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Energy analysis of the direct and indirect refrigerating systems applied to refrigerated ships

SUMMARY

In the paper different design solutions of refrigerating systems are considered, which are applicable to the universal refrigerated ships employed in shipping of refrigerated cargo within a wide range of temperature.

The example systems are analyzed and compared to each other with the use of the exoergic efficiency criterion. Example calculations of defrosting cycle influence on the exoergic efficiency of both analyzed systems are also presented.

INTRODUCTION

Higher and higher demands are made for ships engaged in refrigerated cargo shipping, not only dealing with their technical and operational parameters, but - first of all - with their energy consumption as well as detrimental effects to the environment. Widening application takes place of novel hold insulation techniques, cool air distribution, as well as of sensitive cargo handling, storing and shipping. All that together makes those ships more and more universal, able to carry different goods within a wide range of temperature. At present two basic designs of refrigerating systems are used on the refrigerated ships : the systems with direct evaporation of a refrigerant within hold air coolers and those in which an additional cooling system with brine (C_aCl_2) circulation is applied [2,5]. Opinions about energy efficiency and applicability of both designs are different. The problem is below analyzed in detail.

The universal refrigerated ships are characterized of a large and changeable energy demand depending on ship's route, season of the year, and kind of transported cargo. Therefore the ships should be fitted with the electric energy sources capable - in every conditions - - to satisfy demand from the side of the refrigerating systems installed onboard. On the other hand the refrigerating systems should satisfy technological conditions demanded by the refrigerated cargo, at as - - low - as - possible investment and exploitation costs. In order to apply an appropriate design solution the decision should be made which kind of the refrigerating system - direct or indirect - has to be chosen. For that reason influence of choice of kind of the system on its exoergic efficiency is analyzed in the paper.

SHIP HOLD REFRIGERATING SYSTEMS

If the way of heat absorption from the refrigerated space is assumed as the criterion, two alternative refrigerating systems can be distinguished : direct and indirect one (using brine).

The direct refrigerating system (Fig.1) is characterized of that the heat contained in the refrigerated space is transferred by the refrigerant which evaporates within the diaphragm air cooler being the evaporator of the refrigerating system.

In the indirect refrigerating system (Fig.2) the heat contained in the refrigerated space is transported by means of the brine which is heated in the diaphragm air cooler, and which then gives back the heat within the shell - and - coil evaporator of the refrigerant.

On ships, for the safety reasons, the closed indirect systems are often used in which circulation of the brine between the air cooler and evaporator (brine cooler) is forced by means of pumps. In this case the heat amount absorbed from the refrigerated space, enlarged by a thermal equivalent of work of the circulation pumps, is transmitted to the refrigerant within the shell - and - coil evaporator; the further path of heat is the same as that in the direct refrigerating systems.

Having that in mind one could conclude that the direct refrigerating system is superior over that indirect. However it seems that in the nearest future the refrigerating brine systems will be used on ships still equally often and willingly, a.o. due to much lower consumption of the expensive refrigerating media filling the installations.

Moreover, in many international legal instruments, a.o., application of such design solutions of the refrigerating installations is recommended which contain not large amount of the refrigerating medium, and this recommendation is just satisfied by the indirect systems.

In this paper the direct refrigerating systems and indirect refrigerating ones (using brine) are mutually compared with the use of the exoergic efficiency criterium. Influence of defrosting cycle on increase of energy consumption of both analyzed systems is also assessed.

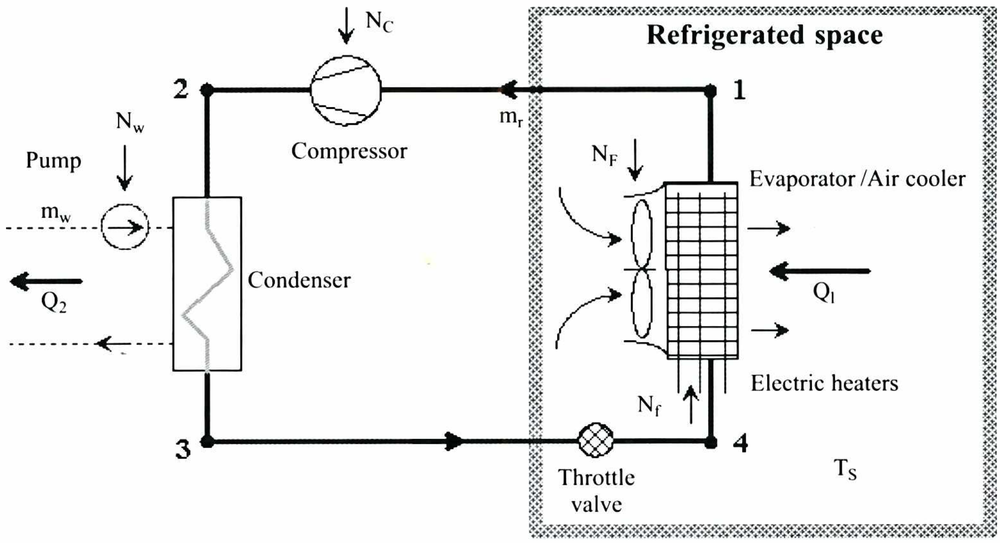

Fig.1. Schematic diagram of the direct refrigerating system

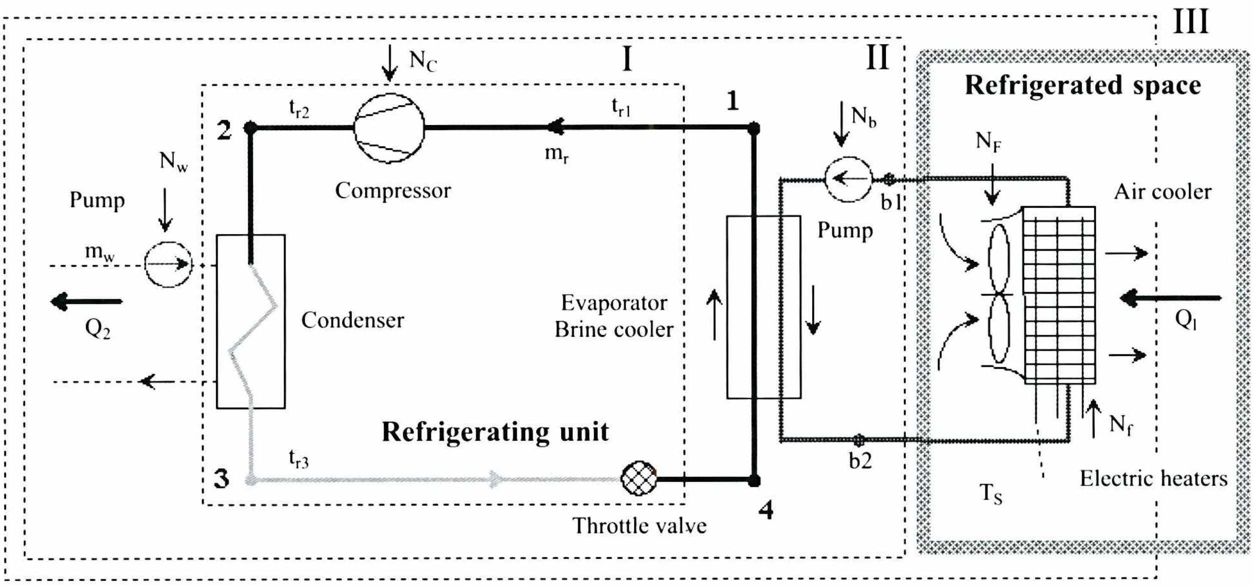

Fig.2. Schematic diagram of the indirect refrigerating (brine) system

EXOERGIC ANALYSIS OF COMPRESSOR REFRIGERATING SYSTEM

Temperature values of the refrigerating media or heat sources applied in refrigerating engineering are usually lower than ambient temperature. In such cases the notion of exoenergy is particularly helpful for rational assessment of energy usefulness of heat or refrigerating medium. For instance, if temperature of the medium colder than ambient temperature, decreases, then usefulness of the medium and its exoenergy being a measure of the usefulness, is greater even though enthalpy of the medium becomes smaller. Change of exoenergy of the heat source colder than environment can be much greater than the heat amount exchanged with the source. This occurs if the heat source temperature T_S is lower than $T_A/2$ [4].

Exoergic efficiency of compressor refrigerating system

In Fig.2. the schematic diagram of the refrigerating system together with the brine and cooling water circuits is presented. Different assumptions can be made for perfectness assessment of the refrigerating processes of such systems [4]. Balance **partition I** shown in Fig.2, separates the refrigerating unit out of the evaporator / brine cooler. Exoergic efficiency of the unit enclosed by the partition determines degree of perfectness of thermodynamic processes within that unit (together with electric motors driving the compressor or pumps, as well as auxiliary equipment in the case of the refrigerating unit). The exoergic efficiency of such unit can be called the gross efficiency η_{Bg} expressed as follows :

$$\eta_{Bg} = \frac{m_r(b_{r4} - b_{r1})}{\dot{B}_d} \tag{1}$$

where :

- m_r – flow rate of circulating refrigerant
- b_{r4}, b_{r1} – specific exoenergy of circulating refrigerant at inlet to and outlet from the evaporator, respectively
- $\dot{B}_d$ – driving exoenergy of the refrigerating unit.

For the exoergic balance of the system contained within **partition I** it can be assumed that the difference of exoenergy of cooling water at the outlet from the condenser and that at the inlet to it, constitutes an external loss of exoenergy.

Balance **partition II** additionally encloses auxiliary equipment of the refrigerating unit (i.e. cooling water and brine pumps) as well as exoenergy losses during transfer of brine and losses of cooling water in the case of land - based systems. Exoergic efficiency of such system can be called the net exoergic efficiency η_{Bn} of the refrigerating unit, expressed as follows :

$$\eta_{Bn} = \frac{m_b(b_{b2} - b_{b1})}{\dot{B}_d + m_{wl}b_w + N_b + N_w - (N_{el})_t} \qquad (2)$$

where :

m_b – brine flow rate

b_{b1} , b_{b2} – specific exoenergy of brine at inlet to and outlet from the brine cooler

m_{wl} , b_w – flow rate and specific exoenergy of the water supplementing loss of cooling water, respectively

N_b , N_w – electric power demand of pump driving motor of brine and cooling water, respectively

$(N_{el})_t$ – total electric driving power demand of auxiliary equipment, i.e. fans, automatic machinery elements.

Balance **partition III** comprises all thermodynamic processes of the refrigerating system. The efficiency of the system enclosed by the partition can be called the total exoergic efficiency η_{Bt}. It accounts for the final effect of the system's operation, i.e. increase of exoenergy of a refrigerated space or refrigerated substance, as follows :

$$\eta_{Bt} = \frac{\Delta\dot{B}_S}{\dot{B}_d + m_{wl}b_w + N_b + N_w + (N_{el})_t} \qquad (3)$$

where :

$\Delta\dot{B}_S$ - increase of exoenergy of a refrigerated space or refrigerated substance.

If the task of the refrigerating system is to maintain constant temperature within a refrigerated space, then $\Delta\dot{B}_S$ can be calculated by using (4) :

$$\Delta\dot{B}_S = -\dot{Q}\frac{(T_S - T_A)}{T_S} \qquad (4)$$

In general, a way of determining the driving exoenergy $\dot{B}_d$ included in (1) , (2) and (3) depends on a type of the refrigerating system. In the compressor refrigerating system its driving exoenergy is equivalent to the electric drive power $(N_{el})_C$ of the compressor, therefore :

$$\dot{B}_d = (N_{el})_C \qquad (5)$$

COMPARISON OF EXOERGIC EFFICIENCY AND RELATIVE EXOENERGY LOSSES OF EXAMPLE COMPRESSOR REFRIGERATING SYSTEMS

Values of exoergic efficiency and relative exoenergy losses of two example compressor refrigerating systems (direct system and indirect one using brine) are determined and compared to each other, for illustration.

Example exoergic balance of the indirect refrigerating system

The compressor indirect refrigerating system described in [4], of the schematic diagram shown in Fig.2, was assumed the object of analysis. The relevant original data of the system are as follows :

- Temperature of the refrigerating space : $t_S = -1°C$
- Ambient temperature : $t_A = 20°C$
- Refrigerating unit output measured at the evaporator : $\dot{Q}_1 = 93.03$ kW
- Circulating refrigerant : ammonia (NH_3)
- Evaporation temperature : $t_o = -12°C$
- Compressor sucks in a little overheated vapour of the temperature $t_{r1} = -10°C$
- Ammonia temperature at outlet of the compressor : $t_{r2} = 119°C$
- Condensation temperature : $t_c = 28°C$
- Ammonia condensate is cooled down to the temperature : $t_{r3} = 25°C$
- **Simplifying assumption** : no pressure drop occurs during flow of the refrigerant through the condenser and evaporator
- Mechanical efficiency of the compressor : $(\eta_m)_C = 0.83$
- Efficiency of the electric motor driving the compressor : $(\eta_{el})_C = 0.90$
- Brine temperature at inlet to and outlet from the brine cooler : $t_{b1} = -5°C$, $t_{b2} = -7°C$, respectively
- Brine specific heat : $c_b = 2.85$ kJ/kgK
- Temperature increase of cooling water within the condenser : $\Delta t_w = 7K$
- Cooling water specific heat : $c_w = 4.187$ kJ/kgK

In Fig.3 the thermodynamical cycle of the refrigerant in question is presented in *lgp-h* reference system. In cycle considerations the thermodynamical medium exoenergy can be calculated in an arbitrary state of reference. In the example in question the following state parameters were assumed :

for ammonia :

$p_r = 2.677$ bar $\qquad t_r = t_A = 20°C$

$h_r = 1743$ kJ/kg $\qquad s_r = 9.247$ kJ/kgK

for brine :

reference state temperature : $t_b^* = -5°C$

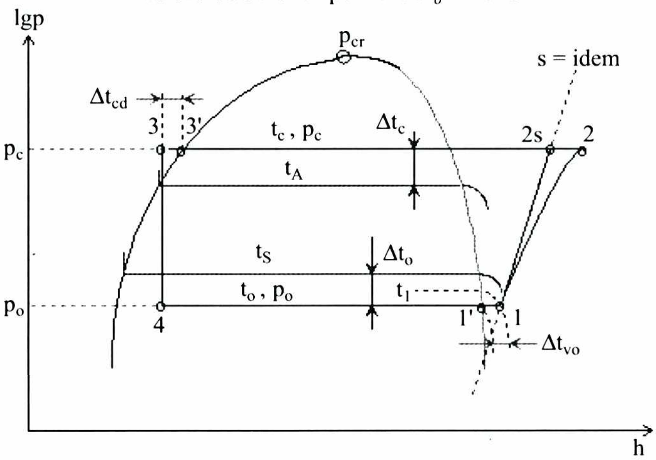

Fig.3. *One-stage, overheated thermodynamical cycle with cooling down, presented in lgp-h reference system, (in compliance with the schematic diagrams of the refrigerating systems, shown in Fig.1 and 2).*
Notice *: Δt_{cd} , Δt_{vo} - cooling down and vapour overheating of the refrigerant, respectively, P_{cr} - critical point, s - specific thermal copacitance*

In the table below values are collected of the specific enthalpy and exoenergy of ammonia and brine.

Values of ammonia and brine parameters used in the example exoergic balance of the compressor refrigerating system, according to [4]

Point no. at Fig.2	*m* Flow rate [kg/h]	*p* Pressure [bar]	*t* Temperature [°C]	*h* Specific enthalpy [kJ/kg]	*b* Excess of specific exoenergy [kJ/kg]
1	295.1	2.7	-10	1671.0	2.8
2	295.1	11	119	1947.0	222.9
3	295.1	11	25	536.0	156.6
4	295.1	2.7	-12	536.0	142.6
b1	58 800	-	-5	19.95	0
b2	58 800	-	-7	14.25	0.554

The ammonia flow rate :

$$m_r = \frac{\dot{Q}_1}{(h_{r1} - h_{r4})} = 295.1 \ \text{kg/h}$$

resulted from the energy balance of the evaporator together with the brine circuit.

The brine flow rate :

$$m_b = \frac{\dot{Q}_1}{(h_{r1} - h_{r2})} = 58800 \ \text{kg/h}$$

was calculated from the energy balance of the refrigerated space.

The heat passed back within the condenser amounted to :

$$\dot{Q}_2 = m_r(h_{r2} - h_{r3}) = 115.6 \ \text{kW}$$

Therefore the cooling water flow rate was :

$$m_w = \frac{\dot{Q}_2}{(\Delta t_w c_w)} = 14200 \ \text{kg/h}$$

The drive exoenergy of the compressor refrigerating system is equal to the power $(N_{el})_C$ of the electric motor driving the compressor. Hence the following value of $\dot{B}_d$ was obtained by applying (5) :

$$\dot{B}_d = (N_{el})_C = \frac{m_r(h_{r2} - h_{r1})}{(\eta_m)_C(\eta_{el})_C} = 30.22 \ \text{kW}$$

At elaborating the balance the brine pump working effect was neglected by assuming that the heat $\dot{Q}_1$ passed back by the brine within the brine cooler was equal to the heat $\dot{Q}_S$ absorbed by the brine from the refrigerated space. The brine enthalpy was calculated by using the temperature t_b = -12°C.

The thermal performance coefficient of the refrigerating unit ε_{th} = 3.08 resulted directly from Fig.3. Therefore the gross exoergic efficiency η_{Bg} calculated from (1) amounted to :

$$\eta_{Bg} = 0.3789$$

In order to calculate values of the exoergic efficiency η_{Bn} and η_{Bt} the magnitudes m_{wl}, b_w, N_b and N_w appearing in (2) and (3) should be determined in advance. In the considered example, however, the electric power of auxiliary equipment was neglected, i.e. $(N_{el})_t = 0$, as for the ship refrigerating systems with the condensers cooled by overboard sea water the problem of supplementing the condenser cooling water (m_{wl}, b_w) does not exist.

The ratio of the power output of the brine pump, N_b, and the drive exoenergy of the refrigerating unit, $\dot{B}_d$, can be expressed as follows :

$$\frac{N_b}{\dot{B}_d} = \frac{\Delta p_b \cdot \varepsilon_{th}}{\eta_{bt} \cdot (\eta_{el})_b \cdot \rho_b (h_{r1} - h_{r4})} \cdot \frac{m_b}{m_r} \qquad (6)$$

where :

- Δp_b – brine pressure increase within the pump
- η_{bt} – total brine pump efficiency
- $(\eta_{el})_b$ – brine pump electric motor efficiency
- ρ_b – brine density.

In the considered example the amount ratio of the brine and circulating refrigerant (ammonia) was equal to m_b/m_r = 199.3. Assuming that : Δp_b = 3 bar ; ρ_b = 1250 kg/m^3; η_{bt} = 0.7 ; $(\eta_{el})_b$ = 0.88 one obtains in accordance with (6) :

$$\frac{N_b}{\dot{B}_d} = 0.2107$$

Therefore it resulted that the brine pump power output is relatively large and in the example in question it amounted to about 21% of the drive exoenergy of the refrigerating unit. In the real refrigerating system the heat $\dot{Q}_S$ absorbed by the brine within the refrigerated space is smaller than the heat $\dot{Q}_1$ transmitted by it within the evaporator (brine cooler).

Assuming that :

⇒ the heat produced within the electric motor, determined by its effciency $(\eta_{el})_b$, flows out to the environment

⇒ the heat of friction within the pump, determined by its efficiency η_{bt}, as well as the heat of friction due to flow through piping, is fully absorbed by the brine,

assuming also that :

→ the heat produced within the electric motor of the fan, and

→ the heat of friction within the fan, as well as

→ the heat of friction due to air flow into the refrigerated space, decreases the refrigerating effect,

one can calculate the heat amount $\dot{Q}_S$ on the basis of the air cooler energy balance as follows :

$$\dot{Q}_S = \dot{Q}_1 - N_b(\eta_{el})_b - N_F = 83.0 \ \text{kW}$$

where the power output of the fan, N_F is calculated by means of (8) as :

$$N_F = 0.1462 \cdot \dot{B}_d = 4.42 \ \text{kW}$$

Therefore one can calculate, by means of (4), the exoenergy increase $\Delta\dot{B}_S$ of the refrigerated space :

$$\Delta\dot{B}_S = 6.40 \ \text{kW}$$

Basing on [4] one can state that due to absorption of the heat of friction of the brine compressed by the pump the exoenergy increase of the brine, effected within the pump, is smaller than the brine exoenergy loss due to flow drag in piping. In result, the brine exoenergy drop, $-(\Delta b_b)_S$, within the refrigerated space is smaller than the exoenergy increase $(\Delta b_b)_V$ within the evaporator, therefore :

$$-(\Delta b_b)_S = (\Delta b_b)_V + \frac{\Delta p_b}{\rho_b} \cdot \frac{1}{\eta_{bt}} \cdot \frac{T_b^* - T_A}{T_b^*} = $$
$$= 0.554 - 0.032 = 0.522 \ \text{kJ/kg}$$

To calculate the $(\Delta b_b)_S$ value, $T_b^* = T_{b1}$ = 268 K was assumed. It can be observed that the brine pump operation not only increases the total drive exoenergy of the process , but also decreases the useful exoergic effect by 0.032/0.554 = 0.0578 (i.e. about 5.8%).

The ratio of the power output of the cooling water pump, N_w, and the drive exoenergy $\dot{B}_d$, can be expressed as follows :

$$\frac{N_w}{\dot{B}_d} = \frac{\Delta p_w \cdot \varepsilon_{th}}{\eta_{wt} \cdot (\eta_{el})_w \cdot \rho_w (h_{r1} - h_{r4})} \cdot \frac{m_w}{m_r} \qquad (7)$$

where :

- Δp_w – cooling water pressure increase within the pump
- η_{wt} – total efficiency of the cooling water pump
- $(\eta_{el})_w$ – electric motor efficiency of the cooling water pump
- ρ_w – cooling water density.

In the considered example the amount ratio of the water cooling the condenser and the circulating refrigerant (ammonia) was equal to m_w/m_r = 14 200/295.1 = 48.1.

Assuming : Δp_w = 3 bar ; ρ_w = 1000 kg/m^3; η_{wt} = 0.7 ; $(\eta_{el})_w$ = = 0.88 one obtained in accordance with (7) :

$$\frac{N_w}{\dot{B}_d} = 0.0636$$

Therefore it can be concluded that in the example in question the power output of the cooling water pump amounts to about 6.4% of the drive exoenergy of the refrigerating unit.

The ratio of the power output of the fan, N_F, and the drive exoenergy $\dot{B}_d$, can be expressed as follows :

$$\frac{N_F}{\dot{B}_d}=\frac{(\Delta p)_F\cdot\varepsilon_{th}}{(\eta_t)_F\cdot\rho_a(h_{r1}-h_{r4})}\cdot\frac{m_a}{m_r} \tag{8}$$

where :

$(\Delta p)_F$ – air pressure increase within the fan
$(\eta_t)_F$ – total efficiency of the fan
ρ_a – air density.

To calculate the ratio value, the following assumptions concerning the air cooler were made :

- the amount ratio of the air and the circulating refrigerant (ammonia) was equal to m_a/m_r = 44 642/295.1 = 151.3 ; $(\Delta p)_F$ = 250 Pa ; ρ_a = 1.3 kg/m^3; $(\eta_t)_F$ = 0.54 ;
- hence in accordance with (8) the following result was obtained :

$$\frac{N_F}{\dot{B}_d}=0.1462$$

The conclusion is that in the example in question the fan power output amounts to about 14.6 % of the drive exoenergy of the refrigerating unit.

Finally, the net exoergic efficiency η_{Bn} of the considered refrigerating unit calculated on the basis of (2) was obtained :

$$\eta_{Bn}=0.1986$$

and the total exoergic efficiency of the refrigerating system accordance with (3) :

$$(\eta_{Bt})_{IRS}=0.1491$$

Example exoergic balance of the direct refrigerating system

For consideration of the other case in which the refrigerating unit directly cooperates with the air fan cooler (Fig.1) the following assumptions were made :

✓ all refrigerant (ammonia) circuit parameters were the same as in the preceding case
✓ the air fan power output N_F was the same as that of the brine air cooler
✓ the same amount ratio of the air and circulating refrigerant m_a/m_r = 151.3 at $(\Delta p)_F$ = 250 Pa ; ρ_a = 1.3 kg/m^3 ; $(\eta_t)_F$ = 0.54.

Therefore the ratio of the power output of the fan, N_F, and the drive exoenergy $\dot{B}_d$, calculated from (8) is obviously the same as in the preceding case i.e. :

$$\frac{N_F}{\dot{B}_d}=0.1462$$

It means that the air fan power output amounts also to about 14.6% of the drive exoenergy of the refrigerating unit.

In the real refrigerating system the heat $\dot{Q}_S$ absorbed by the refrigerant out of the refrigerated space is smaller than the heat $\dot{Q}_l$ absorbed in the evaporator.

Assuming that the heat emitted within the electric motor of the fan as well as the heat due to friction within the fan and that of the air streams flowing into the refrigerated space decreases the refrigerating effect, one can calculate $\dot{Q}_S$ value on the basis of the energy balance of the cooler, as follows :

$$\dot{Q}_S=\dot{Q}_l-N_F=88.614\ \text{kW}$$

Therefore in this case :

➲ the exoenergy increase $\Delta\dot{B}_S$ of the refrigerated space, calculated on the basis of (4) is:

$$\Delta\dot{B}_S=6.84\ \text{kW}$$

➲ and the total exoergic efficiency η_{Bt} of the refrigerating system, calculated accordingly by means of (3) is :

$$(\eta_{Bt})_{DRS}=0.1871$$

Hence it can be concluded that the total exoergic efficiency of the direct refrigerating system increased by as much as 25.5% against that of the indirect brine system.

INFLUENCE OF AIR COOLER DEFROSTING PROCESS ON EXOERGIC EFFICIENCY OF THE REFRIGERATING SYSTEMS

In the influence analysis of the surface-defrosting cyclic process of the air cooler installed in the refrigerating space three different, commonly used defrosting systems which apply thermal energy were taken into account [6,7,8], namely those with the use of :

⇨ electric resistance heaters (CEE)
⇨ water sprinkling the frosted surface (CNZ)
⇨ supply of heating medium ; in the considered case- - the hot refrigerant gas (CGG).

The influence analysis is based on :

- the daily thermal load of the air cooler, averaged over the time of operation of the refrigerating system, and
- accounting for daily operation time periods, namely : the refrigerating time τ_r, defrosting time τ_f and total operation time of the refrigerating system with the defrosting process $\tau_t = \tau_r + n\,\tau_f$, as well as the number of defrosting cycles per day n.

The influence consists in decreasing the heat amount absorbed out of the refrigerated space, which results from thermal losses due to defrosting process and an increase of drive exoenergy $\dot{B}_d$ of the refrigerating system.

In the case of application of **CEE** defrosting system, the influence can be described as follows :

$$(\eta_{Bf})_{CEE}=\frac{-\{\frac{\tau_r}{\tau_t}[\dot{Q}_l-N_F-N_b(\eta_{el})_b]-\frac{n\tau_f}{\tau_t}(\dot{Q}_f)_{CEE}\}\frac{T_S-T_A}{T_S}}{(\dot{B}_d+N_F+N_b+N_w)\frac{\tau_r}{\tau_t}+(N_f)_{CEE}\frac{n\tau_f}{\tau_t}} \tag{9}$$

where :

$(\eta_{Bf})_{CEE}$ – exoergic efficiency of the refrigerating system with the electric defrosting process
$(\dot{Q}_f)_{CEE}$ – flow rate of defrosting heat loss
$(N_f)_{CEE}$ – power demand of the electric defrosting system
τ_r – refrigerating time period
τ_f – defrosting time period
τ_t – total daily operation time of the refrigerating system
n – number of defrosting cycles per day.
Other notions – as explained earlier

In the case of **CNZ** defrosting system the influence can be described by (10) :

$$(\eta_{Bf})_{CNZ}=\frac{-\{\frac{\tau_r}{\tau_t}[\dot{Q}_l-N_F-N_b(\eta_{el})_b]-\frac{n\tau_f}{\tau_t}(\dot{Q}_f)_{CNZ}\}\frac{T_S-T_A}{T_S}}{(\dot{B}_d+N_F+N_b+N_w)\frac{\tau_r}{\tau_t}+(\dot{B}_f)_{CNZ}\frac{n\tau_f}{\tau_t}} \tag{10}$$

where :

$(\eta_{Bf})_{CNZ}$ – exoergic efficiency of the refrigerating system with the water sprinkling defrosting process
$(\dot{Q}_f)_{CNZ}$ – flow rate of defrosting heat loss
$(\dot{B}_f)_{CNZ}$ – defrosting exoenergy.

In the case of application of **CGG** defrosting system the influence can be described by (11) :

$$(\eta_{Bf})_{CGG} = \frac{-\{\frac{\tau_r}{\tau_t}[\dot{Q}_l - N_F - N_b(\eta_{el})_b] - \frac{n\tau_f}{\tau_t}(\dot{Q}_f)_{CGG}\}\frac{T_S - T_A}{T_S}}{(\dot{B}_d + N_F + N_b + N_w)\frac{\tau_r}{\tau_t} + (\dot{B}_f)_{CGG}\frac{n\tau_f}{\tau_t}} \tag{11}$$

where :

$(\eta_{Bf})_{CGG}$ – exoergic efficiency of the refrigerating system with the refrigerant gas defrosting process

$(\dot{Q}_f)_{CNZ}$ – flow rate of defrosting heat loss

$(\dot{B}_f)_{CNZ}$ – defrosting exoenergy.

Values of the exoergic efficiency and relative exoenergy losses for three above mentioned defrosting systems applied to the air coolers of the above analyzed refrigerating systems are below presented by using appropriate examples in which the following earlier determined data have been used :

- Refrigerating time period : τ_r = 16 h/day
- Time of defrosting cycle : τ_f = 0.5 h
- Number of defrosting cycles per day : n = 2 cycles/day
- Total daily operation time of the refrigerating system with the defrosting process : τ_t = 17 h/day
- Refrigerating unit output measured at the evaporator : $\dot{Q}_l$ = 93.03 kW
- Power output of the fan acc. (8) : $N_F = 0.1462 \cdot \dot{B}_d$ = 4.42 kW
- Brine pump power output acc. (6) : $N_b = 0.2107 \cdot \dot{B}_d$ = 6.37 kW
- Brine pump electric motor efficiency : $(\eta_{el})_b$ = 0.88
- Temperature of the refrigerated space : T_S = 272 K
- Ambient temperature : T_A = 293 K
- Drive exoenergy of the refrigerating system acc. (5) : $\dot{B}_d$ = 30.22 kW
- Power output of the cooling water pump acc. (7) : $N_w = 0.0636 \cdot \dot{B}_d$ = 1.32 kW

Example of influence of the electric defrosting system (CEE)

The example CEE defrosting system was considered which had the following parameters and operational data, assumed from an existing electric resistance heater :

- air cooler area :	A = 464 m^2
- air flow rate :	V = 34 340 m^3/h
- drive power output of the fan :	N_F = 4.42 kW
- electric power output of the CEE resistance system :	$(N_f)_{CEE}$ = 35.1 kW

The amount of the heat, $(\dot{Q}_f)_{CEE}$, emitted by the defrosting system to the refrigerated space can be calculated under the assumption that the defrosting system efficiency is rather low and 70% of its power output is lost [6,7,8], as follows :

$$(\dot{Q}_f)_{CEE} = 0.7n\tau_f(N_f)_{CEE} = 24.57 \text{ kWh}$$

After averaging the absorbed heat flow and the drive exoenergy delivered during refrigerating process of the refrigerated space, one calculates - by means of (9) - appropriate average values of the exoergic efficiency related to the total operational time of 17 h, to obtain the results as follows :

for the direct refrigerating system

$$(\eta_{Bf})_{CEE}^{DRS} = 0.1734$$

for the indirect refrigerating system

$$(\eta_{Bf})_{CEE}^{IRS} = 0.1393$$

If the exoergic efficiency of the system fitted with defrosting device is compared to that without defrosting then the following values of the ratio are achieved :

for the direct refrigerating system

$$\frac{(\eta_{Bf})_{CEE}}{(\eta_{Bt})_{DRS}} = \frac{0.1734}{0.1871} = 0.9268$$

for the indirect refrigerating system

$$\frac{(\eta_{Bf})_{CEE}}{(\eta_{Bt})_{IRS}} = \frac{0.1393}{0.1491} = 0.9343$$

It means that the electric defrosting process of the air cooler leads to lowering the total exoergic efficiency and the drop of its value amounts to nearly 7.3% for the direct refrigerating system, and to 6.6% for that indirect.

Example of influence of the water sprinkling defrosting system (CNZ)

It was assumed that :

- ✓ sea water is used for defrosting the air cooler surface, and
- ✓ power demand of the defrosting (i.e. $(N_f)_{CEE}$ = 35.1 kW) as well as the other data are the same as in the preceding case.

Under those assumptions the amount of the heat, $(\dot{Q}_f)_{CNZ}$, emitted by the defrosting system to the refrigerated space would be the same as for the CEE system, i.e. 24.57 kWh.

To determine the exoenergy $(\dot{B}_f)_{CNZ}$ of the surface sprinkling water defrosting system it is necessary first to calculate the amount of water used in the defrosting process. It can be obtained on the basis of the energy balance as follows :

$$m_w = (N_f)_{CEE}\frac{n \cdot \tau_f}{\Delta t_w \cdot c_w}$$

Assuming that c_w = 4.187 kJ/kgK, and Δt_w = 15 K, one obtains :

$$m_w = 2011.9 \text{ kg}$$

As amount of the sprinkling water is small the pump power out-put can be neglected, and the exoenergy of the refrigerating system with the water sprinkling defrosting process can be expressed as follows :

$$(\dot{B}_f)_{CNZ} = \frac{m_w \Delta b_w}{\tau_t}$$

however the relatively small value of the specific exoenergy difference Δb_w makes the assumption of $(\dot{B}_f)_{CNZ} = 0$ justified.

Finally, the following average values of the exoergic efficiency related to the total operational time of 17 h are obtained by means of (10) :

for the direct refrigerating system

$$(\eta_{Bf})_{CNZ}^{DRS} = 0.1838$$

for the indirect refrigerating system

$$(\eta_{Bf})_{CNZ}^{IRS} = 0.1464$$

Comparing now the exoergic efficiency of the refrigerating system with the water sprinkling defrosting system and that of the same system without any defrosting, one obtains the following values of this ratio :

for the direct refrigerating system

$$\frac{(\eta_{Bf})_{CNZ}}{(\eta_{Bt})_{DRS}} = \frac{0.1838}{0.1871} = 0.9824$$

for the indirect refrigerating system

$$\frac{(\eta_{Bf})_{CNZ}}{(\eta_{Bt})_{IRS}} = \frac{0.1464}{0.1491} = 0.9820$$

Hence it is concluded that the water sprinkling defrosting process of the air cooler can lead to lowering the total exoergic efficiency by about 1.8% for both refrigerating systems in question.

Example of influence of the hot refrigerant gas defrosting system (CGG)

The CGG system is applicable only to the direct refrigerating systems.

As the power demand of the defrosting system $(N_f)_{CEE}$ = 35.1 kW) as well as the other relevant data were assumed the same as in the above considered cases, the amount of the heat $(\dot{Q}_f)_{CGG}$ emitted to the refrigerated space by the CGG system is also of the same value, i.e. 24.57 kWh.

The amount of ammonia necessary for the defrosting process, determined from the energy balance of the air cooler, can be expressed as :

As the earlier given values (see table) of the ammonia specific enthalpy are :

$$(h_{rg})_2 = 1947 \text{ kJ/kg and } (h_{rg})_3 = 536 \text{ kJ/kg}$$

hence :

$$m_{rg} = \frac{(N_f)_{CEE} \cdot n \cdot \tau_f}{(h_{rg})_2 - (h_{rg})_3} = 89.55 \text{kg}$$

Next, the exoenergy of the CGG system is calculated as follows :

$$(\dot{B}_f)_{CGG} = \frac{[(b_{rg})_2 - (b_{rg})_3]}{\tau_t} m_{rg}$$

therefore for the ammonia specific exoenergy $(b_{rg})_2$ = 222.9 kJ/kg and $(b_{rg})_3$ = 156.6 kJ/kg (see table) it yields :

$$(\dot{B}_f)_{CGG} = 97 \text{ W}$$

and the following average value of the exoergic efficiency of the direct refrigerating system , related to the total operational time of 17 h is obtained by means of (11) :

$$(\eta_{Bf})_{CGG}^{DRS} = 0.1833$$

Finally, the ratio of the exoergic efficiency of the direct refrigerating system with the hot refrigerant gas defrosting system to that of the same system without any defrosting, is obtained :

$$\frac{(\eta_{Bf})_{CGG}}{(\eta_{Bt})_{DRS}} = \frac{0.1833}{0.1871} = 0.9797$$

In Fig. 4 to 7 the exoergic efficiency η_{Bf} and the relative one η_{Bf}/η_{Bt} of both refrigerating systems in question, analyzed with taking into account the three considered defrosting processes, is presented in function of the number of defrosting cycles *n*.

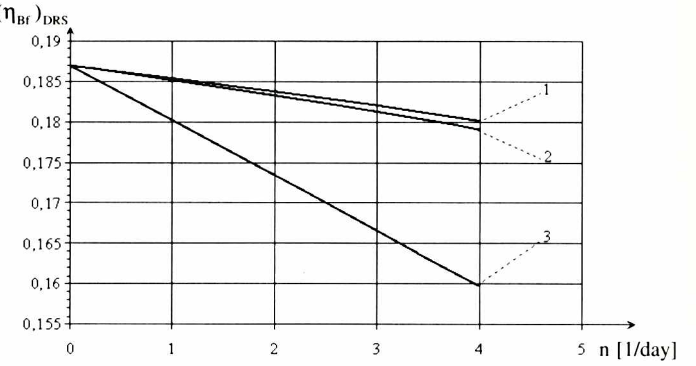

Fig.4. *Results of example calculations of the exoergic efficiency of the direct refrigeration system in function of the number of defrosting cycles n for three air cooler defrosting systems : 1 – of water sprinkling, 2 – of hot refrigerant gas, 3 – of electric heating*

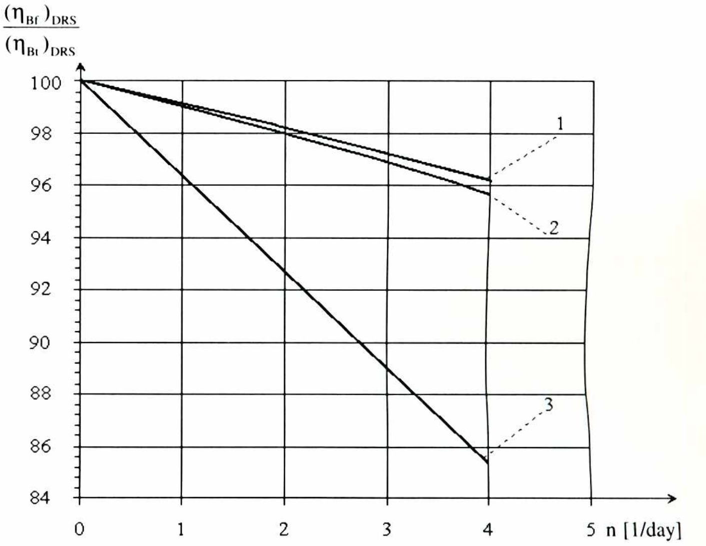

Fig.5. *Results of example calculations of the relative exoergic efficiency of the direct refrigeration system in function of the number of defrosting cycles n for three air cooler defrosting systems : 1 – of water sprinkling, 2 – of hot refrigerant gas, 3 – of electric heating*

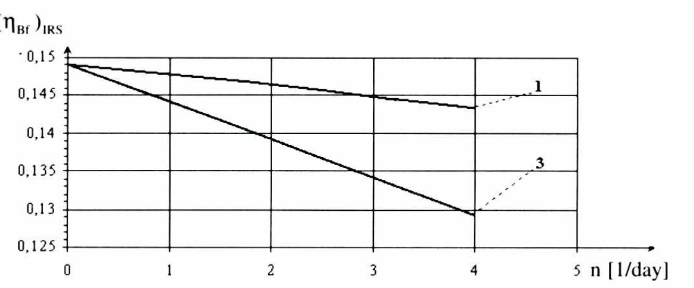

Fig. 6. Results of example calculations of the exoergic efficiency of the indirect refrigeration system in function of the number of defrosting cycles n for two air cooler defrosting systems : 1 – of water sprinkling, 3 – of electric heating

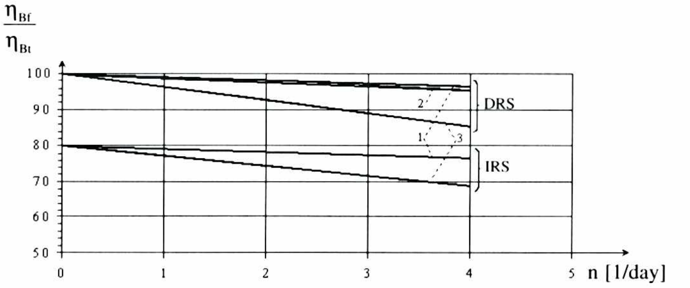

Fig. 7. *Results of example calculations of the relative exoergic efficiency of the direct and indirect refrigeration systems in function of the number of defrosting cycles n for three air cooler defrosting systems : 1 – of water sprinkling, 2 – of hot refrigerant gas, 3 – of electric heating*

Comments on the calculation results

- As it results from Fig.4 and 5 the hot refrigerant gas defrosting process applied to the direct refrigerating system leads, at n = 2, to lowering the total exoergic efficiency of the system by about 2%.
- Advantages of the sea-water sprinkling defrosting or the hot refrigerant gas defrosting of the air cooler are obvious as they cause only a small drop of the exoergic efficiency of the direct refrigerating system : by about 1% at n = 1 to about 4% at n = 4.
- However if the electric heating defrosting system is applied at n = 4 the exoergic efficiency of the entire system substantially drops (by 14.5%).
- The relative exoergic efficiency of the indirect refrigerating system is much lower than that of the direct system (see Fig.7). Moreover the efficiency shows an additional important decrease if the electric heating defrosting system is applied, and such drop is much lower if the sea-water sprinkling method is used. Nevertheless in the case of the indirect brine refrigerating system the loss impairs its not very high total exoergic efficiency η_{Bt}.
- As it results from the course of η_{Bf} versus *n* (Fig. 4 and 6) the extreme difference of the efficiency of the direct refrigerating system with the water sprinkling defrosting and that of the indirect one with the electric heating defrosting, reaches 27%, at the same value of n = 4. In practice however the electric heating defrosting method is often applied to the direct systems, and the water sprinkling one - in the indirect brine systems ; hence the difference between the efficiency values of both systems drops to about 9% at n = 4.
- In the analyzed cases the share of the electric defrosting system work in the total exoergic efficiency of the entire refrigerating system reaches 14.5%. It should be added that this result is valid only for the assumed value of the specific defrosting power coefficient of 75.6 W/m^2 which is rather low as its practical values are usually contained within 70÷300 W/m^2 range [8].

CONCLUSIONS

On the basis of the performed comparisons of the exoergic efficiency of the different example designs of the refrigerating systems it can be stated that :

- the exoergic efficiency of the direct refrigerating system is by at least 25% higher than that of the indirect brine refrigerating system
- the sea-water sprinkling defrosting system which is advantageous for marine applications insignificantly lowers the exoergic efficiency of the entire system
- the hot refrigerant gas defrosting system and the water sprinkling one has a rather small influence on the refrigerating system exoergic efficiency
- the electric heating defrosting system highly impairs (by about 7%) the exoergic efficiency of the entire system; hence it seems that such defrosting system should be excluded from practical applications.

In spite of that the performed exoergic analyses show superiority of the direct refrigerating systems over the indirect ones, the indirect brine systems are expected to be still in use on ships due to other economical and technical reasons. This is first of all connected with a lack of international regulations on limitation of energy-wasteful designs applied in many engineering branches.

Appraised by Edward Parcchaladze, Assoc.Prof.,D.Sc.

NOMENCLATURE

b – specific exoenergy
$\dot{B}_d$ – driving exoenergy
CEE – electric defrosting system
CNZ – water sprinkling defrosting system
h – specific enthalpy
B – exoenergy
c – specific heat
CGG – refrigerant gas defrosting system
m – mass flow rate
n – number of defrosting cycles per day
N – power
N_{el} – power of driving electric motor
p – pressure
$\dot{Q}$ – heat flow rate
s – specific thermal capacitance
t – temeprature [°C]
T – temperature [K]
t*, T* – reference state temperature
ε_{th} – thermal coefficient of performance
η – efficiency
η_B – exoergic efficiency
η_{Bt}, η_{Bf} – total exoergic efficiency of the refrigerating system, without defrosting process and incuding it, respectively
η_{el} – efficiency of driving electric motor
η_m – mechanical efficiency
ρ – density
τ – time

Indices :

a – air
A – ambient
b – brine
c – condensation
C – compressor
CEE – electric defrosting system
d – driving
DRS – direct refregerating system
el – electric motor
f – defrosting
F – fan
g – gross
IRS – indirect refrigerating system
m – mechanical
n – net value
o – evaporating
r – refrigerant, refrigerating
rg – refrigerant gas
S – refrigerated space (heat source)
t – total
V – evaporator
w – water
wl – loss of supplementing water

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prepared by S. Filin, D.Sc., B. Zakrzewski, D.Sc., and A. Owsicki, M.Sc., the representatives of Technical University of Szczecin,Poland.