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Accurate Modeling Of Heat Pump Cycles With Steam Injection In Off-design Regimes Utilizing An Integrated Approach

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ABSTRACT

The calculation of heat pump cycles is usually made with a number of assumptions. Common mistakes in this early stage are to improperly account for the hydraulic resistance of the heat exchange equipment, the use of constant compressor efficiency (independent from operating mode), and etc. All of these assumptions critically affect the change in parameters, important to the consumer.

The authors of this article consider an integrated method for the heat pump cycle designing in various operating modes. The proposed approach includes the preliminary design of the compressor unit and takes into account the influence of the heat exchangers' hydraulic resistance. The integration process of the calculated systems enables accurate evaluation of the parameters after the 1st stage of the compressor, taking into account the steam injection. The authors then compare the proposed approach against a cycle analysis which doesn't consider the additional influence of real equipment parameters.

Keywords: heat pump, cycle, heat exchanger, hydraulic resistance, steam injection, design, analysis.

1. INTRODUCTION

In the twenty-first century, we are seeing the rise of two trends which are causing a change of attitude:

- the striking increase in energy prices (Figure 1) ; and
- the consequences of climate change (Figure 2).

These problems draw attention to the need for greater use of renewable energy sources such as heat pumps [1].

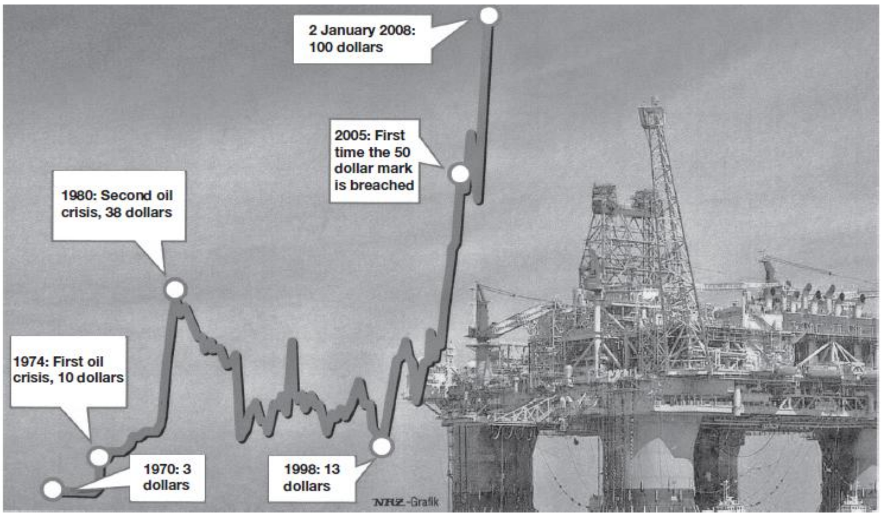

Figure 1: Increasing crude oil prices [1]

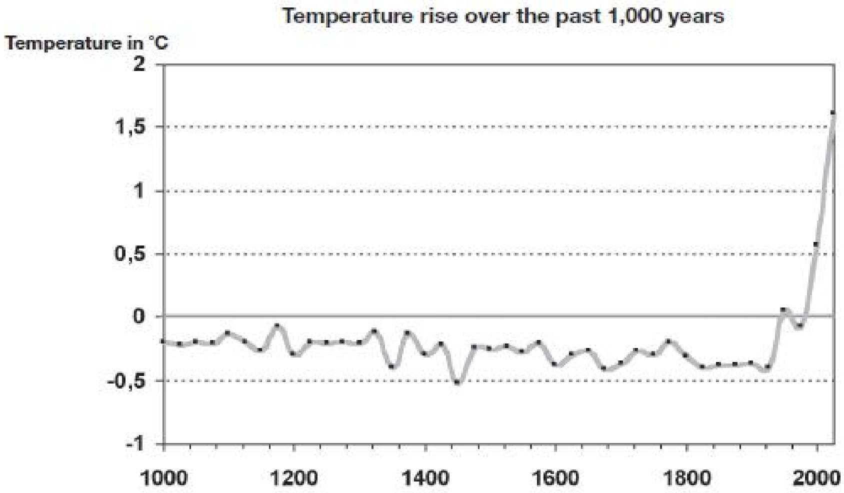

Figure 2: Extreme temperature rises at the turn of the century [1]

Heat pumps are not a state-of-the-art technology. Air is the most common heat source, while larger heat pumps are making use of geothermal and hydrothermal sources, especially in recent years. These heat sources can be used for industrial installations such as for district heating. Heat pumps can be classified according to several design or function characteristics. They can be divided into two classes - vapor compression and absorption heat pumps. The former follows an inverse thermodynamic cycle and uses a compressor driven by an electric motor or other kind of engine. The second type, absorption heat pumps, do not have a mechanical compressor. They use a mixture of two fluids with different vapor pressures. The more volatile fluid evaporates, and then recombines with the less volatile fluids [8]. Absorption heat pumps typically use mixtures of water and lithium-bromide or water and ammonia. The article considers the first type of heat pumps as they are the most common in heating system application.

With the development of greater and more complex technology comes the challenge of producing high-quality designs considering inverse thermodynamics cycle. Very often, the early stage of preliminary design in a heat pump comes down calculations generated from the thermodynamic cycle performance. The system components are then chosen based on these calculations. However, this approach does not take into account the influence of equipment on the entire cycle. The combined design and analysis process is a difficult task. An approximate definition of the compressor efficiency, the lack he hydraulic resistance of evaporator, and condenser leads to significant discrepancies and complicates the overall heat pump design process. Nowadays, it is necessary to apply an integrated approach to solving engineering design challenges, which enables engineers to evaluate the effects of each component and make an accurate analysis of how these components interact. Evaluation of the cycle must include accurate preliminary design of the compressor units, accurate design of heat exchangers accounting for heat transfer, and evaluate how teach component operates throughout the cycle. The system design approach applied to heat pump design is one potential method to reduce development duration and associated costs. Such a method along with performance analysis is proposed by the authors of this article and described as follows.

2. HEAT PUMP PRELIMINARY DESIGN

2.1. Boundary conditions

The first step involves designing the heat pump cycle for a district heating system. The installation will work as a source of heat for the surrounding buildings and the temperature of the heated water will depend on the outdoor temperature. The heat pump is located in its own space, which is separated from any residential buildings. The placement for the ground evaporator is restricted by its performance capabilities. Thus, the design process for the heat exchangers has to take into account this restriction, and its influences on the heat transfer area while ensuring proper evaporation and condensation. This means that it is crucial to look at the cycle as a whole. Preliminary design is performed based on boundary conditions established in the cycle assume maximum possible . The maximum working regime assumes a temperature range of 363.15/343.15 K, where the maximum hot water temperature equals 363.15 K and the maximum returned (cold) water temperature equals 343.15 K . The assessment of the thermodynamic cycle was done with the initial data given in Table 1.

Table 1. Technical Characteristics of Designed Heat Pump

Designed Heat power, kW	≈1000
Refrigerant	Ammonia
Maximum hot water temperature, K	363.15
Maximum cold water temperature, K	343.15
Type of heat exchanges - evaporator and condenser	Shell-and-tube with finned tubes
Type of compressors	Centrifugal
Application area	District heating systems

The selection of low-grade heat is made from the soil using an intermediate coolant - 55% propylene glycol. The design of the system was performed for outdoor temperatures ranging between 281.15 K to 251.15 K to keep the indoor temperature equal 293.15 K

2.2. Refrigerant selection

It should be noted that six heat pump cycles with different refrigerants have been reviewed and compared with each other in order to choose the most optimal working fluid. The cycles were performed with identical parameters in order to ensure the maximum values of the designed cycle. When choosing a refrigerant for the system, the following factors were taken into account:

- Thermodynamic cycle efficiency;
- System dimensions;
- Hazards presented by the refrigerant (flammability and toxicity);
- Environmental friendliness.

The analysis of the different thermodynamics cycles was critical in obtaining the needed parameters, as well as estimating the physical properties and behavior of each fluid.

Table 2. Thermodynamic Refrigerant Values and Physical Properties

Fluid Type	R134A	R152A	R600a	R717	R1234ze
Mass Flow Rate, kg/s	8.018	4.077	4.246	0.7011	9.257
Heat Capacity, $kJ/kg \cdot K$	0.8544	1.0414	1.557	2.5419	0.8455
Enthalpy of Vaporization, kJ/kg	205.97	316.98	363.54	1296.7	190.23
Maximum Pressure, bar	31.64	28.07	17.77	47.68	25.61
Maximum Temperature, K	383.15	403.15	371.15	547.15	375.15
COP in Current Design Conditions	1.656	1.962	1.927	1.986	1.821
Global Warming Potential (GWP)	Medium / 1430	Low / 124	Low / 3	No / 0	Low / 7
Ozone Depleting Potential (ODP)	None / 0	None / 0	None / 0	None / 0	None / 0
ASHRAE Safety Group	A1	A2	A3	B2L	A2L
ASHRAE Flammability	No	Yes	Yes (Highly flammable)	Yes (Low)	Yes (Low)
ASHRAE Toxicity	No	No	No	Yes	No

(Table values are estimated at T=263.15 K, x=1)

A comparison shows that for operational and technical reasons, refrigerant R717 (Ammonia NH_3) has a clear advantage over the others. Ammonia possesses higher thermodynamics efficiencies, low flammability,

and zero global warming potential. However, the disadvantages of this refrigerant are its toxicity and its high pressure and temperature at the outlet of a compressor. The large overheating relative to the saturation line at the outlet of the compressor is due to the characteristics of the saturation line of this fluid and makes the operation of the condenser less optimal. This does not, however prevent it from having a high COP. In terms of lower pressure after the compressor, the best option is R600a, which has a high value of COP and is nontoxic, but has a high flammability factor. The high heat capacity and enthalpy of vaporization values of R717, however, allows for a decreasing mass flow rate value of refrigerant in the system (in 6 times compare with R600a). This provides the possibility to decrease the metal consumption and installation dimensions. In addition, ammonia has fewer costs compared with other considered fluids. Therefore, the lower mass flow rate value of R717 ensures a lower cost of running the entire heat pump.
According to the ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) this fluid is completely environmentally friendly (Index GWP - 0, ODP - 0), which satisfies the design criteria. However it should be noted that the installation must meet the needs of PGS-13.

2.3. Thermodynamics Cycle Scheme Simulation

The simulation of the heat pumps cycle with all components has been performed in AxCYCLE™ - 0D software for designing and analyzing the cycle for a variety of power systems and performance of new or existing systems at “off-design” operating conditions. The thermodynamic cycle includes main components found in heat pumps (Figure 3).

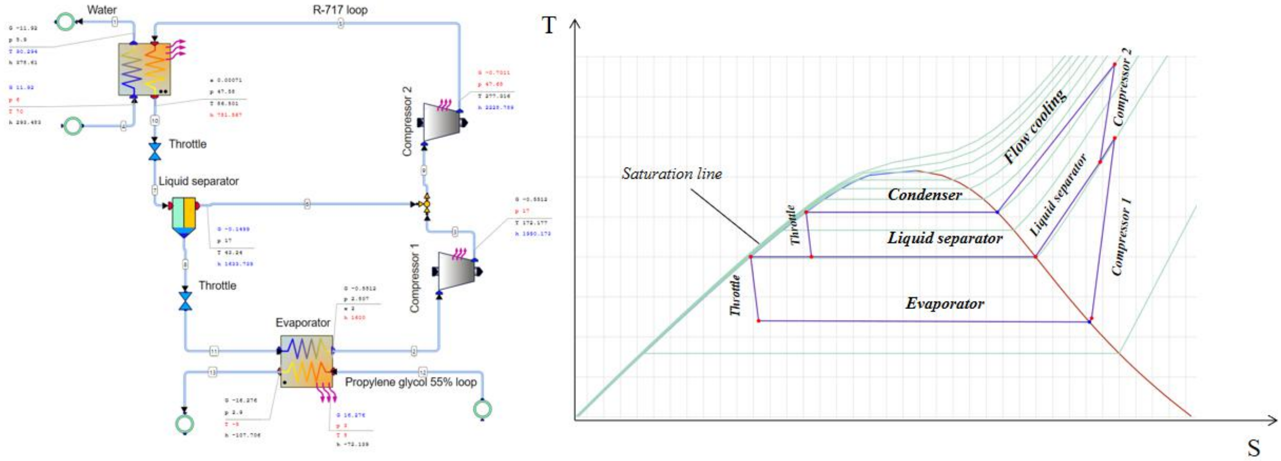

Figure 3: Heat pump scheme and thermodynamics process in AxCYCLE™

In the evaporator, refrigerant undergoes phase transition due to heat transfer from the propylene glycol 55%. In doing so, the refrigerant absorbs a lot of energy, which is later used for heating. The gas R717 inflow to the first compressor where its pressure and temperature are increased. Between the second centrifugal compressor, the refrigerant temperature is decreased and the mass flow rate of the gas is increased due to the additional gas injection from the liquid separator. After the second compressor, the gas refrigerant reaches the necessary pressure to ensure the water temperature to the consumer. In the condenser, the refrigerant becomes a liquid due to the gas phase transition. The cycle includes two thermal expansion valve components. The first valve is established after the condenser where the throttling process takes place with constant enthalpy. The liquid separator after the valve is used to divide the gas phase of the refrigerant from the liquid phase, which occurs due to the throttling process in the valves. The second valve reduces the refrigerant’s physical parameters before the evaporating process in the heat exchanger. The liquid refrigerant is pressed under high pressure. After passing through the valve, the refrigerant expands. Both the pressure and temperature of the refrigerant fall significantly. A way to save energy in the case of a large temperature difference between thermal sources, i.e., large pressure ratio, is by injecting vapor into the compressor at an intermediate pressure [8]. Thereby, the complication of the heat pump cycle by adding a separator to the overall thermodynamic cycle allows achieving a greater value of the efficiency due to decreasing refrigerant parameters before the second compressor by the cold gas injection.

2.4. Heat Exchangers Simulation

The preliminary design and layout definition of the heat exchangers has been performed using the methodology described in [7]. Typically, the heat exchangers are modeled using copper given its large thermal conductivity, but the evaporator and condenser have been designed using stainless steel since copper cannot be used with ammonia. With this in mind, the heat exchange and corresponding heat transfer coefficients were obtained for a stainless steel material. The heat exchanger's design places the refrigerant in the shell side and water\propylene glycol 55% insight tubes. As mentioned above, according to the technical task, the evaporator heat transfer area is limited by the location of the evaporator. The condenser heat exchanger includes two parts: the first part corresponds to the refrigerant precooling zone and the second part is where the full heat transfer of ammonia occurs in a liquid state. As a result, the calculations obtain the main heat exchanger's geometric parameters and layouts, such as the number of fluid passes, tubes length, and diameters. The fluids' boundary conditions on each side have been selected from thermodynamics cycle performance. Obtained geometric parameters enables its use for tubes and shell side hydraulic resistance simulation; pressure difference between inlet flow and outlet flow, estimation of the thermal parameters such as wall temperatures, heat flows, and heat transfer coefficients. In order to simulate the flow in the evaporator and condenser, the AxSTREAM NET™ software package for thermal-fluid system modeling and analysis was used. It provides a method to model fluid path and solid structures as a set of 1D elements, which can be connected to each other to form a thermal-fluid network. For each fluid path section (represented by appropriate elements), fluid flow parameters for the inlet and outlet cross-sections (such as velocity, pressure, density, temperature, mass flow rate, etc.) were calculated. The scheme of the evaporator is shown in Figure 4 and the condenser model in Figure 5.

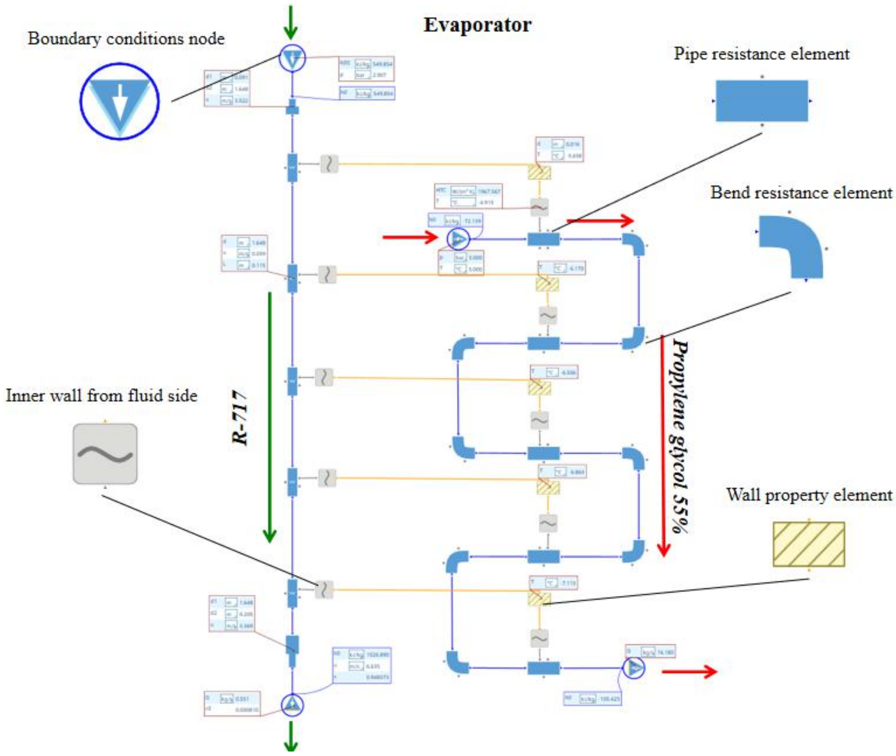

Figure 4: Evaporator Simulation in AxSTREAM NET™

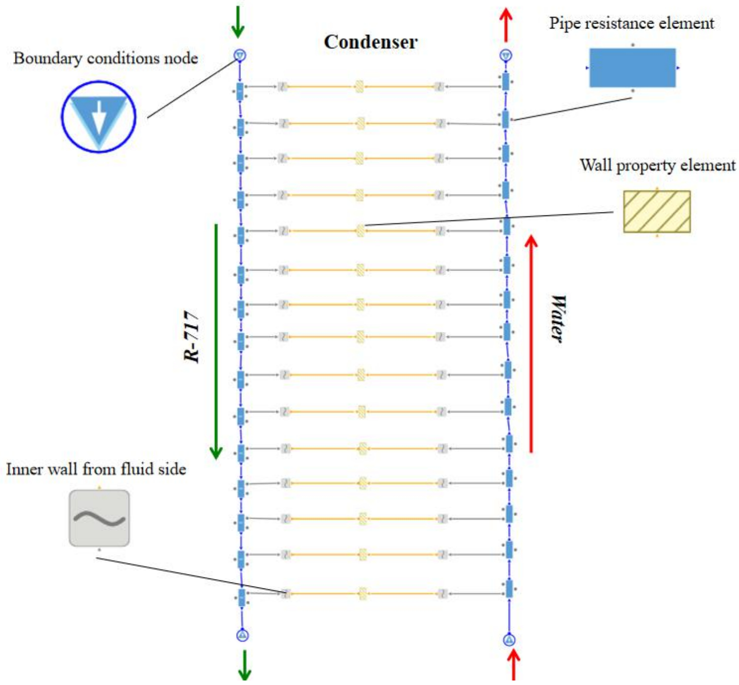

Figure 5: Condenser Model in AxSTREAM NET™

The previously performed preliminary design to get boundary conditions from the thermodynamics cycle and obtain main geometric parameters allows the authors to estimate the pressure drop from fluids on each side (both due to tubes and the number of moves resistance and flow around a tube bundle in the annuls. The values from the heat exchangers design process are presented (Table 3):

Table 3. Parameters of Designed Heat Exchangers

Evaporator	
Propylene glycol 55% side	
Inlet total pressure, bar	3
Inlet total temperature, *K*	278.15
Outlet mass flow rate, kg/s	16.18
Tubes length, m	5.285
Tubes diameter, m	0.015
Number of tubes	46
Number of passes	5
Pressure drop, bar	2.062
R717 side	
Inlet total pressure, bar	2.907
Inlet total temperature, *K*	265.15
Outlet mass flow rate, kg/s	0.551
Pressure drop, bar	0.0008
Condenser	
Water side	
Inlet total pressure, bar	6
Inlet total temperature, *K*	343.15
Outlet mass flow rate, kg/s	11.918
Tubes length, m	6
Tubes diameter, m	0.015
Number of tubes	48
Number of passes	1
Pressure drop, bar	0.092

R717 side	
Inlet total pressure, bar	47.68
Inlet total temperature, K	543.073
Outlet mass flow rate, kg/s	0.701
Pressure drop, bar	0.0043

2.5. Compressors Preliminary Design

The parameters of the centrifugal compressors were acquired assuming work with different boundary conditions. This occurs due to additional gas refrigerant injection before the second stage in order to cool the refrigerant. These compressors will be presumably located on different independent shafts with different angular velocities. It should be noted that the preliminary design of the compressors does not consider additional equipment choices, such as the estimation of electrical motors, gearboxes, and multipliers. Despite this, as it will be shown later, the completed and integrated approach provides the possibility to quickly update any block in the detailed design. The preliminary design and further analysis of the compressors have been performed in AxSTREAM® (Figure 6), a multidisciplinary design, analysis and optimization software platform, that provides a fully integrated and streamlined solution, encompassing the complete turbomachinery design process. The boundary conditions for compressors design are shown in Table 4.

Table 4. Boundary Conditions for Compressors Design

Compressor 1	
Parameter	Minimum
Inlet total pressure, bar	2.807
Inlet total temperature, K	265.15
Outlet total pressure, bar	17.0
Mass flow rate, kg/s	0.551
Compressor 2	
Parameter	Minimum
Inlet total pressure, bar	17.0
Inlet total temperature, K	415.495
Outlet total pressure, bar	47.68
Mass flow rate, kg/s	0.7011

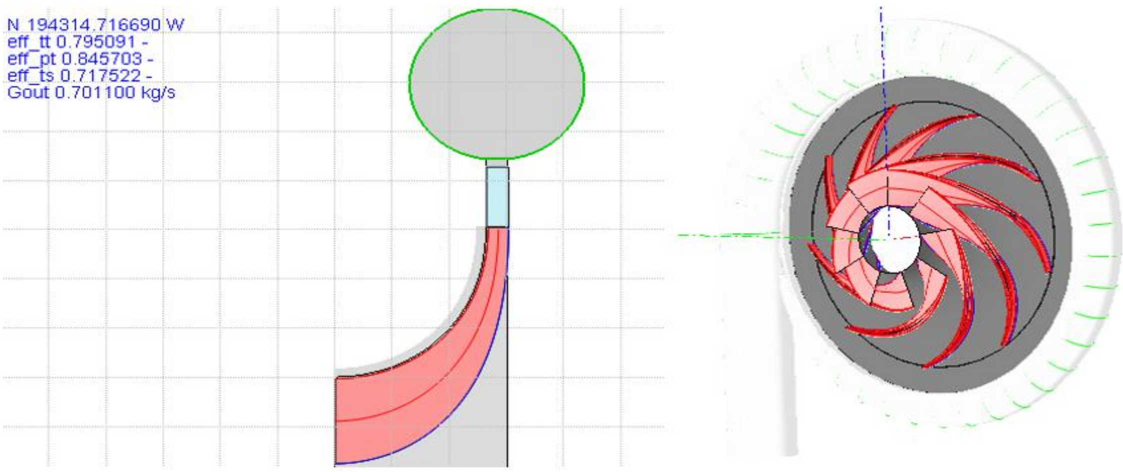

Figure 6: Centrifugal compressor stage

The required values for future integrated design and analysis in off-design regimes for each compressor is shown in Table 5.

Table 5. Output Values

Compressor 1	
Isentropic efficiency	0.7895
Mechanical and electrical efficiency	0.90
Shaft rotational speed	140000.0
Equivalent diffuser factor	1.69
Compressor 2	
Isentropic efficiency	0.7951
Mechanical and electrical efficiency	0.90
Shaft rotational speed	198000.0
Equivalent diffuser factor	1.7

Real efficiency of compressors can be estimated as:

$$eff = \Delta h_{is} / \Delta h \qquad \text{Eq. (1)}$$

2.6. Integrating Process for Heat Pump Preliminary Design

The engineering process for the preliminary system needs to integrate a variety of tools for design/ thermodynamics simulation of each specific component or subsystem of the heat pump in a single iterative process. Basically, thermodynamic calculations are performed to obtain the operational characteristics of real compressors and estimate the entire heat load on all heat exchangers. After the evaluation, obtained values can be used in order to select already existed types of equipment. In a new design process, obtaining preliminary boundary conditions, the process of thermodynamic cycle estimation assumes users define first-order variables for the heat exchangers, hydraulic resistance, and compressor efficiencies. Estimations create the possibility of a mistake. Therefore after each separate cycle process has been performed, i.e. the thermodynamics cycle, heat exchanger modeling, and compressor design, an accurate integrated method should be applied which can provide a joint process of designing. Thus, the correct next step towards the final product can be proposed. Allowing designers to connect all of the individual modules into a singular design provides for accurate results in the shortest possible time. This integration can be performed using AxSTREAM ION™ - a software tool, which automates the design process, optimization tasks, and helps to structure and to accelerate the overall design process. The execution process is shown in Figure 7.
Each presented block describes modules of execution. This includes thermodynamics cycle, thermal-hydraulic networks of the heat exchangers, compressors design and analysis modules, blocks of parameters comparison with assigned accuracy between each block, and design method blocks. The boundary conditions binding from upstream to downstream blocks. The main stages of a heat pump design process are as follows:

- Initial data block;
- Thermodynamics cycle with main heat pump component calculations;
- Initial data for heat exchangers design;
- Evaporator thermal-hydraulic estimation;
- Evaporator heat load comparison block;
- Evaporator tube length reassignment block;
- Condenser thermal-hydraulic estimation;
- Condenser heat load comparison block;
- Condenser tube length reassignment block;
- Compressor first stage preliminary design;
- Compressor second stage preliminary design;
- Parameters comparison blocks.

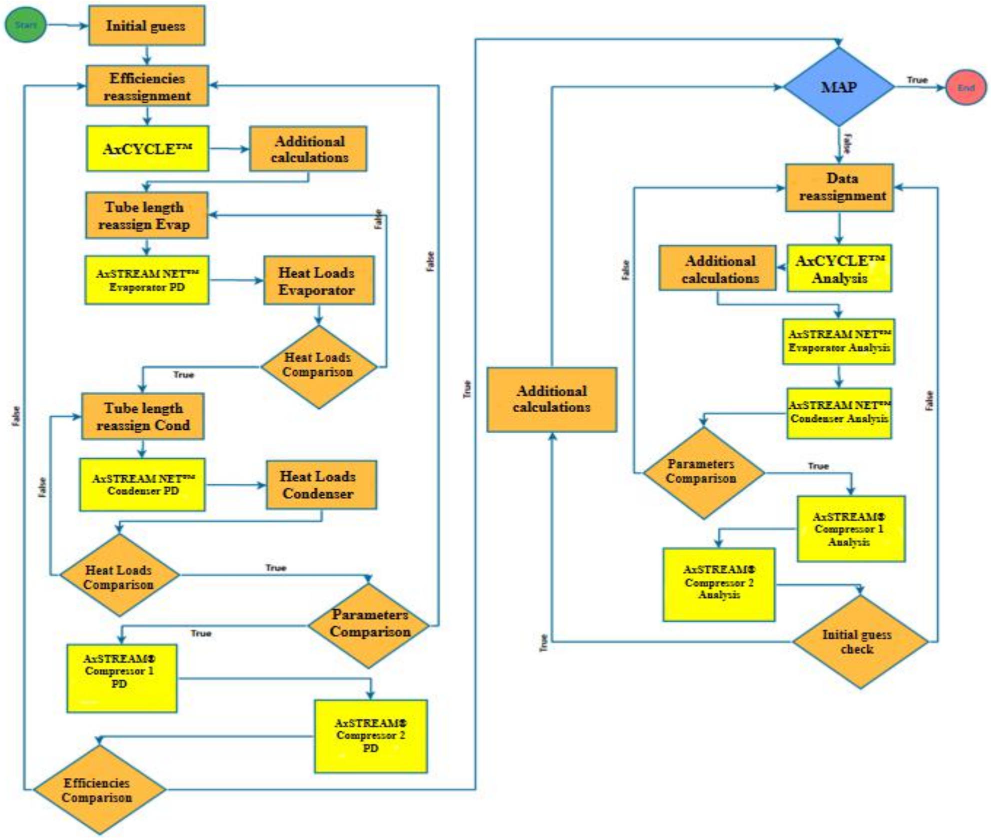

Figure 7: Integrated Execution Process of Heat Pump Design and Analysis

2.6.1. Initial Data Block

In order to start the iterative process of the design workflow, the initial block binds the first-order data for cycle calculation. The compressors' first order efficiencies/pressure drops of each heat exchanger are passed to the thermodynamic cycle block.

2.6.2. Thermodynamic Cycle Block

Initial data from the upstream block allows the cycle calculation in the first order to be performed. Cycle execution provides parameters to specific modules in order to accurately design the main components of the cycle. After each iteration, all specified values are returned again, and the process of design is repeated until satisfactory.

2.6.3. Heat Exchanger Geometry Parameters Initial Block

In order to start the preliminary design process of the evaporator and condenser, it is necessary to specify the initial length and number of heater tubes. These parameters are passed to the thermal-hydraulic modeling blocks of both evaporator and condenser.

2.6.4. Evaporator Thermal-hydraulic Calculation Blocks

Modeling the scheme of the evaporator and estimated pressure drop from refrigerant and propylene glycol 55% side require the inlet and outlet boundary conditions to be defined from each fluid component. Thus, from the thermodynamics cycle calculation, the pressures and enthalpies are transmitted to the evaporator model as inlet boundary conditions scheme for both fluid sides. For outlet boundary conditions, the cycle provides the mass flow rates for each fluid. Composition of the heat exchanger includes the geometry (length, tubes passes, tubes diameters, number, etc.) of the tubes and shell side flow simulation. Pressure drops from the evaporator returns to the thermodynamics cycle block.

2.6.5. Condenser Thermal-hydraulic Calculation Blocks

The condenser design is similar to evaporator. The thermodynamics cycle provides the pressures and enthalpies for refrigerant fluid as inlet boundary conditions and fluid mass flow rate as outlet condition parameters. In order to simulate the waterside in the tubes, the inlet pressure and temperature of water from a customer is provided from the cycle as inlet boundary conditions and water mass flow rate as outlet value. The goal of the simulation is to transmit obtained pressure drops from both fluids to the cycle block for the next iteration performance.

2.6.6. Heat load Comparison Block

Execution of the heat exchangers design process assumes the heat load values from the thermodynamic cycle and evaporator\condenser thermal-hydraulic simulation should be compared and in ideal case, equal to each other. Thus, this block provides a comparison of the hydraulic and cycle scheme heat loads. If these values are not equal, the length of tubes will change and the iteration process returns to the reassignment of the mentioned values. In order to realize the iteration process of the tubes length definition during heat loads changing in the iteration process, the secant method has been used. The algorithm is repeated until a pipe length is established so that the heat loads are equal with 1% accuracy. In the first iteration to the tubes length the first-order value was assigned. In the second iteration, a new value of the length increases by 10 %. If the thermodynamics cycle heat load values are not equal to the thermal-hydraulic scheme of heat exchanger heat load, a third iteration will be performed using the next equation:

$$L_{n+1} = L_n + (Q_n^{NET} - Q_n^{AxC}) / (Q_n^{AxC} - Q_{n-1}) \cdot (L_n - L_{n-1}) \qquad \text{Eq. (2)}$$

2.6.7. Comparison of the Parameters Block

After the heat exchanger design process is complete and the heat loads obtained in both the cycle estimation and hydraulic modeling are equal, next stage parameter comparisons can be made. In this block, user compare the estimated total parameters the system’s components, i.e. the pressure drops and enthalpies from each side of heat exchangers, in the evaporator - pressure drops and enthalpies of the propylene glycol 55% fluid and the refrigerant; in the condenser -pressure drops and enthalpies from both the waters side and the refrigerant side must be compared. If the difference is less than 1%, then the integration process proceeds to compressor preliminary design. If the difference is greater than 1%, these new values obtained from the thermal-hydraulic network of the heat exchangers must return to the cycle, and the execution process repeats until the results are within range to proceed.

2.6.8. First Compressor Preliminary Design Block

When the thermodynamic cycle and thermal-hydraulic calculation has been successfully performed, the parameters of minimum and maximum inlet (before the stage) and outlet (after that stage) pressures, mass flow rates of the fluid refrigerant, and minimum and maximum inlet temperatures from the updated thermodynamics cycle taking into account evaporator and condenser flow resistance is transmitted to the first stage preliminary design block. As a result, the efficiency of the designed first compressor is determined.

2.6.9. Second Compressor Preliminary Design Block

The second compressor design process is the same as the first. The data is transmitted from the thermodynamic cycle. The difference is in the value of the temperature and mass flow rate. Due to gas injection from the liquid separator, the temperature of the fluid is reduced. It should be noted that the whole integrated process allows asymmetrically accurate estimation of the additional injection influence on the designed heat pump. The efficiency of the second compressor has been obtained.

2.6.10. Efficiency Comparison Block

The end of the preliminary design process assumes providing a comparison of the obtained efficiency from the compressor preliminary design modules with the previously obtained efficiencies added to the thermodynamic cycle. If the results do not satisfy the established discrepancies equal to 1 %, the process returns to the beginning. The reassignment of efficiency values can be performed.

3. HEAT PUMP OFF-DESIGN ANALYSIS

When the preliminary design of the heat pump has been successfully completed, the accurate integrated approach provides the opportunity to evaluate the main cycle and its components parameters in off-design regimes. Analysis has been provided depending on the environmental temperature in order to support comfort temperature insight into the building. The regulation scheme of the heat pump assumes changing of hot water temperature depends on the outdoor temperature. The usual practice in heat pump regulation is to change the compressor's angular velocity and refrigerant mass flow rate for outlet water temperature regulation. Dependence of cold water temperature from hot water temperature is presented below (Figure 8):

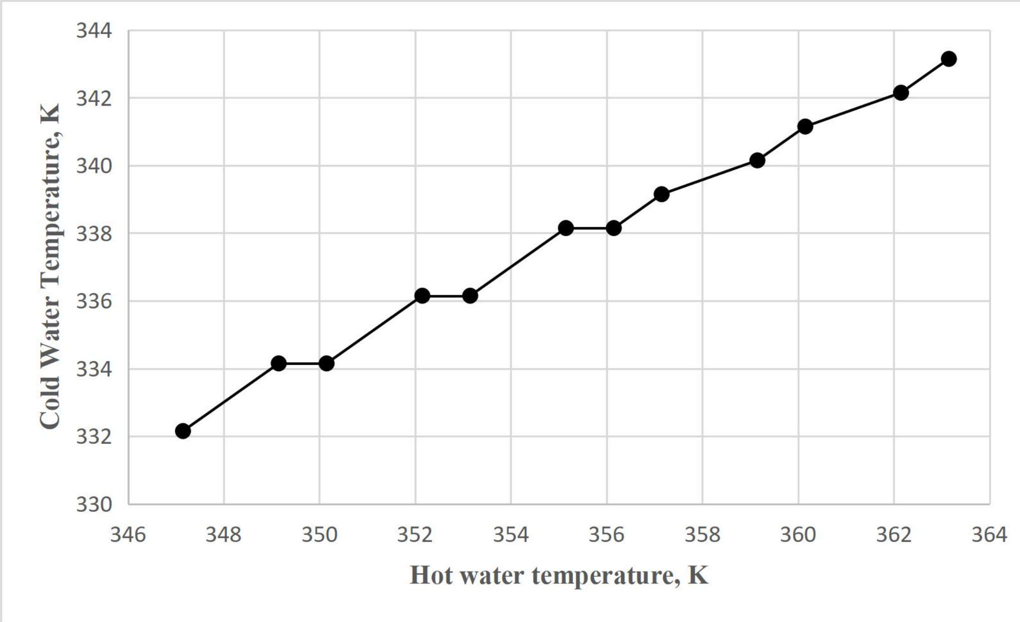

Figure 8: Dependence of Cold Water Temperature From Hot Water Temperature

The engineering task was to estimate the main characteristic of the designed heat pump - Coefficient of Performance (COP) as shown in Eq. (1) and establish dependence from rotor speed and coolant supply temperature to the customer in different regimes.

$$COP = Q_{cond} \cdot (N_1 + N_2)^{-1} \qquad \text{Eq. (3)}$$

The integrated process is similar to the preliminary design algorithm. Already engineered values of the process components are reflected in the new iteration process block workflow. The data from the cycle that passes to the thermal-hydraulic simulation and compressors analysis is the same as in the preliminary design integrated approach. The difference exists in the heat exchangers analysis, which does not require design blocks such as the definition of geometry parameters, comparison of heat loads and recalculation with changed geometry. In order to estimate dependency COP from the rotation speed of the compressor, there an additional Map block was added. The map block allows variables for analysis to be added and set up a range of any experiment. Calculated data from all blocks (thermodynamics, heat exchangers, compressor stage analysis, and comparisons of discrepancies) go to the Map block and the cycle repeats to finish the calculation of all off-design regimes. Off-design analysis has been performed for 6 different regimes of the heat pump, working in the automatic iterative process from 100% to 70% power. The regulation was performed by changing the compressors shaft rotor speed. The graphics of dependency from COP and water temperature to the customer to rotor frequency are shown below (Figures 9 and 10).

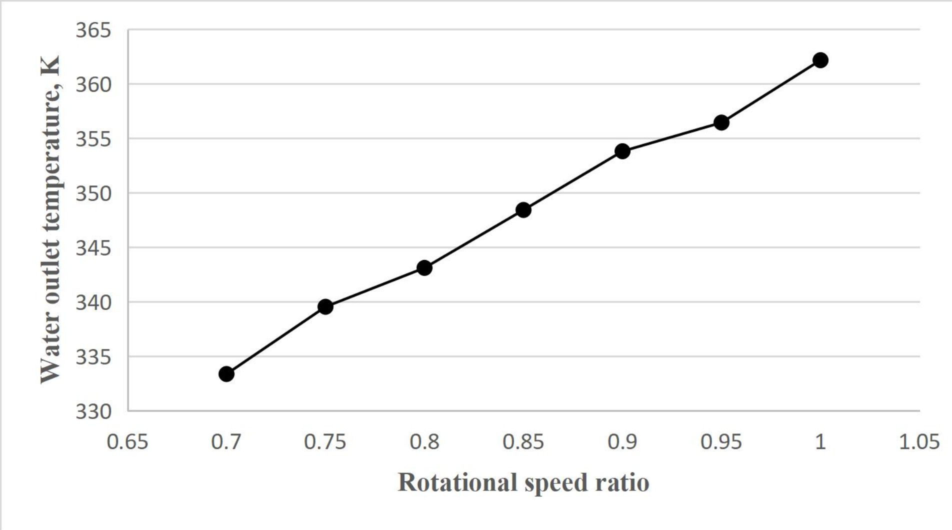

Figure 9: Dependency of Compressor Rotational Speed Ratio from Water Outlet Temperature

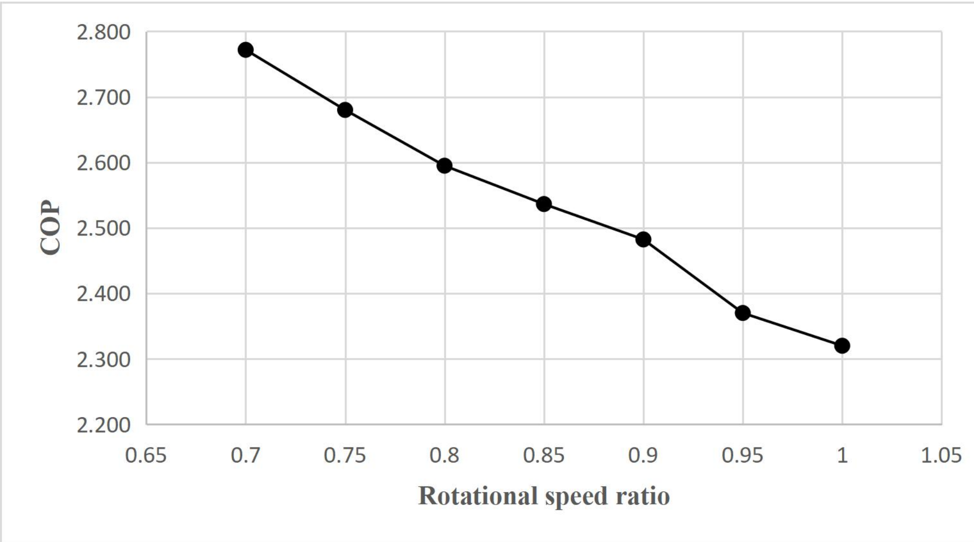

Figure 10: Dependency of Compressor Rotational Speed Ratio from Coefficient of Performance

Estimation of a preliminary designed heat pump allowed us to evaluate the main cycle parameters which depend on the compressors' angular speed. Thus, an evaluation showed that rotation frequency at different mass flow rates influences the pressure after the compressor. The algorithm of quantity regulation assumed changing the mass flow rate value at the highest possible efficiency of compressors, according to the map to ensure stable working without any breakaways. The pressure value and its efficiency effect on outlet water pressure and temperature is non-linear. By obtaining and understanding the dependence of the compressor rotational speed and water outlet temperature, this allows for considering the possible quantity regulation of the heat pump cycle . Additional estimation of these dependencies should be considered in a detailed design because of the many factors in the entire cycle. The scope of this article, however does not include consideration of the detailed design, which may be the subject of another discussion. Thus, such an integrated approach reduces the estimation time of the newly designed system and evaluates how it would work during changes in outer temperature. This research provides a quick and accurate estimation after preliminary design of the heat pump which allows for more attention to be placed on additional issues that should be improved in the detailed design phase of the system.

4. COMPARISON OF AVERAGE-BASED METHOD WITH ACCURATE INTEGRATED APPROACH

In the widely used average-based method, the calculation of thermodynamics cycle depends on boundary conditions and design task. The obtained values from the cycle estimation are then used for selecting already existing equipment. In order to prevent a lack of heat transfer in the heat exchangers - the evaporator and condenser - common practice calls for additional safety factors for heat transfer area, which could increase one up to 30%. This margin increases the cost of equipment. Another negative factor to consider is the determination of unknown parameters of the cycle as first-order values. Values of efficiencies, and hydraulic resistances in heat exchangers aren't taken into account. Therefore, the averages-based method could cause errors in the calculation, and increase the time of the project. The additional settings of the heat pump can be provided and accounted for during the checkout process. It is important to note that utilizing a design approach that uses 1D and 0D solvers provides the possibility to perform custom-made installations and rapidly estimate all the necessary parameters in one design approach, which influence the installation and provides an opportunity to obtain accurate heat areas of heat exchangers, taking into account pressure drop values, and optimal characteristics of compressors. The proposed method of accurate heat pump design has been compared with an average-based method and does not account for all the above mentioned parameters. The difference between the length of tubes, and consequently the hydraulic resistance of the heat exchangers, has been researched and is shown below (Table 5).

Table 5. Heat Exchangers Hydraulic Resistance Difference

	Averaged-based method	Integrated approach	Δ%
Evaporator			
Pressure drop (PG side),bar	2.062	2.74	24.74
Pressure drop (R717 side), bar	0.00081	0.001	19.0
Condenser			
Pressure drop (water side),bar	0.092	0.124	25.81
Pressure drop (R717 side), bar	0.00434	0.00435	0.23

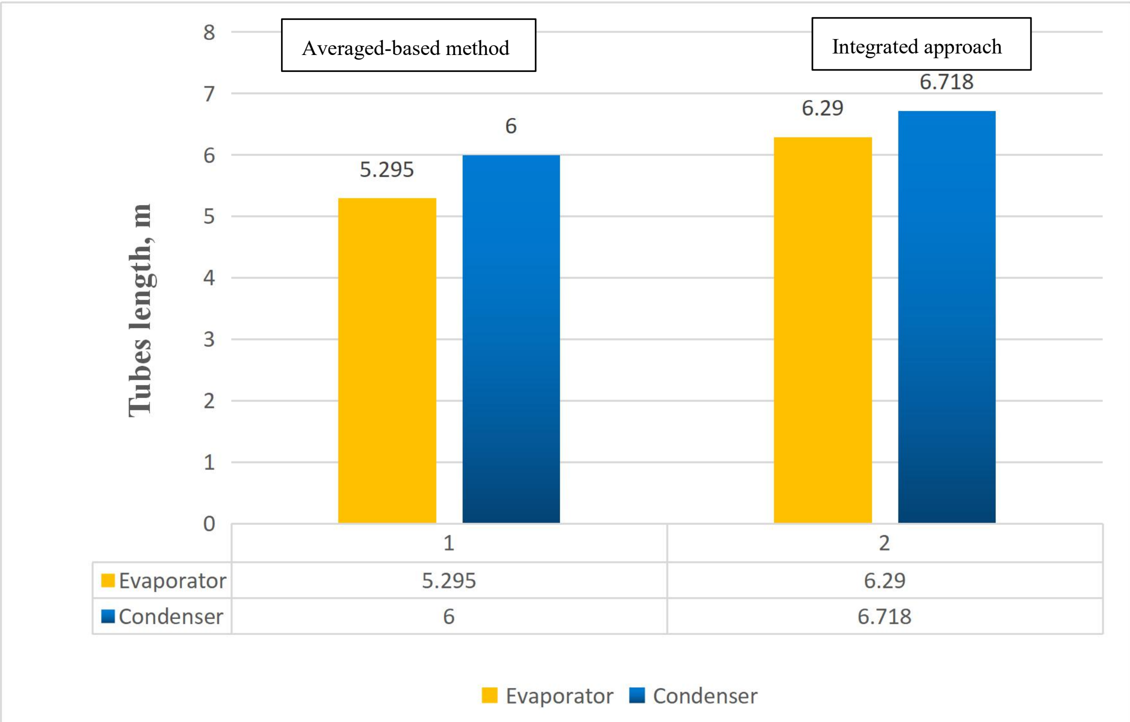

Figure 11: Tubes Length Difference Between Averaged-based Results and Integrated Approach

The inaccuracy in the heat exchanger has a significant influence on the cycle parameters. Thus increasing the lengths of the evaporator tubes by 15.81% and the condenser by 10.68% also increased the

heat loads of each heat exchanger (evaporator - from 579 kJ/s to 640 kJ/s, condenser - from 1014 kJ/s to 1038 J/s), the required mass flow rate of the propylene glycol 55%, and water. These changes were due to accurate estimation of cycle parameters. Integration of the compressors' real efficiencies to the cycle results in a decrease of the refrigerant temperature before the condenser.
Compared with previous first-order values - 0.7 after preliminary design the first compressor efficiency increased to 0.787 and the second one to 0.795.
A comparison of the different working regimes using an integrated approach verse a common approach for heat pump calculation reveals a significant difference in the obtained parameter values (represented in table 6). Disregarding the real equipment influence on the cycle directly affects the Coefficient of Performance and consequently the entire cycle. Thus, when the operating power at installation decreases, the heat pump's COP increases. Those values can be disbursed throughout the system by ignoring the real equipment influence.

Table 6. Off-design Regimes: Comparison of Approaches

Rotational speed ratio	Averaged-based method	Integrated approach	Δabs, %
1	2.095	2.320	9.698
0.95	2.138	2.370	9.789
0.9	2.201	2.482	11.322
0.85	2.257	2.536	11.002
0.8	2.359	2.595	9.094
0.75	2.447	2.680	8.694
0.7	2.498	2.772	9.906

5. CONCLUSIONS

Due to the inaccuracy of averaged-based methods of heat pump design, the authors of this article considered and proposed an advanced, integrated method. The approach enables the performance of complex engineering tasks with suitable accuracy and minimum computational design time. The proposed integrated method could be used for both preliminary and detailed design of the system and components and provides data on how the installation behaves at off-design modes. The influence of the heat exchangers' hydraulic resistance and parameters of the compressor component is very significant. Failing to take them into account leads to large errors, increasing the time of design phases and cost during the project. To offset this risk, engineers will often add about 30% heat load capacity to the heat exchangers to ensure proper function in real-world application. The proposed approach can be used to decrease additional reserves associated with inaccurate calculations. In the preliminary phase, this approach provides information about possible issues that should be resolved in the detailed phase. Such an approach can be applied in a real, practical setting to increase the quality of any design, as well as manufacturing.

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NOMENCLATURE

COP	Coefficient of Performance	N_1	First compressor power on shaft, W
Q_{cond}	Condenser heat load, kJ/s	N_2	Second compressor power on shaft, W
L_{n+1}	Tube length in next iteration, m	L_n	Tube length in current iteration, m
L_{n-1}	Tube length in previous iteration, m	Q_n^{NET}	Heat load from thermal-hydraulic simulation in current iteration, kJ/s
Q_n^{AxC}	Heat load from thermodynamic cycle simulation in current iteration, kJ/s	Q_{n-1}	Heat load in previous iteration, kJ/s
eff	Compressor internal efficiency	Δh_{is}	Isentropic enthalpy difference, $J/kg\cdot K$

Δh	Enthalpy difference, $J/kg \cdot K$	x	Fluid quality
T	Temperature, K		

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