | 762.3 | 791.8 | 817.1 | 871.5 | 907.9 | 933.8 | 953.2 |
| iD | [kJ/kg] | Enthalpy of steam on diffusor inlet | xxx | xxx | 2399.7 | 2394.1 | 2388.7 | 2375.4 | 2365.2 | 2357.2 | 2350.9 |
| iLizen | [kJ/kg] | Enthalpy of steam mixture after isentropic compression | xxx | xxx | 2609.8 | 2620.9 | 2630.4 | 2650.6 | 2664.0 | 2673.4 | 2680.5 |
| iL= | [kJ/kg] | Enthalpy of steam after diffusor | xxx | xxx | 2633.2 | 2646.2 | 2657.3 | 2681.2 | 2697.2 | 2708.6 | 2717.1 |
| tl | [°C] | Temperature of steam after diffusor | xxx | xxx | 73.3 | 80.2 | 86.2 | 99.0 | 107.6 | 113.7 | 118.3 |
| pl | [bar] | Pressure of steam after diffusor | xxx | xxx | 0.286 | 0.319 | 0.352 | 0.438 | 0.512 | 0.571 | 0.623 |

exhaust steam from the injectors to be used, depending on the degree of ejection, for both vacuum heat exchangers as well as for deaerator tanks. In addition, the results obtained for the lower ejection levels at a supply pressure of 19.5 bar indicate the possibility of using the steam in overpressure exchangers.

The calculated results of the regenerative injector for intermediate steam bleed with an absolute pressure of 6.6 bar indicate that it is possible to use exhaust steam from the injector in the vacuum heat exchanger for the assumed ejection levels σ = [0.142; 0.500].

Calculations for the steam supplying the steam injector with a pressure of 3.0 bar and the exhaust steam from the turbine feedwater pump at a pressure of 3.1 bar were carried out for the ejection ratio σ = [0.142; 0.429]. For higher ejection ratios, the exhaust steam pressure of the ejector was too low.

Due to the low values of the steam pressure after the injector, the calculation results of the injector fed with steam from the low-pressure bleed (LP Bleed – 1.5 bar) were omitted from further analysis.

Figure 4 shows the dependence of the steam pressure leaving the injector on the ejection degree used. The determined values are a set of possible parameters of injector exhaust steam for use in additional calculations of this proposed system (Figure 1).

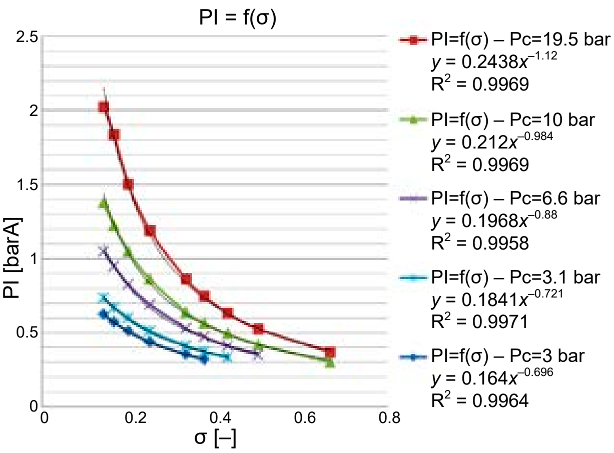


**Figure 4. Correlation of injector steam exhaust pressure as a function of the ejection ratio**

## Conclusions

The obtained results indicate that increasing the ejection level (which positively impacts the degree of regeneration) decreases the pressure and enthalpy of the injector exhaust steam. On the other hand, this decrease will reduce the maximum achievable boiler feedwater temperature, which will result in a reduction in the regeneration degree of cycle.

Using steam with higher parameters to supply the injector (pressure and temperature of superheating) increased the steam pressure leaving the injector. This enabled feed water higher temperatures to be obtained and broadens the potential applications of the obtained steam. However, the use of high-parameter bleed steam from the turbine to drive injectors decreased the available enthalpy drop across the turbine stages and reduced the plant's efficiency.

## References


1. Adamkiewicz, A. & Grzesiak, S. (2017) Evolution of energy efficiency of modern LNG carrier's steam turbine propulsion plant. *Rynek Energii* 130, 3, pp. 67–76 (in Polish).
2. Adamkiewicz, A. & Grzesiak, S. (2018) Koncepcja zwiększenia stopnia regeneracji turbinowego obiegu parowego. *Zeszyty Naukowe Akademii Morskiej w Gdyni, Scientific Journal of Gdynia Maritime University* 108, pp. 9–21.
3. Adamkiewicz, A. & Grzesiak, S. (2019) Identification of waste heat energy sources of a conventional steam propulsion plant of LNG carrier. *Archives of Thermodynamics* 40, 3, pp. 195–210.
4. Dzida, M. & Mucharski, J. (2009) On the possible increasing of efficiency of ship power plant with the system combined of marine diesel engine, gas turbine and steam turbine in case of main engine cooperation with the gas turbine fed in parallel and the steam turbine. *Polish Maritime Research* 2 (60), 16, pp. 40–44.
5. Goliński, A. & Troskolański, T. (1979) *Strumienice: Teoria i konstrukcja*. Warszawa: WNT.
6. Gryboś, R. (1956) Regeneracja ciepła w siłowni z turbiną bezupustową. *Zeszyty Naukowe Politechniki Śląskiej* 1, 5, pp. 59–80.
7. Grzesiak, S. (2018) Alternative Propulsion Plants for Modern LNG Carriers. *New Trends in Production Engineering* 1, 1, pp. 399–408.
8. Grzesiak, S. & Adamkiewicz, A. (2018) Application of Steam Jet Injector for Latent Heat Recovery of Marine steam Turbine Propulsion Plant. *New Trends in Production Engineering* 1, 1, pp. 235–246.
9. Hegazy, A. (2007) Possible Waste Heat Recovery in the Condenser of a Regenerative Steam Cycle. *Journal of Thermal Science and Technology* 2, 1, pp. 1–12.
10. Hirdaris, S., Cheng, Y., Shallcross, P., Bonafoux, J., Carlson, D., Prince, B. & Sarris, G. (2014) Considerations on the potential use of Nuclear Small Modular Reactor (SMR) technology for merchant marine propulsion. *Ocean Engineering* 79, pp. 101–130.
11. IGU (2018) *2018 World LNG Report*. [Online] June 28, 2018. Available from: https://www.igu.org/news/2018-world-lng-report [Accessed: August 2018].
12. Kowalczyk, T., Głuch, J. & Ziółkowski, P. (2016) Analysis of possible application of high-temperature nuclear reactors to contemporary large-output steam power plants on ships. *Polish Maritime Research* 23 (90), pp. 32–41.