ITMO UNIVERSITY

# Improving performance of cascade refrigeration system using low GWP refrigerants for some cities in Russia

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April 2021

Introduction

Modeling

Validation

Optimization

Results

Conclusion

# Source of Heat pump

- Air
- Waste water
- Ground

**Air:**

- Not useful in cold climate condition
- Low cost of installation
- Not need to drill
- Capable to use in residential and commercial buildings

**Waste water / Ground:**

- Useful in cold climate condition
- High cost of installation
- Need to drill
- Capable to use in residential and commercial buildings

## Optimization of performance

- Max COP
- Max exergy efficiency
- Min total cost

Parameters

Evaporator and condenser temperature, power consumption, Cooling or heating capacity, climate conditions, soil thermal properties, soil depth,

List of the studies about optimization of heat pumps

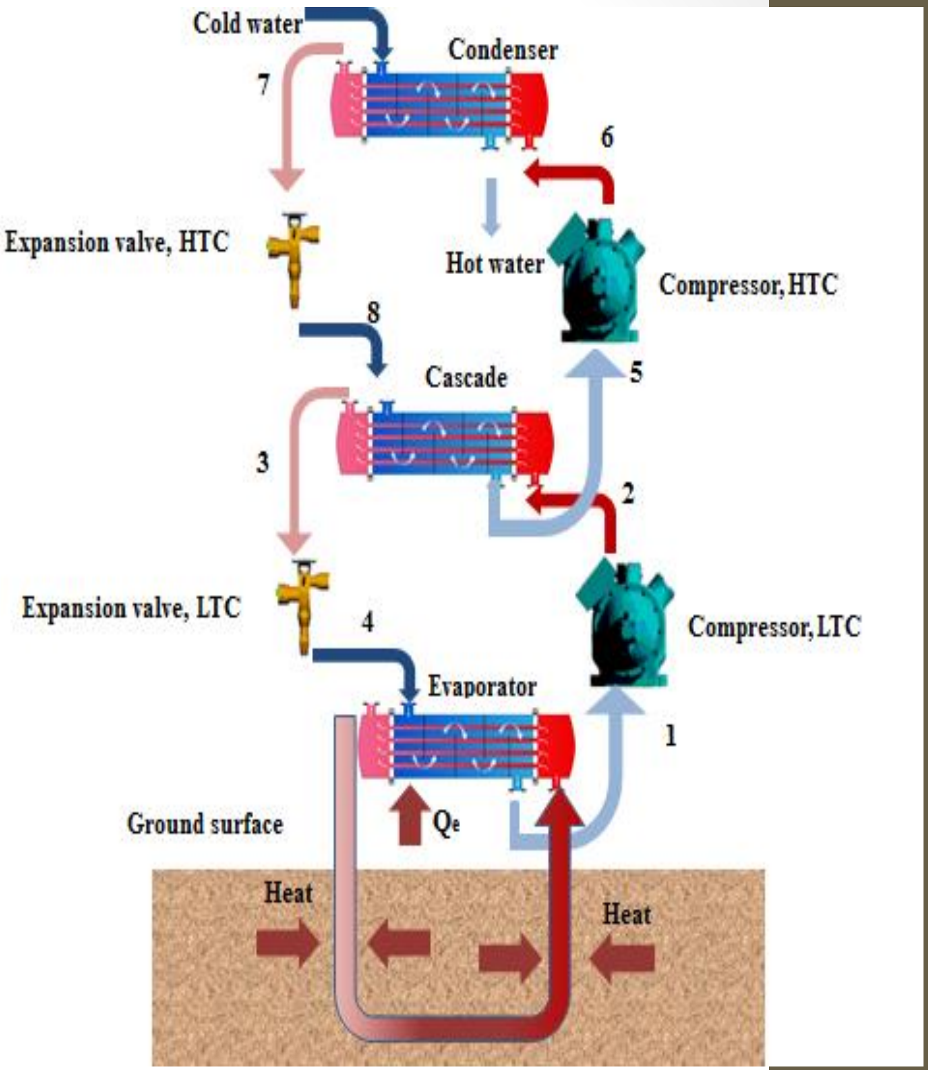
| References | Refrigerant | Heat source | Optimization |
|---|---|---|---|
| **Jin et al. [13]** | $CO_2$ | Ground | Energy |
| **Sarbu et al. [14]** | CFC | Ground | Energy, economic and environmental |
| **Sivasakthive et al. [15]** | R22 | Ground | Energy |
| **Onder Ozgener et al. [16]** | R22 | Ground-solar | Energy, Exergy efficiency |
| **H. Weeratunge et al [17]** | Water | Ground-solar | Energy, economic |

# Innovation

- Using cascade refrigeration system
- Exergy-economic analysis
  1. COP
  2. Exergy efficiency
  3. Total cost rate
- Investigation 5 cities in Russia
  1. Climate condition
  2. Soil thermal properties
  3. Soil temperature

1- Optimal value of evaporator temperature
2-Optimal value of condensation cascade temperature
3- Selection the best value of COP, exergy and cost

- Refrigerant receives heat from ground at point 4 and leaves evaporator in the form of saturated vapor.
- Saturated vapor becomes superheated at point 2 through a compression process that is assumed adiabatic.
- Heat is transferred from high pressure refrigerant in LTC to HTC refrigerant.
- In 5-6 process, refrigerant becomes superheated by means of HTC compressor.
- The high pressure and temperature refrigerant in condenser rejects heat to the wate for domestic application and then HTC refrigerant is converted to the saturated liquid state at condenser outlet (point 7).




| Cycle | Refrigerant | Critical temperature (°C) | Boiling point (°C) | ODP | GWP |
|---|---|---|---|---|---|
| HTC | R161 | 102.2 | -37.1 | 0 | 12 |
| LTC | R41 | 44.1 | -78.1 | 0 | 97 |

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# Geographical region


| City | Average winter temperature (°C) | Humidity (%) |
|---|---|---|
| **Yekaterinburg** | -17 | 80 |
| **Nizhny Novgorod** | -15 | 83 |
| **Moscow** | -13 | 82 |
| **Saint Petersburg** | -10 | 83 |
| **Sochi** | 5 | 75 |

# Energy-exergy analysis

## Evaporator:

$$\dot{m}_4 = \dot{m}_1 = \dot{m}_{LTC}$$

$$\dot{Q}_{eva} = \dot{m}_{LTC}(h_4 - h_1)$$

$$\dot{E}x_{eva} = \dot{m}_{LTC}(ex_4 - ex_1) + \dot{Q}_{eva}(1 - \frac{T_{air}}{T_{eva}})$$

## LTC and HTC Compressor:

$$\dot{m}_2 = \dot{m}_1 = \dot{m}_{LTC}, \dot{m}_5 = \dot{m}_6 = \dot{m}_{HTC}$$

$$\dot{W}_{com,LTC} = \dot{m}_{LTC}(h_2 - h_1), \dot{W}_{com,HTC} = \dot{m}_{HTC}(h_6 - h_5)$$

$$\dot{E}x_{com,LTC} = \dot{m}_{LTC}(ex_1 - ex_2) + \dot{W}_{com,LTC}, \dot{E}x_{com,HTC} = \dot{m}_{LTC}(ex_5 - ex_6) + \dot{W}_{com,HTC}$$

$$\dot{W}_{tot} = \dot{W}_{com,HTC} + \dot{W}_{com,LTC}$$

$$\eta_{com,LTC} = \frac{h_{2s} - h_1}{h_2 - h_1}, \eta_{com,HTC} = \frac{h_{6s} - h_5}{h_6 - h_5}$$

## Condenser:

$$\dot{m}_6 = \dot{m}_7 = \dot{m}_{HTC}$$

$$\dot{Q}_{con} = \dot{m}_{HTC}(h_6 - h_7)$$

$$\dot{E}x_{con} = \dot{m}_{HTC}(ex_6 - ex_7) + \dot{Q}_{con}\left(1 - \frac{T_{air}}{T_{con}}\right)$$

# Energy-exergy analysis

## LTC and HTC Expansion Valve:

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_{HTC}, \dot{m}_3 = \dot{m}_4 = \dot{m}_{LTC}$$

$$h_7 = h_8, h_3 = h_4$$

$$\dot{E}x_{exp,HTC} = \dot{m}_{LTC}(ex_7 - ex_8), \dot{E}x_{exp,LTC} = \dot{m}_{LTC}(ex_3 - ex_4)$$

## Cascade Heat Exchanger:

$$\dot{m}_2 + \dot{m}_8 = \dot{m}_3 + \dot{m}_5$$

$$\dot{Q}_{cas} = \dot{m}_{LTC}(h_3 - h_2)$$

$$\dot{E}x_{cas} = \dot{m}_{LTC}(ex_2 - ex_3) + \dot{m}_{HTC}(ex_8 - ex_5)$$

$$COP = \frac{\dot{Q}_{eva}}{\dot{W}_{tot}}$$

$$\dot{e}x = \dot{m}[(h - h_{air}) - T_{air}(s - s_{air})]$$

## Total exergy destruction

$$\dot{E}x_{tot} = \dot{E}x_{eva} + \dot{E}x_{con} + \dot{E}x_{com,HTC} + \dot{E}x_{com,LTC} + \dot{E}x_{cas} + \dot{E}x_{exp,HTC} + \dot{E}x_{exp,LTC}$$

$$\eta_{ex} = \frac{\dot{W}_{tot} - \dot{E}x_{tot}}{\dot{W}_{tot}}$$

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# Economic-environmental analysis

$$\dot{Z}_{tot} = \sum \dot{Z}_{cm} + \dot{Z}_{op} + \dot{Z}_{em}$$

- $\dot{Z}_{tot}$: Total cost rate
- $\sum \dot{Z}_{cm}$: Capital and maintenance cost
- $\dot{Z}_{op}$: Operational cost
- $\dot{Z}_{em}$: Cost rate due to $CO_2$ emission

$$\dot{Z}_{op} = Nh \times \dot{W}_{tot} \times \beta_{el}$$

- $Nh$: Operational hours
- $\beta_{el}$: Electricity price

$$\dot{Z}_{em} = \mu_{CO_2} \times \dot{W}_{tot} \times \beta_{CO_2}$$

- $\mu_{CO_2}$: Emission factor
- $\beta_{CO_2}$: $CO_2$ penalty price

Installation cost of 61 $ per meter depth from the ground surface is added to the total cost of GSHP system.

$$CRF = \frac{ir(ir+1)^n}{(1+ir)^n - 1}$$

Interest rate of *ir*=14% and life time of n=15 year

Ø=1.06

| Component | Capital and maintenance cost rate |
|---|---|
| Evaporator | $\emptyset \times CRF \times 1397 \times A_{eva}^{0.89}$ |
| Compressor, LTC | $\emptyset \times CRF \times 10167.5 \times \dot{W}_{com,LTC}^{0.46}$ |
| Compressor, HTC | $\emptyset \times CRF \times 9624.2 \times \dot{W}_{com,HTC}^{0.46}$ |
| Condenser | $\emptyset \times CRF \times 383.5 \times A_{con}^{0.65}$ |
| Expansion Valve, HTC | $\emptyset \times CRF \times 114.5 \times \dot{m}_{HTC}$ |
| Expansion Valve, LTC | $\emptyset \times CRF \times 114.5 \times \dot{m}_{LTC}$ |
| Cascade heat exchanger | $\emptyset \times CRF \times 383.5 \times A_{cas}^{0.65}$ |

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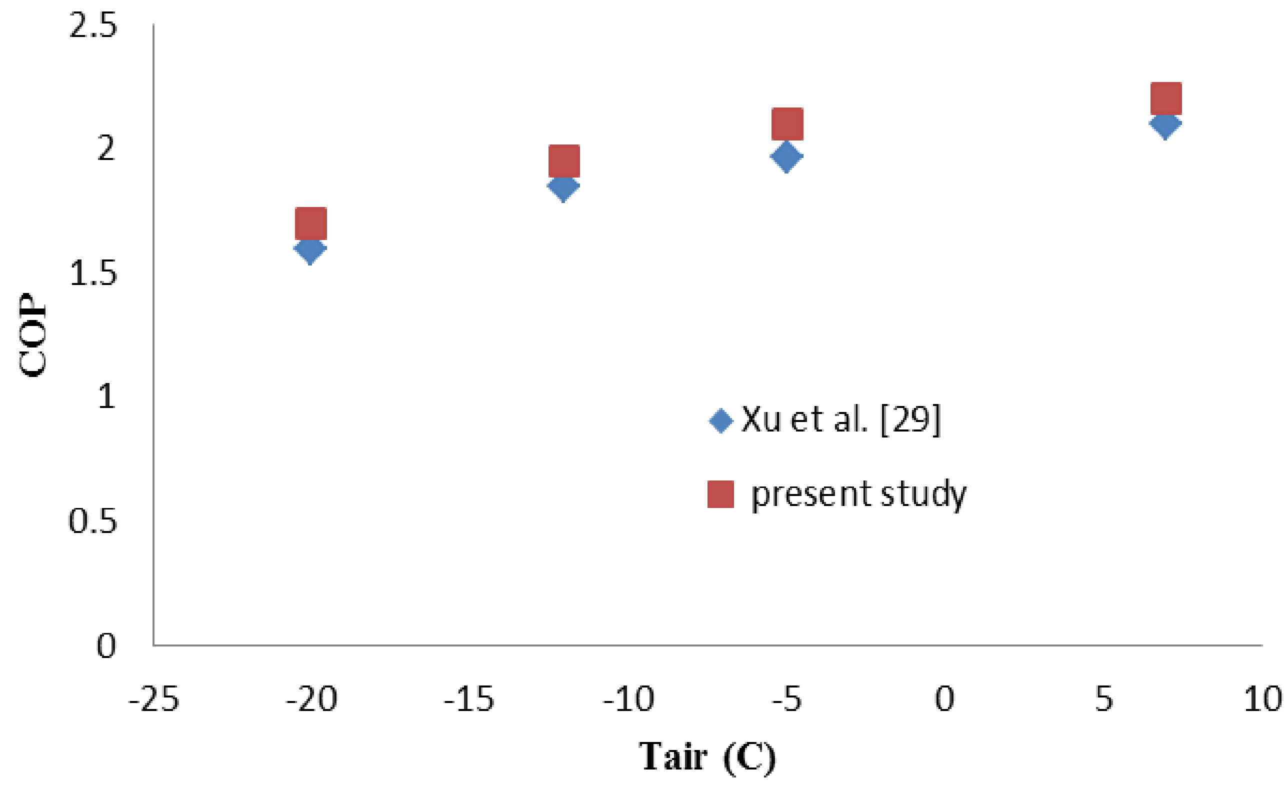
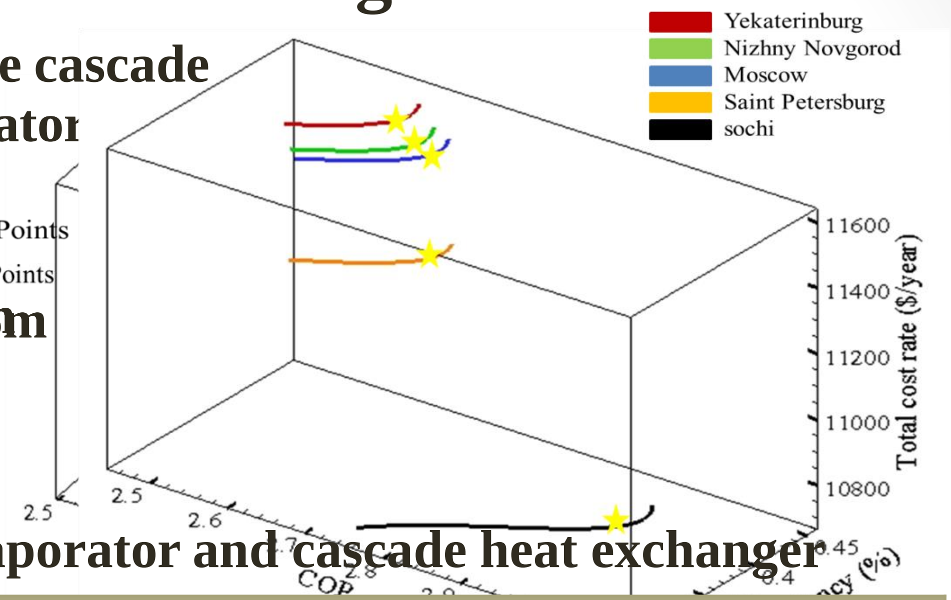
- Cascade air source heat pump
- Increasing temperature of water as high as 75 ºC using R404A and R134a in LTC and HTC
- COP variations in terms of ambient temperature



# Optimal temperature of evaporator and condensation cascade heat exchanger

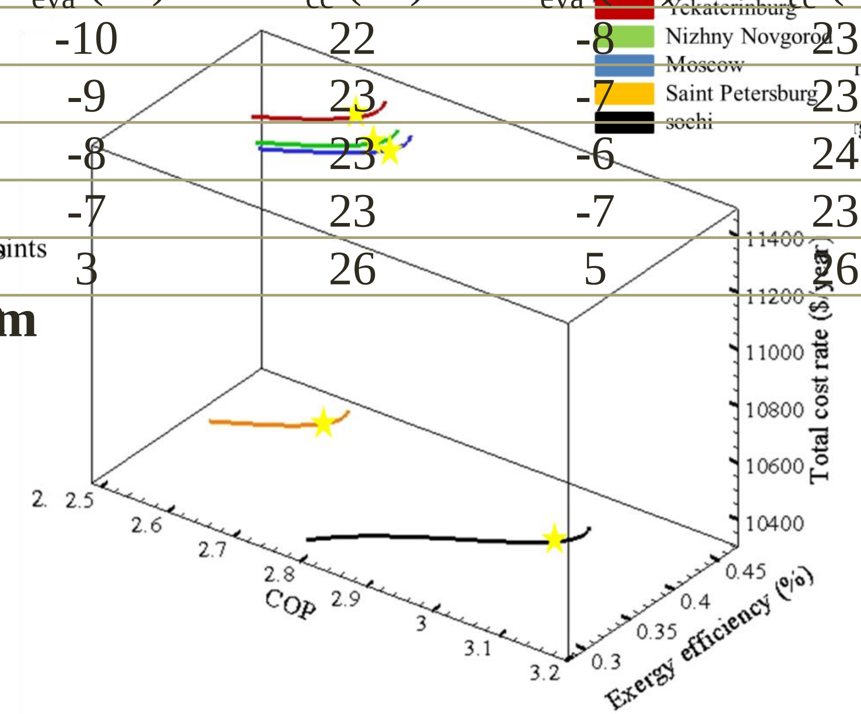
Optimal condensation temperature cascade

Optimal temperature of evaporator




**Optimal temperature of evaporator and cascade heat exchanger**

| | City | 0.5 (m) | | 1 (m) | |
|---|---|---|---|---|---|
| | | $T_{eva}$ (°C) | $T_{cc}$ (°C) | $T_{eva}$ (°C) | $T_{cc}$ (°C) |
| 1 | Yekaterinburg | -10 | 22 | -8 | 23 |
| 2 | Nizhny Novgorod | -9 | 23 | -7 | 23 |
| 3 | Moscow | -8 | 23 | -6 | 24 |
| 4 | Saint Petersburg | -7 | 23 | -7 | 23 |
| 5 | Sochi | 3 | 26 | 5 | 26 |



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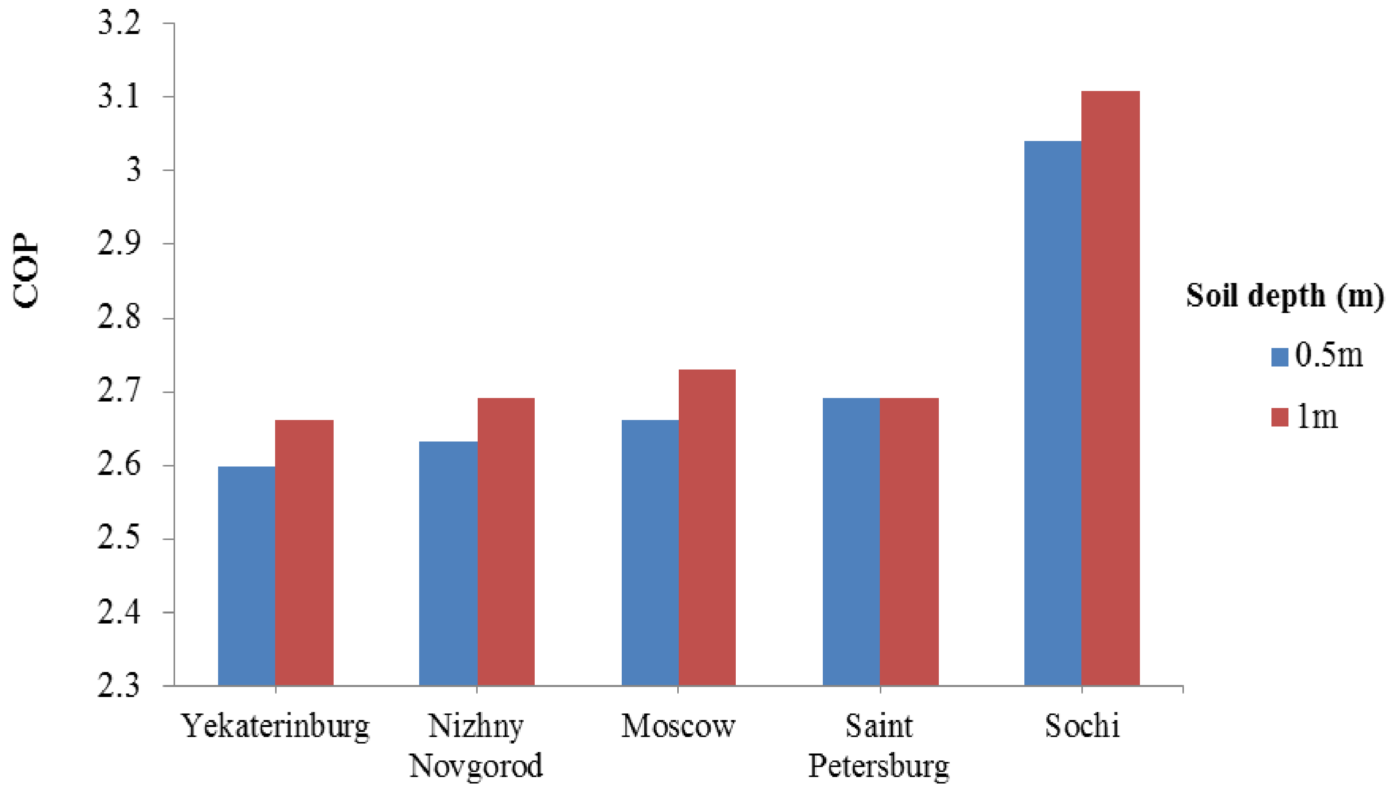
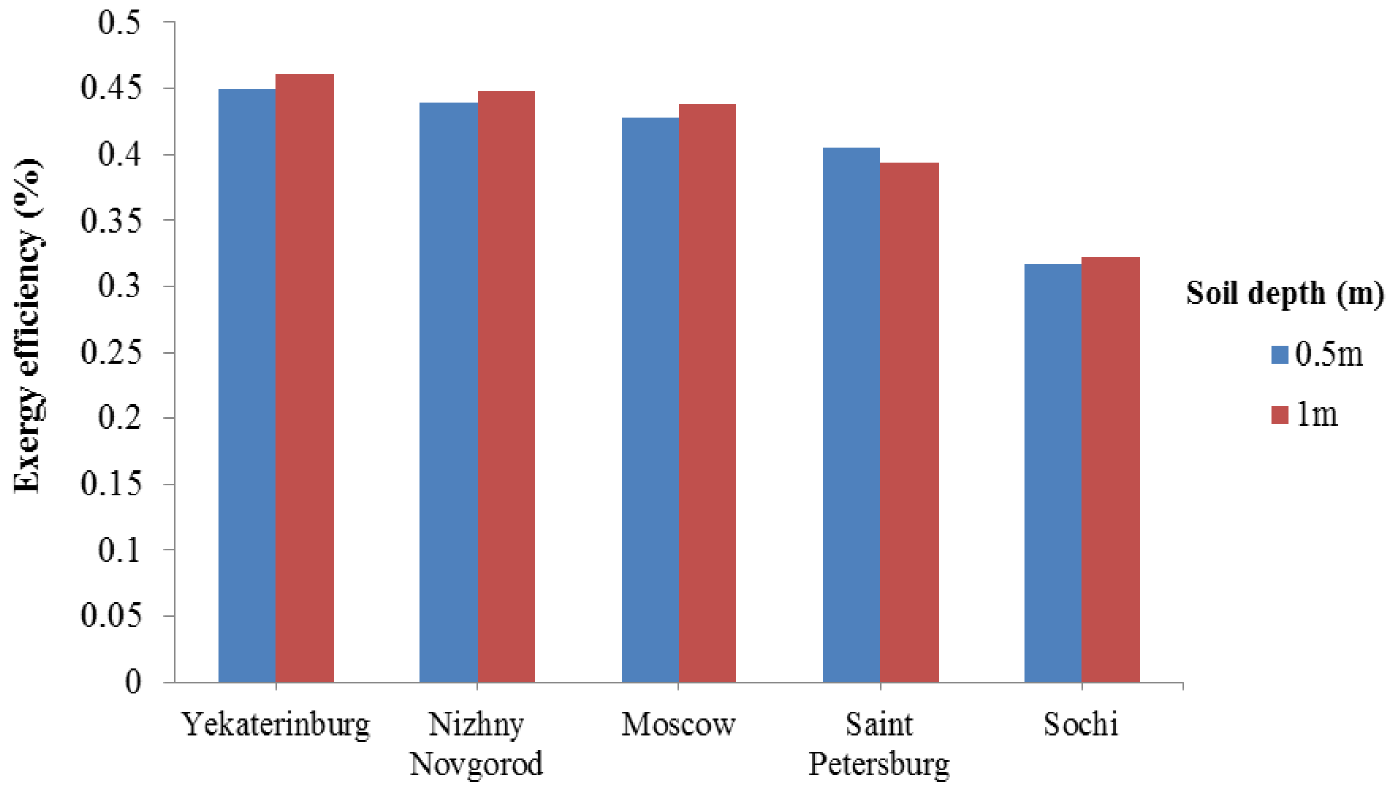
# Comparison of optimization results for COP



# Comparison of optimization results for exergy efficiency



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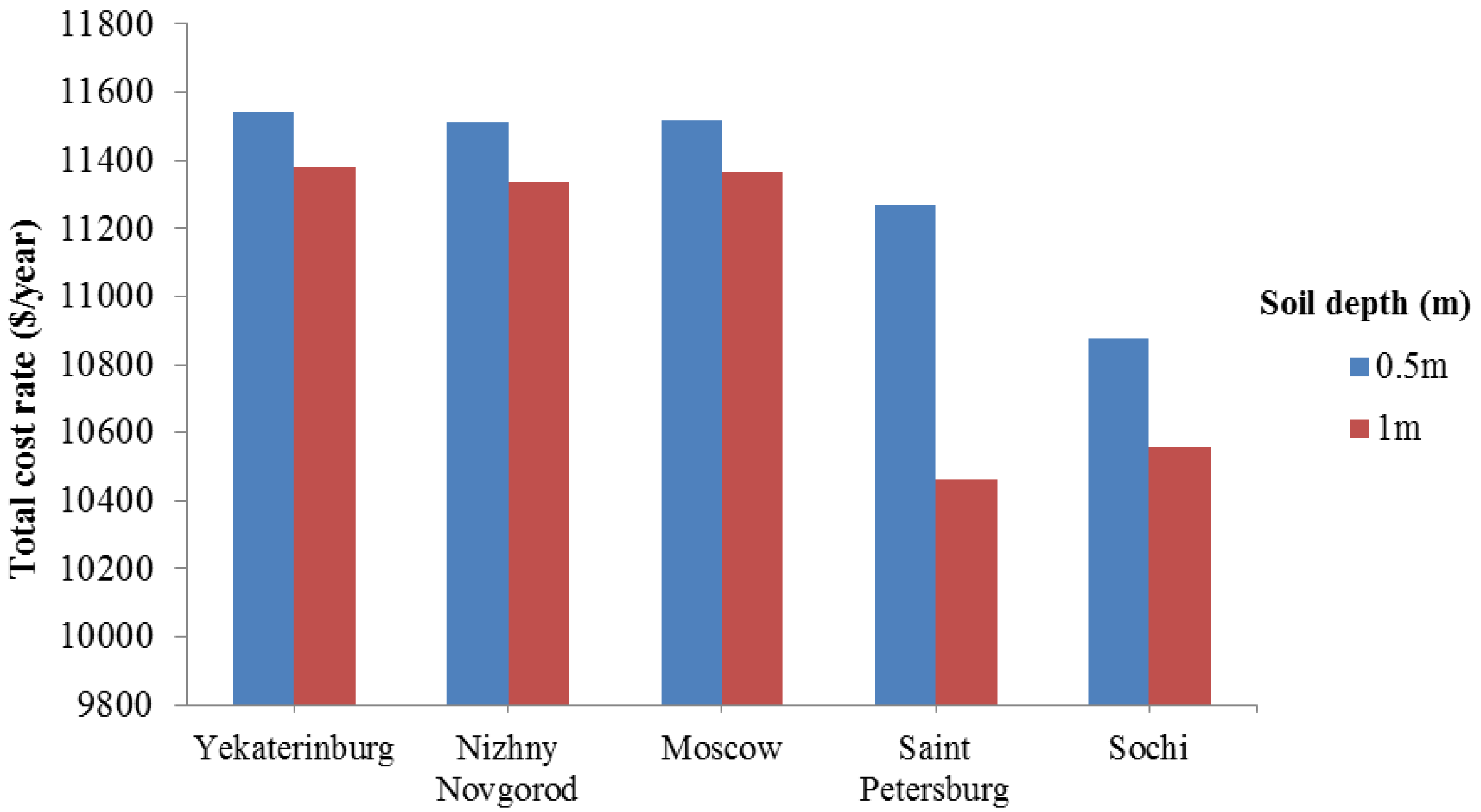
# Comparison of optimization results for total cost rate



- In optimal point, maximum COP, maximum exergy efficiency and minimum total cost rate are considered to calculate optimal value of evaporator temperature and condensation temperature of cascade heat exchanger
- Increasing evaporator temperature and condensation temperature of cascade heat exchanger leads to increase COP and exergy efficiency and decrease the total cost rate.
- Increasing soil depth leads to increase evaporator temperature and total cost rate due to the increasing cost drilling for higher depth.
- Maximum COP among the 5 cities in Russia is equal to 3.11 for Sochi while maximum exergy efficiency is equal to 0.46 for Yekaterinburg.
- The most cost-effective total cost rate is obtained for Saint Petersburg which is 10462 ($/year).

Thank you