EFFICIENCY OF HEAT RECOVERY FROM EXHAUST VENTILATION AIR OF UNDERGOUND MINES

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***Abstract**: On production shafts intake ventilation air reaches underground, but also compressed air, service water and drinking water pipelines are placed too, and these that can freeze in winter. As a result usually the intake ventilation air is heated using electrical resistance heating. On the other hand exhaust ventilation air will disperse chemical and physical contaminants like gases, dusts, humidity and heat. Depending on mine conditions temperature of exhaust mine can be significantly higher then temperature of intake air. This paper studies the possibilities of recovering heat from exhaust ventilation air using heat pumps. For intake ventilation air is needed only in winter, another heat consumer was identified with heat demand all year round, namely domestic hot water production. Efficiency of heating air using heat pumps was found to be 6.18 times higher then using electrical resistance heating, while figures for hot water production are 2.81 to 5.96.*

***Keywords**: axial fan, heat recovery, heat pumps, efficiency*

1. Introduction

Usually on production shafts intake ventilation air reaches underground, but in production shafts compressed air, service water and drinking water pipelines are placed too. In harsh winter these pipelines can freeze. As a result usually the intake ventilation air is heated, using electric resistance heating, which can be an expensive way to heat air. In industry there are many sources of waste heat like combustion exhausts, process exhausts, hot gases from drying ovens, cooling water and other similar sources.

In the mining industry we can identify two main source of waste heat: compressor cooling water and ventilation exhaust air. In Romanian mining industry due to reduced activity large turbo compressors are replaced with smaller ones resulting smaller amount of waste heat. In this paper heat recovery from mine exhaust air is studied.

2. Problem formulation

In mines allowed temperatures at workplaces are set by regulations, in our case the maximum allowed temperature is 25 °C [1]. The temperature lift is caused by the following processes: heat due to oxidation processes 31.5 to 25.6%; heat from the surrounding rock 44.6 to 52.2%; heat from underground transportation systems 8.8 to 9.3 9%; heat released from coal cut from 8.3 to 9.1%; other sources of release 4.9 to 6.5%. Add to this the geothermal temperature gradient.

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As shown, due to factors listed above, the exhaust air will be warmer than the intake ventilation air. Depending on air flow rate a great amount of heat is released in the surroundings.

Heat recovery is a major concern all over the world since primary energy recourses are declining. In this context guidelines are developed [2] in order to help experts in various industry fields to find the best ways for heat recovery. According to this guide the ventilation exhaust air from mines is a low temperature waste heat source.

For this kind of source the best way to recover heat is using [3] [4] mechanical vapor compression heat pumps, Fig. 1.

All heat pumps perform the same three functions: receipt of heat from the waste-heat source; increase of the waste-heat temperature; delivery of the useful heat at the elevated temperature.

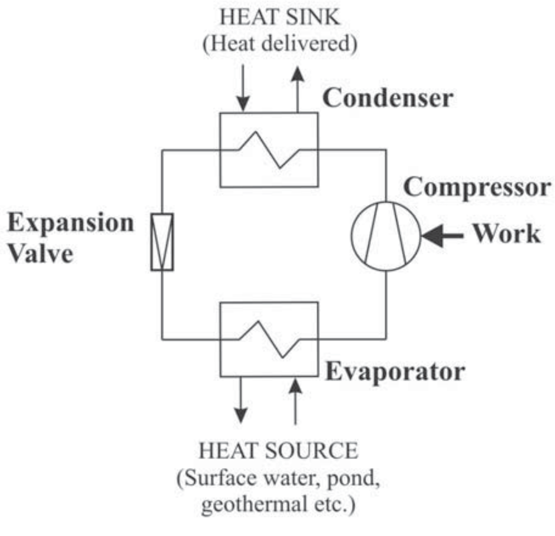

***Fig. 1.** Schematic of heat pump.*

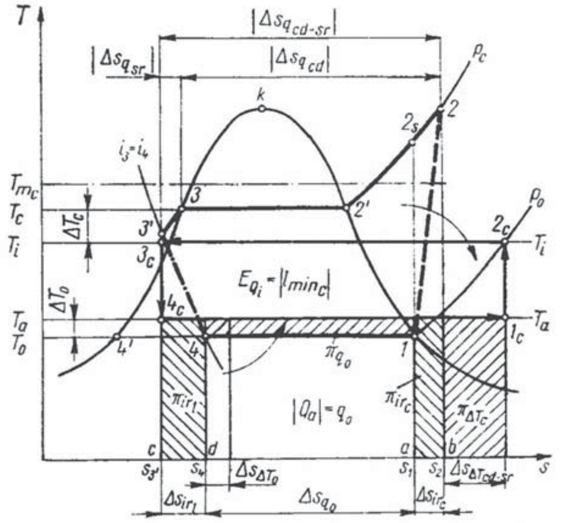

Fig. 2. Working cycle of heat pump [4].

In Fig. 2 the temperature–entropy diagram of the vapor-compression cycle is presented.

Waste heat is delivered to the heat-pump evaporator in which the heat-pump working fluid is vaporized. In case of water source heat pumps, the heat for evaporator is drawn from surface water or pond using loops. As known, in ponds even if their surface is icy, temperature at the bottom can be only as low as 4 °C, while in study source heat will be warm ventilation exhaust air.

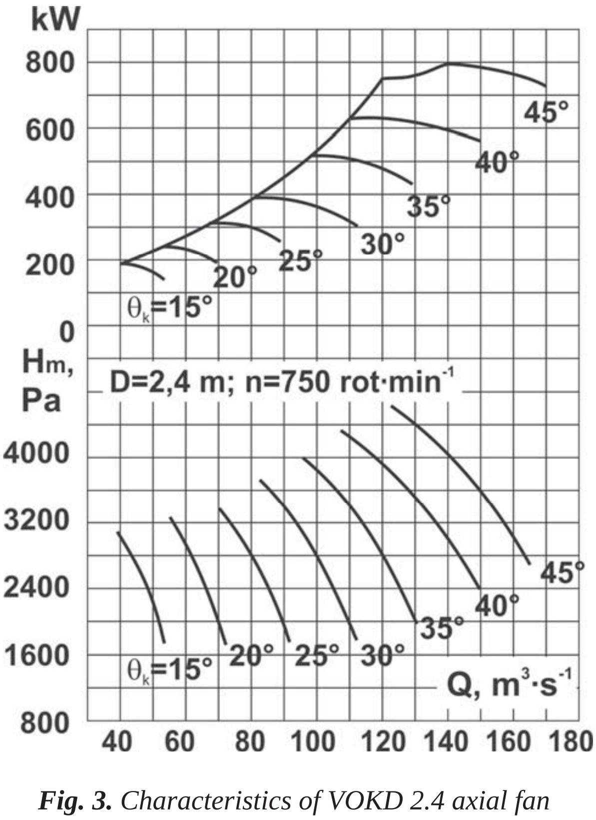

Fig. 3. Characteristics of VOKD 2.4 axial fan

Most of the main ventilation fans used in Romanian mines axial type soviet construction VOKD fans. In Fig. 3 characteristics of VOKD 2.4 fan is presented [5]. As depicted in Fig. 3 the flow rate of these fans can be adjusted by adjusting the angle of blades.

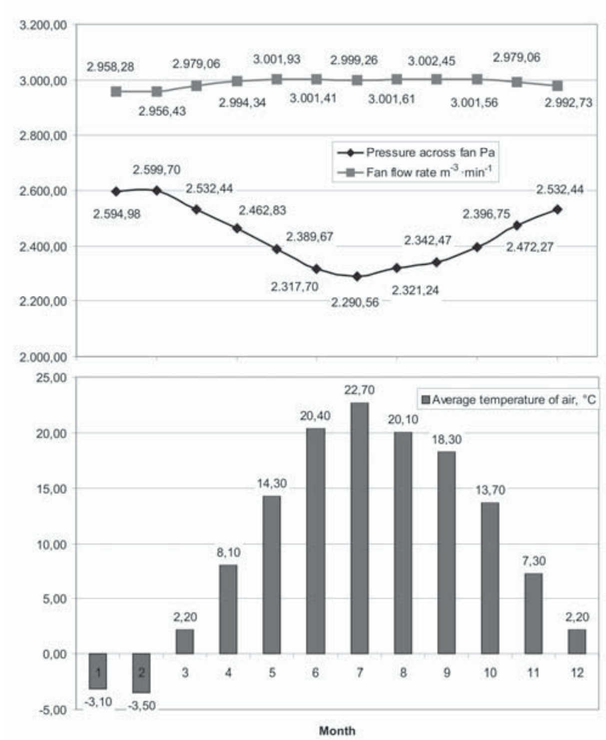

Fig. 4. Average fan flow rate and pressure across fan depending on monthly average temperatures.

Data regarding the operation of a VOKD type axial fan is presented in Fig. 4, for different monthly average temperatures.

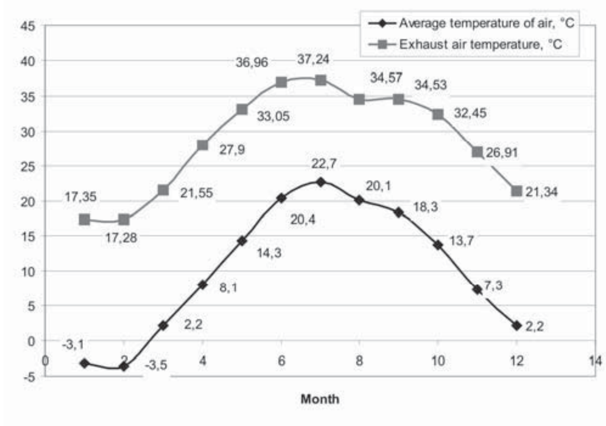

Fig. 5. Temperature of shale air at main ventilation shaft.

Analyzing data in Fig. 5, can be seen that in winter, temperature rise of ventilation air is higher since cold air is starting to warm almost instantly as he enters the production shaft from the surrounding rocks, while in summer this process is inversed since the air is hotter than the surrounding rocks, resulting a cooling effect, so temperature rise is not so high.

Finding a heat source is only one aspect of heat recovery, the other one equally important is to find consumers for the heat being recovered.

Mine ventilation is a continuous activity resulting waste heat all the year, while heating intake ventilation air is a current task only in winter and in cooler autumn and spring nights. For using the heat that can be recovered another heat consumer must be found with heat demand for the entire year.

Fortunately mining activities are continuous type activities demanding hot water for bathing and cafeteria at temperatures of 60 °C to avoid legionella disease.

The heat required for raising ventilation air temperature can be calculated using formula [6]:

$$Q = \dot{V} \cdot c_p \cdot \rho_a \cdot \left(t_f - t_{ext}\right), \quad \frac{kJ}{h} \tag{1}$$

where $\dot{V}$ is the air flow rate in the production shaft in $m^3 \cdot h^{-1}$; c_p is the specific heat capacity of air in $kJ \cdot (kg \cdot °C)^{-1}$; ρ_a is the density of air in $kg \cdot m^{-3}$ at t_{ext} ambient temperature in °C; t_f is the final temperature of air in °C that must not exceed 70 °C.

The heat demand for daily hot water consumption can be calculated considering an average of 1,000 miners. According to regulations, hot water consumption at a temperature of 60 ºC is 0.09 m^3 per person per day for bathing, and 0.0075 m^3 per person per day in cafeterias for 1 meal [7], resulting a daily demand of 29.640,79 MJ.

Table 1. *Average heat demand for ventilation air heating*

Month	Fan flow rate $m^{-3} \cdot min^{-1}$	Average air temperature, °C	Barometric pressure, mmHg	RH, %
1	2,958.28	-3.1	710.0	84
2	2,956.43	-3.5	711.6	86
3	2,979.06	2.2	699.9	86
4	2,994.34	8.1	700.0	88
11	2,992.73	7.3	708.2	80
12	2,979.06	2.2	708.2	84
Worst case	2,861.31	-18	710	40

continued

Month	Enthalpy of air, $kJ \cdot kg^{-1}$	Density $kg \cdot m^{-3}$	Enthalpy of heated air $kJ \cdot kg^{-1}$	Heat demand, $MJ \cdot s^{-1}$
1	3.441	1.219	27.385	1.439
2	2.956	1.224	27.762	1.496
3	12.603	1.178	28.061	0.904
4	24.307	1.152	28.482	0.240
11	21.062	1.169	26.593	0.323
12	12.24	1.192	27.429	0.899
Worst case	-17.264	1.292	18.256	2.189

In Table 1 the average heat demand is calculated for heating atmospheric air from average monthly temperature to 10 °C, and the heat demand for the worst case scenario, using the rated outdoor temperature for cold season -18 °C (for Jiu Valley region) [8]. In the worst case scenario the heat demand is highest. In this case, exhaust temperature of ventilation air can be approximated to 5 °C since the cold air will acquire heat from surrounding rocks [9] and the moisture from underground mine will condensate causing an additional increase of temperature due to latent heat of vaporization.

3. Case study

Heat demand computed above is supplied in a 24 h period of time; as a result, the heat load of the heat exchanger will be 14.3 kW (considering 6 hours shifts for miners and 40% of miners working on first shift). A heat exchanger is selected with rated heat load of 15 kW, 16 bar operating pressure, inlet outlet temperatures for cold fluid side 10/60 °C, inlet outlet temperatures for hot fluid side 80/60 °C, pressure drop hot side/cold side 0.41/0.07 bar.

Since heat required for ventilation air heating is higher than the heat needed for hot water two scenarios will be considered for calculating heat recovery efficiency:

- Heating ventilation air;
- Preparing hot water.

In the case of air heating additional data for calculations are: T_i=10 °C – temperature of heated air, T_a=5 °C – ambient temperature (temperature of exhausted ventilation air), ΔT_c=15 °C – temperature difference required for heat transfer in condenser (heat delivery), ΔT_0= 15 °C – temperature difference required for heat transfer in evaporator, ΔT_{sr}=5 °C – temperature difference for sub-cooling, mechanical efficiency η_{em}= 0.9, employed refrigerant R717 (ammonia).

Results are presented in Table 2, where high coefficient of performance (COP) can be observed.

__Table 2.__ Efficiency of air heating using heat pump.

Nom.	Worst situation
Condenser outlet temperature, °C	10
Heat source temperature, °C	5
Cycle compression work **l**, kJ·kg^{-1}	196.222
Work supplied to compressor $\mathbf{P_e}$, kW	353.867
Evaporator heat transferred $\mathbf{q_0}$, kJ·kg^{-1}	1,155.41
Condenser heat transferred $\mathbf{q_c}$, kJ·kg^{-1}	1,352.54
Work of Ideal Carnot cycle $\mathbf{l_{minC}}$, kJ·kg^{-1}	23.877
LOSS due to irreversibility of:	
Theoretical COP $\boldsymbol{\mu}$	6.87
Practical COP $\boldsymbol{\mu_e}$	6.18
Compression $\boldsymbol{\pi_{irc}}$, kJ·kg^{-1}	17.798
Expansion $\boldsymbol{\pi_{irl}}$, kJ·kg^{-1}	7.554
Heat transfer in evaporator $\boldsymbol{\pi_{q0}}$, kJ·kg^{-1}	65.860
Heat transfer in condenser $\boldsymbol{\pi_{\Delta Tc}}$, kJ·kg^{-1}	81.752

This high COP, higher than 1.6 listed in literature [10], results as the temperature lift is smaller, only 5 °C, while in the cited case study the heat pump temperature lift was from around -25 °C ambient temperature to 20 °C temperature needed in houses, namely 45 °C.

In case of hot water preparation data required for calculation are: Q_i=15 kW – required heat delivery; T_i=60 °C – temperature of hot water, T_a– ambient temperature (temperature of exhausted ventilation air) corresponding values in Fig. 2, ΔT_c=10 °C – temperature difference required for heat transfer in condenser (heat delivery), ΔT_0= 10 °C – temperature difference required for heat transfer in evaporator, ΔT_{sr}=10 °C – temperature difference for sub-cooling, mechanical efficiency η_{em}= 0.9, employed refrigerant R717 (ammonia).

Results are presented in Table 3, 4, 5 and 6 for lowest average temperature (February), for highest average temperature (July), for a temperature close to the average temperature of the two months, namely October, and for worst case scenario (-18 °C ambient temperature).

__Table 3.__ Heat pump efficiency for medium temperatures of February

Nom.	February
Condenser outlet temperature, °C	60
Heat source temperature, °C	17.28
Cycle compression work **l**, kJ·kg^{-1}	335.327
Work supplied to compressor $\mathbf{P_e}$, kW	4.247
Evaporator heat transferred $\mathbf{q_0}$, kJ·kg^{-1}	980.709
Condenser heat transferred $\mathbf{q_c}$, kJ·kg^{-1}	1,316.036
Theoretical COP $\boldsymbol{\mu}$	3.92
Practical COP $\boldsymbol{\mu_e}$	3.53
Work of Ideal Carnot cycle $\mathbf{l_{minC}}$, kJ·kg^{-1}	168.756
LOSS due to irreversibility of:	
Compression $\boldsymbol{\pi_{irc}}$, kJ·kg^{-1}	41.001
Expansion $\boldsymbol{\pi_{irl}}$, kJ·kg^{-1}	22.516
Heat transfer in evaporator $\boldsymbol{\pi_{q0}}$, kJ·kg^{-1}	34.972
Heat transfer in condenser $\boldsymbol{\pi_{\Delta Tc}}$, kJ·kg^{-1}	68.603

__Table 4.__ Heat pump efficiency for medium temperatures of July

Nom.	July
Condenser outlet temperature, °C	60
Heat source temperature, °C	37.24
Cycle compression work **l**, kJ·kg^{-1}	187.559
Work supplied to compressor $\mathbf{P_e}$, kW	2.637
Evaporator heat transferred $\mathbf{q_0}$, kJ·kg^{-1}	997.914
Condenser heat transferred $\mathbf{q_c}$, kJ·kg^{-1}	1,185.473
Theoretical COP $\boldsymbol{\mu}$	6.32
Practical COP $\boldsymbol{\mu_e}$	5.69
Work of Ideal Carnot cycle $\mathbf{l_{minC}}$, kJ·kg^{-1}	80.987
LOSS due to irreversibility of:	
Compression $\boldsymbol{\pi_{irc}}$, kJ·kg^{-1}	18.018
Expansion $\boldsymbol{\pi_{irl}}$, kJ·kg^{-1}	8.771
Heat transfer in evaporator $\boldsymbol{\pi_{q0}}$, kJ·kg^{-1}	33.221
Heat transfer in condenser $\boldsymbol{\pi_{\Delta Tc}}$, kJ·kg^{-1}	47.862

__Table 5.__ Heat pump efficiency for medium temperatures of October

Nom.	October
Condenser outlet temperature, °C	60
Heat source temperature, °C	34.53
Cycle compression work **l**, kJ·kg^{-1}	204.807
Work supplied to compressor $\mathbf{P_e}$, kW	2.843
Evaporator heat transferred $\mathbf{q_0}$, kJ·kg^{-1}	995.944
Condenser heat transferred $\mathbf{q_c}$, [kJ·kg^{-1}]	1,200.752
Theoretical COP $\boldsymbol{\mu}$	5.86
Practical COP $\boldsymbol{\mu_e}$	5.28
Work of Ideal Carnot cycle $\mathbf{l_{minC}}$, kJ·kg^{-1}	91.799
LOSS due to irreversibility of:	
Compression $\boldsymbol{\pi_{irc}}$, kJ·kg^{-1}	20.829
Expansion $\boldsymbol{\pi_{irl}}$, kJ·kg^{-1}	10.170
Heat transfer in evaporator $\boldsymbol{\pi_{q0}}$, kJ·kg^{-1}	33.457
Heat transfer in condenser $\boldsymbol{\pi_{\Delta Tc}}$, kJ·kg^{-1}	49.637

Table 6. *Heat pump efficiency for -18 °C ambient temperatures*

Nom.	Worst case
Condenser outlet temperature, °C	60
Heat source temperature, °C	5
Cycle compression work **l**, kJ·kg^{-1}	445.864
Work supplied to compressor $\mathbf{P_e}$, kW	5.339
Evaporator heat transferred $\mathbf{q_0}$, kJ·kg^{-1}	967.089
Condenser heat transferred $\mathbf{q_c}$, kJ·kg^{-1}	1,422.953
Theoretical COP $\boldsymbol{\mu}$	3.12
Practical COP $\boldsymbol{\mu_e}$	2.81
Work of Ideal Carnot cycle $\mathbf{l_{minC}}$, kJ·kg^{-1}	234.916
LOSS due to irreversibility of:	
Compression $\boldsymbol{\pi_{irc}}$, kJ·kg^{-1}	59.790
Expansion $\boldsymbol{\pi_{irl}}$, kJ·kg^{-1}	34.349
Heat transfer in evaporator $\boldsymbol{\pi_{q0}}$, kJ·kg^{-1}	36.065
Heat transfer in condenser $\boldsymbol{\pi_{\Delta Tc}}$, kJ·kg^{-1}	90.947

Mechanical losses are not listed in table since they were considered constant.

4. Conclusions

Analyzing results presented in Tables 3 to 6 regarding efficiency of producing hot water from ventilator exhaust heat, following conclusions can be drawn:

- COP of heat pump increases as temperature of heat source increases. Since the temperature of condenser is fixed COP will increase as temperature lift will decrease. COP varies from 2.81 in case of 5 °C (55 °C temperature lift) to 5.69 for 37.24 °C (22.76 °C temperature lift) temperature of heat source. Data is consistent with those found in Table 2 where for temperature lift of 5 °C the COP is 6.18.

- according to variation of COP the work supplied to compressor P_e, decreases from 5.339 kW for 5 °C to 2.637 kW for 37.24 °C temperature of heat source.

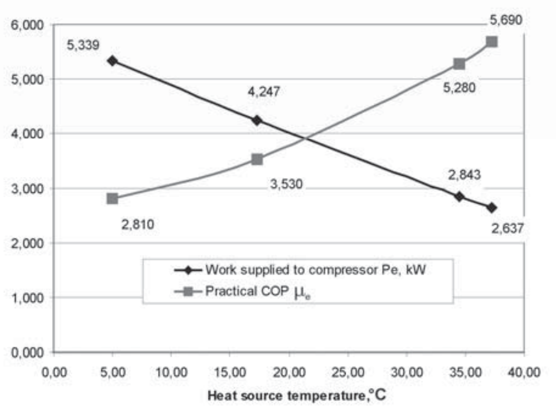

Fig. 6. *Work supplied to compressor and COP depending on heat source temperature*

Above analyzed data are shown in Fig. 6 where the trend of COP and the work supplied to compressor P_e, can be seen.

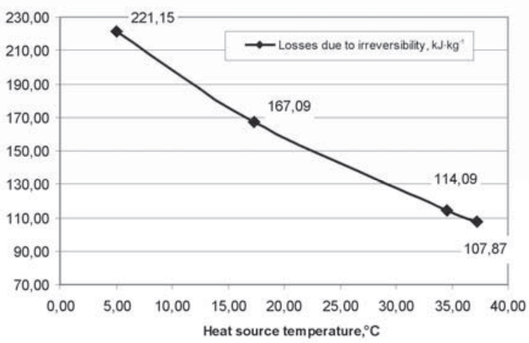

Fig. 7. *Total losses due to irreversibility of processes in heat pump.*

In Fig. 7 total losses due to irreversibility are shown. Losses have the same trend with work supplied to compressor P_e, decreasing as COP increases with increasing heat source temperature.

Current solution for heating ventilation air that is provided through production shaft is by electric resistance heating. In order to compare efficiency of the studied cases let's assume that hot water is produced employing same method

As all the incoming electric energy is converted to heat, electric resistance heating can be considered 100% energy efficient [11], resulting that the electrical energy needed to heat the ventilation air is 2.189 MWh while hot water production will use 15 kWh.

By definition COP is a ratio of heating or cooling provided to electrical energy consumed.

Comparing the required energy with the energy provided using heat pumps, and considering the definition of COP, results that energy demand will be smaller 6.18 times for air heating and between 2.81 to 5.69 times for hot water production.

One drawback of air source heat pumps is the occurrence of frost on the evaporator for low temperatures.

This issue has been solved using heat pumps with defrost cycles along with electric resistance that tempers drop in supply air temperature during defrost cycles but does not explicitly defrost.

Even if heat pumps are expensive equipment, the amount of energy saved makes them a good alternative for heating, and recovering waste heat.

Study [11] highlights that air source heat pumps have paybacks less than 5 years vs. fuel oil or electric baseboard heating, in case of household use. Conclusively using heat pumps has to be considered for waste heat recovery and heating, since they provide both primary resources saving and money savings.

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