A heat exchanger system is described that includes an inlet and an outlet for a first fluid and a heat exchanger between the inlet and the outlet wherein the first fluid circulates, wherein the heat exchanger comprises at least one deflector to guide the flow of a second fluid. A method is also described to exchange heat between a first and a second fluid using free convection velocity field to create forced convection in the heat exchanger of a heat exchanger system. A method to exchange heat between a first and a second fluid comprising providing a heat exchanger system between the first and the second fluids, said heat exchanger system comprising a heat exchanger wherein the first fluid circulates and increasing the flow turbulences of a second fluid around the heat exchanger.
Fig. 4
CROSS REFERENCE TO RELATED APPLICATIONS

CROSS REFERENCE TO RELATED APPLICATIONS


FIELD

[0002] The present disclosure relates generally to subsea cooling of fluids. More particularly, the present disclosure relates cooling of fluids in a subsea environment using a heat exchanger with free convection on the ambient seawater side.

BACKGROUND

[0003] Subsea compression is a relatively recent technology developed to enhance the lifetime of existing fields. Installation of subsea compression systems enables balancing of the reservoir depletion over time. The production plateau is extended while also increasing total recovery of fields. These systems face substantial challenges both from a technical and commercial point of view. In the absence of any upstream processing on the well stream, the production fluid flow is multiphase containing mainly gas, but also condensate, water and sand. Additionally, the equipment faces various subsea constraints including availability, flexibility and robustness.

[0004] Subsea process cooling is an important aspect for subsea applications. Standard heat exchangers or coolers with forced convection on ambient side requires additional equipment, such as sea water circulation pumps, guiding piping, and/or shells to force the coolant fluid towards the cooling area. Circulation pumps rely on a power supply as well as control systems. Subsea constraints make reliability and operation of forced convection equipment both challenging and expensive.

[0005] A free convection velocity field generated by buoyancy force is usually much lower than velocity intensity that can be obtained in forced convection. Therefore, corresponding heat transfer is less efficient with free convection when compared to forced convection. Since the heat transfer performance is inherently lower, the exchange surface needs to be increased in order to maintain sufficiently high cooling.

SUMMARY

[0006] According to some embodiments a heat exchanger system is described for transferring heat between a first fluid and a surrounding ambient second fluid. The system includes: an inlet configured to accept the first fluid; an outlet configured to expel the first fluid; a plurality of vertically oriented parallel conduits positioned between the inlet and the outlet configured to carry the first fluid therein, the conduits each having an exterior surface that is exposed to the surrounding second fluid when the system is submerged in the second fluid, wherein heat is transferred between the first fluid flowing through the conduits and the second fluid flowing a vertical direction along the exterior surfaces of and parallel to the conduits by free convection; and at least one deflector fixedly mounted exterior to the conduits configured to impart non-vertical momentum in the flowing second fluid thereby enhancing heat transfer between the first fluid and the second fluid.

[0007] According to some embodiments, the conduits are tubular pipes grouped into one or more groups, with the tubular pipes of each group being arranged symmetrically about a central axis of the group. The defectors can be a number of horizontally oriented disks (such as two, three, four or five) for each group of tubular pipes. The group of pipes and the horizontally oriented disks can each have a large central opening configured to allow free passage of the second fluid there-through, and the horizontally oriented disks can be configured to force the second fluid into and out of the central opening of the group of pipes, thereby enhancing heat transfer.

[0008] According to some embodiments, defectors can be non-horizontally oriented structures that are asymmetric with respect to the central axis of the group. The structure can be configured to impart momentum in a tangential direction with respect to the central axis of the group. According to some embodiments, the non-horizontally oriented structures can be helical in shape.

[0009] According to some embodiments, the conduits can be tubular pipes arranged into a rectangular pattern of columns and rows. The defectors can be one or more horizontally arranged baffles or, according to some embodiments, they can be non-horizontally arranged.

[0010] According to some embodiments, no powered equipment is used to force the second fluid to flow past the conduits. According to some embodiments, the coolant second fluid is seawater, and the first fluid includes hydrocarbon gas produced from one or more wellbores penetrating an underground rock formation.

[0011] According to some embodiments, a method is described for transferring heat between a first fluid and an ambient second fluid surrounding a plurality of conduits through which the first fluid flows. The method includes: exposing exterior surfaces of the plurality of conduits to the surrounding second fluid; and flowing the first fluid through the plurality of conduits, wherein the plurality of conduits are vertically oriented and parallel to each other, and heat is transferred between the first fluid and the second fluid flowing in a vertical direction along the exterior surfaces of an parallel to the conduits by free convection and wherein at least one stationary deflector is fixedly mounted exterior to the conduits configured to impart non-vertical momentum in the flowing second fluid thereby enhancing heat transfer between the first fluid and the second fluid. According to some embodiments, the flowing includes pumping the first fluid through the plurality of pipes. According to some embodiments, heat is transferred from the first fluid to the second fluid thereby cooling the first fluid. According to some other embodiments, heat is transferred from the second fluid to the first fluid thereby heating the first fluid.

[0012] According to some embodiments, a heat exchanger system is described that includes: an inlet and an outlet for a first fluid, and a heat exchanger between the inlet and the outlet where the first fluid circulates. The heat exchanger includes at least one deflector to guide the flow of an ambient second fluid. According to some embodiments, the deflector guides the flow of the second fluid to transfer the vertical momentum from the gravity induced free convection flow of the second fluid to horizontal velocity. According to some embodiments, the deflector is shaped depending upon the
heat exchanger design configuration. In one example, the heat exchanger comprises tubular pipes and the at least one deflector includes shapes surrounding the pipes. According to some embodiments, the deflectors are shaped as a horizontal disk, staggered plates or helical screw-like shapes.

According to some other embodiments, a method is described for exchanging heat between a first and a second fluid using a free convection velocity field to create a form of non-powered “forced” convection in the heat exchanger without the use of a pump or other powered equipment.

According to some other embodiments, a method is described for exchanging heat between a first and a second fluid that includes increasing flow turbulences of the second fluid around the heat exchanger wherein the first fluid is circulated. In one example, the method includes increasing the velocity field, the turbulence level and the flow mixing of the second fluid. In another aspect the method includes breaking the thermal layer of the second fluid by creating transverse flow and unsteady drag effects within the second fluid.

According to some embodiments, the method includes increasing the momentum of the ambient fluid around the heat exchanger. In one embodiment, increasing the momentum around the heat exchanger includes increasing the amount of the second fluid participating in.

According to some embodiments, the method includes breaking boundary layers of the second fluid so that layers of the second fluid remote from the heat exchanger are dragged towards the heat exchanger. It is known that free convection patterns generate a thermal boundary layer that tends to act as insulation which offsets the heat transfer increase due to the vertical momentum increase of the second fluid.

These together with other aspects, features, and advantages of the present disclosure, as well as the various features of novelty, which characterize the invention, are pointed out with particularity in the claims annexed to and forming a part of this disclosure. The above aspects and advantages are neither exhaustive nor individually or jointly critical to the spirit or practicality of the disclosure. Other aspects, features, and advantages of the present disclosure will become readily apparent to those skilled in the art from the following description of exemplary embodiments in combination with the accompanying drawings. Accordingly, the drawings and description are to be regarded as illustrative in nature, and not restrictive.

BRIEF DESCRIPTION OF THE DRAWINGS

To assist those of ordinary skill in the relevant art in making and using the subject matter hereof, reference is made to the appended drawings, in which like reference numerals refer to similar elements:

FIG. 1 is a schematic representation of a subsea production setting with which some of the cooling systems and methods described herein can be used, according to some embodiments;

FIG. 2 is a perspective view of a heat exchanger, according to some embodiments;

FIGS. 3-1 and 3-2 illustrate further aspects of a heat exchanger having horizontal baffles, according to some embodiments;

FIG. 4 is a perspective view illustrating aspects of a heat exchanger having helical-shaped baffles, according to some embodiments;

FIG. 5 is a perspective view illustrating aspects of a heat exchanger having a staggered combination of horizontal baffles, according to some embodiments;

FIG. 6 is a graph showing the cooler group of pipes and single pipe cooling performances versus the mass flow according to the experimental results;

FIGS. 7, 8-1 and 8-2 illustrate flow patterns on the ambient (sea water) side of the cooling group of pipes having horizontal and helically-shaped baffles, according to some embodiments;

FIGS. 9-1, 9-2 and 9-3 illustrate further aspects of ambient flow patterns and heat transfer for heat exchangers, according to some embodiments; and

FIGS. 10-1, 10-2 and 10-3 illustrate using one or more baffles in different configurations for a non-symmetrical arrangement of cooling system pipes, according to some embodiments.

It should be understood that the drawings are not to scale and that the disclosed embodiments are sometimes illustrated diagrammatically and in partial views. In certain instances, details that are not necessary for an understanding of the disclosed method and apparatus, or that would render other details difficult to perceive may have been omitted. It should be understood that this disclosure is not limited to the particular embodiments illustrated herein.

DETAILED DESCRIPTION

Some embodiments will now be described with reference to the figures. Like elements in the various figures may be referenced with like numbers for consistency. In the following description, numerous details are set forth to provide an understanding of various embodiments and/or features. However, it will be understood by those skilled in the art that some embodiments may be practiced without many of these details and that numerous variations or modifications from the described embodiments are possible. As used here, the terms “above” and “below”, “up” and “down”, “upper” and “lower”, “upwardly” and “downwardly”, “upstream and downstream”, and other like terms indicating relative positions above or below a given point or element are used in this description to more clearly describe certain embodiments. However, when applied to equipment and methods for use in wells that are deviated or horizontal, such terms may refer to a left to right, right to left, or diagonal relationship, as appropriate.

It will also be understood that, although the terms first, second, etc. may be used herein to describe various elements, these elements should not be limited by these terms. These terms are only used to distinguish one element from another. For example, a first object or step could be termed a second object or step, and, similarly, a second object or step could be termed a first object or step, without departing from the scope of the invention. The first object or step and the second object or step are both objects or steps, respectively, but they are not to be considered the same object or step.

The terminology used in the description of the invention herein is for the purpose of describing particular embodiments only and is not intended to be limiting. As used in the description and the appended claims, the singular forms “a”, “an” and “the” are intended to include the plural forms as well, unless the context clearly indicates otherwise. It will also be understood that the term “and/or” as used herein refers to and encompasses any and all possible combinations of one
or more of the associated listed items. It will be further understood that the terms “includes,” “including,” “comprises,” and/or “comprising,” when used in this specification, specify the presence of stated features, integers, steps, operations, elements, and/or components, but do not preclude the presence or addition of one or more other features, integers, steps, operations, elements, components, and/or groups thereof. 

As used herein, the term “if” may be construed to mean “when” or “upon” or “in response to determining” or “in response to detecting,” depending on the context. Similarly, the phrase “if it is determined” or “if [a stated condition or event] is detected” may be construed to mean “upon determining” or “in response to determining” or “upon detecting [the stated condition or event]” or “in response to detecting [the stated condition or event],” depending on the context. 

In the specification and appended claims, the terms/phrases “connect,” “connection,” “connected,” “in connection with”, and “connecting” are used to mean “in direct connection with” or “in connection with one or more elements,” and the term “set” may mean one element or “more than one element.” Further, the terms “couple”, “coupling”, “coupled”, “coupled together”, and “coupled with” are used to mean “directly coupled together” or “coupled together via one or more elements.”

In the following detailed description of some embodiments, reference is made to the accompanying drawings, which form a part hereof, and within which are shown by way of illustration specific embodiments by which the invention may be practiced. It is to be understood that other embodiments may be utilized, and structural changes may be made without departing from the scope of the invention.

The particulars shown herein are by way of example and for purposes of illustrative discussion of the embodiments of the present invention only and are presented in the cause of providing what is believed to be the most useful and readily understood description of the principles and conceptual aspects of the present invention. In this regard, no attempt is made to show structural details of the present invention in more detail than is necessary for the fundamental understanding of the present invention, the description taken with the drawings making apparent to those skilled in the art how the several forms of the present invention may be embodied in practice. Further, like reference numbers and designations in the various drawings indicate like elements.

The following abbreviations and relations shall be used herein:

- **CFD**: Computational Fluid Dynamics;
- **RANS**: Reynolds Averaged Navier-Stokes;
- **SP**: Single Pipe;
- **SST**: Shear Stress Transport;
- **WGC**: Wet Gas Compressor;
- **amb**: Ambient side of the cooler;
- **ext**: Cooler external side;
- **ini**: Cooler inlet, process side;
- **int**: Cooler internal side;
- **out**: Cooler outlet, process side;
- **A**: Area, [m²];
- **Cp**: Heat Capacity, [J/kg/K];
- **D**: Pipe Diameter, [m];
- **e**: Internal energy, [J/kg];
- **g**: Gravity, [m/s²];
- **HTC, h<sub>in</sub>, h<sub>out</sub>**: Heat Transfer Coefficient, [W/m²·K];
- **k**: thermal conductivity, [W/m/K];
- **LMTD**: Log Mean Temperature Difference

\[
LMTD = \frac{(T_{sat} - T_{amb}) - (T_{sat} - T_{amb})}{\ln\left(\frac{T_{sat} - T_{amb}}{T_{sat} - T_{amb}}\right)}
\]

- **N<sub>u</sub>** = \(\frac{hL}{k}\) Nusselt number
- **Pr** = \(\frac{\mu C_p}{k}\) Prandtl number
- **Ra** = \(\frac{g\beta \Delta T L^3 C_p}{\mu k}\) Rayleigh number
- **Re** = \(\frac{nu D}{\mu}\) Reynolds number

FIG. 1 is a schematic representation of a subsea production setting with which some of the cooling systems and methods described herein may be used, according to some embodiments. Two wells, 112 and 114, are being used to extract production fluid from subterranean formation 100. The produced fluids from wells 112 and 114 move into manifold 118 from wellheads 122 and 124 via sea floor pipelines 146 and 144 respectively. Manifold 118 includes a compressor system 152 and according to some embodiments a subsea heat exchanger system 150. The production fluid flows from manifold 118 upwards through flowline 132 to a surface production platform 130 on the sea surface 104 of seawater 102. As will be described in greater detail infra, the heat exchanger system 150 has a plurality of parallel pipes through which the production fluid is produced. The external surface of the pipes is exposed to the seawater 102 which cools the production fluid as will be described in further detail infra.

Note that although many embodiments are described herein as being used in an exemplary application of subsea compression systems, the methods and structures described herein are generally applicable to many other types of heat exchangers. According to some embodiments, the heat exchanger system components described herein are used for nuclear power generation cooling (or heating) applications. According to some other embodiments, the heat exchanger components described herein are used for heating and/or cooling applications in chemical processing applications.

Heat exchanger design. The cooling principle for heat exchangers is to transfer heat from one fluid (the cooled fluid) to another (the coolant fluid). Heat exchangers are commonly designed using a forced convection heat transfer principle for both the cooled and the coolant fluids. This is due to the higher heat transfer rate that can be obtained using forced convection versus free convection. As used herein, the term “free convection” refers to a mechanism or type of heat transport in which the fluid motion is not generated by any external power source (e.g., pump, fan, suction device, etc.), but rather only by density differences in the fluid occurring due to temperature gradients. Note that although several
embodiments are described herein with respect to the application of cooling, the methods and structures described herein are equally applicable to heating applications. In such cases the heat is transferred from the ambient fluid (the heating fluid) to the fluid flowing through the pipes (the heated fluid).

[0067] The subsea environment is complex and aggressive. For example, routine maintenance, inspection and cleaning possibilities are both limited and challenging. In the case of forced convection, difficulties are further increased by the nature and the multiplicity of the equipment (pumps for example) used to generate forced convection on the ambient side (the coolant fluid). Conventional forced-forced convection (i.e. both the cooled fluid and coolant fluid are pumped) heat exchanger technology is hence not well suited for subsea applications. It has been found that a passive design based on free convection on the ambient side (the coolant fluid) is often more appropriate to face to challenges of the subsea.

[0068] According to some embodiments, a heat exchanger system is designed both to give a large turn down on thermal performance for operation flexibility and, for subsea applications, to handle flow assurance issues like sand accumulation, hydrates formation, wax deposition, etc.

[0069] FIG. 2 is a perspective view of a heat exchanger, according to some embodiments. Heat exchanger 150 includes a plurality of vertical pipes 200 and distributing pipes 206, inlet manifolds, outlet manifolds and collecting pipes 208 to distribute/collect uniformly the flow of the cooled fluid. In particular, the vertical pipes 200 are arranged in eight groups of pipes that are symmetrically arranged about a central axis. The cooled fluid enters through inlet 202 which leads to distribution pipes 206. The distribution pipes 206 distribute the cooled fluid, as is shown by the solid arrows to each of the eight groups of pipes. Each group of pipes is fed by an inlet manifold. For example, group of pipes 210 includes an inlet manifold 212. The cooled fluid exits the pipes 200 through outlet manifolds and collecting pipes 208, which lead to outlet 204. According to one example, each group, such as group 210, has 33 pipes arranged symmetrically about a central longitudinal axis of the group. The manifolds, distribution and collecting pipes are designed to uniformly distribute and collect the multiphase flow to and from the cooling pipes while avoiding excessive head loss. The heat exchanger 150 is arranged such that it is symmetric and modular hence adaptable to different cooling requirements. For further details of such symmetric heat exchangers, see U.S. patent application Ser. No. 13/259,789, which is incorporated by reference herein. Each of the groups of pipes 200 has 5 horizontal baffles mounted perpendicular to the vertically oriented pipes. In group 210, the horizontal baffles 230, 232, 234, 236 and 238 are mounted around pipes 220. As will be described in further detail, infra., the baffles arranged as shown are effective in altering the predominantly vertical flow of the free convecting coolant fluid so as to significantly enhance cooling performance. Note that although the coolant fluid is shown in FIG. 2 as flowing from top to bottom, the techniques described herein are also applicable to heat exchangers in which the cooled (or heated in the case of heat exchangers operating in a heating mode) fluid flow through the pipes in an upwards direction.

[0070] According to some embodiments, the free convection velocity field is used to create a form of non-powered “forced” convection on the ambient side (coolant fluid). According to some embodiments, described heat exchange systems, the heat exchanger includes one or more external shapes to guide the coolant fluid flow and transfer the vertical momentum of the freely convecting coolant fluid from the gravity field to generate horizontal velocity. Various shapes can be used to both generate either radial velocity and circumferential velocity with respect to the longitudinal axis of the pipes.

[0071] FIGS. 3-1 and 3-2 illustrate further aspects of a heat exchanger having horizontal baffles, according to some embodiments. FIG. 3-1 is a perspective view showing further detail of a single cooler group 210, that may be combined with other cooler groups to form a heat exchange system as shown for example in FIG. 2. Note that although many of the embodiments described herein refer to the heat exchanger as a “cooler” in which a “cooled fluid” within the pipes is being cooled by an ambient “coolant fluid,” all of the methods and structures described herein can be equally applied to heat exchangers operating as a “heater” in which a “heated fluid” within the pipes is being heated by an ambient “heating fluid.”

[0072] In this particular example, group 210 is made up of 33 vertically oriented pipes 200 that are symmetrically arranged in two concentric rings about a central axis 310 such that there is a large central open space within group 210. As can be seen, each of the five horizontal baffles 230, 232, 234, 236 and 238 is a disk-shaped piece mounted horizontally (i.e. perpendicular to the vertically oriented pipes 200). FIG. 3-2 is a plan view of single baffle. As can be seen each of the baffles (in this case baffle 238) has a large central opening 300 that is dimensioned to match the central opening of the arrangement of pipes 200. Referring again to FIG. 3-1, the direction of flow of the cooled fluid is shown by the solid arrow 320 that in this example is downwards (the negative z-direction). The coolant fluid generally moves in an upward vertical direction (the positive z-direction) due to buoyancy resulting from differences in fluid density. When the vertically moving coolant fluid encounters a horizontal baffle, its direction is altered as shown by the dotted arrows, and as will be described in greater detail infra. Note that in the case of simple horizontal baffles such as shown in FIGS. 3-1 and 3-2, the induced velocity of the coolant fluid includes a substantial radial component. As will be described in greater detail infra, an unexpected result of the baffles is to: (1) force the ambient water “inside the group” to flow out (out of area 300) to the area outside of the group (into area 302) as illustrated by dotted arrow 304; and (2) force the ambient water “outside the group” (out of area 302) to flow inside the group (into area 300) as illustrated by dotted arrow 306.

[0073] FIG. 4 is a perspective view illustrating aspects of a heat exchanger having helical-shaped baffles, according to some embodiments. A single cooler group 400 of vertically oriented pipes 220 are shown as in the FIG. 3-1, that may be combined with other cooler groups to form a heat exchange system as shown for example in FIG. 2. In this particular example, two helical baffles 430 and 432 are attached to the external surface of the pipes 220 as shown. It has been found that by arranging the baffle structures at a non-horizontal angle to the pipes the pipes can aid in overall cooling performance in many applications. In the case shown in FIG. 4, the helical-shaped baffles 430 and 432 are effective in inducing substantial velocity in the coolant fluid in both the radial and circumferential directions with respect to the group central axis 310 as illustrated by the dotted arrows.

[0074] FIG. 5 is a perspective view illustrating aspects of a heat exchanger having a staggered combination of horizontal baffles, according to some embodiments. Many other shapes...
and arrangements of baffles (such as horizontal, helical, and/or diagonal, etc.) or other structures can be used to transfer momentum in the coolant fluid from vertical directions to non-vertical directions. FIG. 5 shows one such example that is a combination of baffle sections 530, 532, 534, 536 and 538, which are provided to impart non-vertical velocity on the coolant fluid. In this example, the baffle sections impart a substantial radial velocity with respect to the central axis 310 of the cooler group 500.

[0075] In order to evaluate the proposed heat exchanger system performance and to compare it with empirical results for simple geometries, the heat exchanger part of the heat exchanger system and a single pipe were run in parallel. The single pipe had the same characteristics (diameter, length, thickness, material . . . ) as the cooler pipes. The cooling pipe length was 4.3 meters. In an example test, a slipstream of dry nitrogen was provided from an existing compressor discharge and routed through the test pipes. Flow/pressure through the pipes was controlled by means of the chokes on the compressor outlet and control valves downstream test pipes. The test operating conditions, taken in this example test, are listed in Table 1 for a flow of process fluid to be cooled in a subsea environment:

<table>
<thead>
<tr>
<th>TABLE 1</th>
<th>example of the test operating conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Units</td>
<td>Value</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>60-90 °C</td>
</tr>
<tr>
<td>Ambient temperature</td>
<td>12 °C</td>
</tr>
<tr>
<td>Inlet pressure</td>
<td>20-40 kPa</td>
</tr>
<tr>
<td>Massflow (per pipe)</td>
<td>0.1-0.4 kg/s</td>
</tr>
</tbody>
</table>

[0076] The process flow was measured by means of Coriolis mass flow meters. Temperature was measured upstream and downstream the test objects. Pressure was measured upstream the test objects. Six individual temperature measurements were made in the seawater along the test objects to determine ambient temperature and to check any temperature layering in the pit. The head loss across the bundle of pipes (or group of pipes) was in addition measured using a differential pressure sensor.

[0077] Using pressure and temperature measurements at the test pipes inlet, the gas thermodynamic properties (density and heat capacity) are calculated and hence the amount of heat removed to the process fluid (the cooled fluid) passing through the pipes was obtained.

[0078] The thermal performances of the two test objects (the group of cooler pipes and the single cooler pipe) were characterized for different mass flow amounts. The global heat transfer coefficient U can be calculated according to the following equations.

[0079] The heat transfer is defined by:

\[ Q = U A \Delta T \]

with A the object area and \( \Delta T \) the temperature difference between the process gas and the ambient water.

[0080] The heat transfer Q is directly related to the heat removed to the gas:

\[ Q = \dot{m}_{\text{gas}} C_{p_{\text{gas}}} (T_{\text{outlet}} - T_{\text{inlet}}) \]

[0081] Replacing the temperature difference between the process gas and the ambient water by the Log Mean Temperature Difference (LMTD) as the process gas is not constant all along the cooling pipes, the global heat transfer coefficient is calculated using the following formula:

\[ U = \frac{n_{\text{gas}} C_{p_{\text{gas}}} (T_{\text{outlet}} - T_{\text{inlet}})}{\text{LMTD}} \]

[0082] It should be noted the global range from the tests is quite uncommon due to the wide dimensions and cooling capacities of the studied case (see Table 2: global results). The length scale, the temperature difference and the total cooling load are likely to be outside normal test conditions used to define the empirical correlations. Comparison of Nusselt number based on 3 different approaches which are experimental, analytical and numerical, for such as high Rayleigh number range up to \( 10^{12} \) makes this study very valuable.

<table>
<thead>
<tr>
<th>TABLE 2</th>
<th>Global Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Units</td>
<td>Min Value</td>
</tr>
<tr>
<td>Massflow (cooler)</td>
<td>kg/s</td>
</tr>
<tr>
<td>Heat transfer (cooler)</td>
<td>kW</td>
</tr>
<tr>
<td>Reynolds number (gas)</td>
<td></td>
</tr>
<tr>
<td>Rayleigh number (ambient)</td>
<td></td>
</tr>
</tbody>
</table>

[0083] FIG. 6 is a graph showing the cooler group of pipes and single pipe cooling performances versus the mass flow according to the experimental results. In this case the group of pipes had two horizontal baffles arranged perpendicular to the cooler pipes. The cooling performance U is shown by curve 610 for the group of pipes with the baffles and by curve 612 for the single pipe without any baffles. Using 2nd order polynomial interpolation the performance ratio is obtained (dotted line 614 using right scale). The cooling performance of the group having baffles is about 23% higher than the case of the single pipe without baffles. Note that according to some embodiments, greater numbers of baffles, such 5 baffles as shown in FIGS. 2 and 3-1, will result in an even greater increase in heat transfer performance.

[0084] The measured performance increase given in the FIG. 5 is defined by:

\[ \Delta p_{\text{cooler vs. sp}} = \frac{U_{\text{cooler}}}{U_{\text{sp}}} \]

[0085] This relation is explained by the pipes interaction for the cooler group with the baffles. The heat transfer from the pipe’s external surfaces to the ambient seawater is driven by free convection. An external flow is generated due to the density variation induced by the temperature increase caused by proximity to the pipes. In the case of the group of cooler pipes, the pipes’ closeness causes the seawater flows to interact. The obtained momentum is thus higher than the one obtained with a single pipe. This phenomenon, called chimney effect, is described in further detail infra.

[0086] Simulations. The heat transfer from the process gas (the cooled fluid) to the ambient (cooler fluid) observed on the test objects can be decomposed into the 3 following features: (1) internal forced convection between the bulk gas and the internal surfaces of the pipes; (2) conduction across the
walls of the pipes; and (3) external free convection between the external surfaces of the pipes and the ambient water.

Based on the physical mechanisms as split above, the global heat transfer coefficient can be defined by the following formula:

\[
U = \frac{1}{\frac{D_{\text{int}}}{2k_{\text{steel}}} + \frac{1}{D_{\text{ext}}} + \frac{1}{D_{\text{ext}} h_{\text{ext}}}}
\]

The thermal performance difference obtained between the SP (single pipe) and the cooler group of pipes is related to the free convection pattern and intensity on the ambient side. In order to characterize and to understand in detail the phenomena occurring and the complex three-dimensional (3D) flow that develops around the pipes, Computational Fluid Dynamics (CFD) simulations are performed on both the cooler group and the single pipe. The case with a mass flow of 0.22 kg/s has been studied using commercial CFD software.

Numerical method. The geometry simulated consists of three domains: (1) the gas flowing inside the pipes; (2) the solid pipe walls; and (3) the ambient side of the pipes.

The two flows (ambient and gas) are described by the Reynolds Averaged Navier Stokes (RANS) system coupled to the internal energy equation for the ambient side by the buoyancy force based on the Boussinesq assumption:

\[
\begin{align*}
\frac{\partial p}{\partial t} + \nabla \cdot (\rho u) &= 0 \\
\frac{\partial (\rho u)}{\partial t} + \nabla \cdot (\rho u u) &= -\nabla p + \nabla \cdot \tau + S_m \\
\frac{\partial (\rho e)}{\partial t} + \nabla \cdot (\rho e u) &= \nabla \cdot (\lambda \nabla T) - p\nabla \cdot u + \tau : \nabla u.
\end{align*}
\]

The momentum source is hence for the water domain:

\[
S_{\text{ext}} = p_{\text{ext}} \beta (T - T_{\text{ref}})
\]

Turbulence is solved using the Shear Stress Transport (SST) model. Both fluids, water and gas, are considered incompressible. In the example simulations, the solid domain material is stainless steel described by its thermal conductivity.

The boundary condition for the top and bottom faces of the ambient side is set to opening. In the proposed example, the lateral face representing infinite is set to wall with free slip condition and imposed temperature to 12°C. For the gas domain, standard incompressible boundary conditions are used, i.e. mass flow imposed at the inlet (top) and static pressure imposed at the outlet (bottom). The inlet temperature is fixed to 90°C.

Single pipe simulation. As shown in Table 3, the heat transfer for the single pipe is very well simulated. The discrepancy between test data and simulation is only about 5%. The difference between test data and empirical correlation is about 10%, which is also acceptable regarding correlation accuracy. Comparison of the CFD and test results for the single pipe is a step that validates the numerical approach.

**Comparison between numerical and analytical approaches with test results**

Cooler group simulation. Due to the compressed gas flowing inside the pipes, the external pipe wall was warm and thus a vertical flow in the ambient is generated due to free convection. The gas flow continuously delivers heat to the ambient side and the averaged gas temperature drops off from about 90°C, down to about 60°C.

FIGS. 7 8-1 and 8-2 illustrate flow patterns on the ambient (sea water) side of the cooling group of pipes having horizontal and helically-shaped baffles, according to some embodiments. In the case of the cooler group 210 with horizontal baffles shown in FIG. 7, on the seawater side starting from the bottom, the water free convection flow is routed from the ambient into the vicinity of the pipes. What is remarkable and quite unexpected with the cooler design having horizontal baffles is that the baffles block the vertical flows and force the ambient water “inside the group” (area 300) to flow out and in again. This phenomenon which generates radial flows around the baffles was found to have a very positive effect on the heat transfer as: (1) it increases the mean velocity field and hence the heat transport; (2) it increases the turbulence level and the flow mixing around the cooling pipes; and (3) the flow is convective and not affected by the baffle blocks.

It has been found that the baffles generate secondary flows of the ambient fluid around the baffles. The seawater is ejected from the area in the center of the group of pipes just below each baffle (positive radial velocity with respect to central axis 310) and then just above each baffle the water is routed back into the central area (negative radial velocity). This transverse flow generated firstly increases the momentum level and by consequence the heat removal is improved. A second aspect is that it increases the turbulence level creating some turbulent structures.

FIG. 8-1 illustrates simulated streamlines for the cooler group 400 shown in FIG. 4 having helical-shaped baffles. As can be seen, a substantial circumferential velocity is imparted to the ambient fluid flow that has been found to further increase the effects described above with respect to the horizontal baffle arrangement. It has thus been found that helical baffles are often even more effective in increasing heat transfer than designs using a horizontal baffle arrangement. FIG. 8-2 illustrates simulated streamlines for cooler group 400 showing the substantial circumferential velocity imparted to the ambient fluid flow using helical baffles, according to some embodiments.

Test results and comparisons of the thermal performance between the single pipe and the cooler group highlighted an increase of the heat transfer for the cooler group. Due to the similarity of the two test objects and the physical mechanisms decomposition of heat transfer, the free convection on the ambient side has been identified as the key phenomena to explain test results deviation.
The CFD analyses performed on the two objects revealed a complex 3D flow development for the cooler. This 3D flow is responsible for the external HTC increase. FIGS. 9.1, 9.2 and 9.3 illustrate further aspects of ambient flow patterns and heat transfer for heat exchangers, according to some embodiments. FIG. 9.1 shows a pipe 220 through which the cooled fluid is pumped. The coolant fluid exposed to the exterior surface of the pipe 220 moves upward due to buoyancy forces as shown by arrow 910. The moving ambient fluid has an outer boundary 920 as shown. It has been found that two effects interact and are coupled enhancing heat transfer, according to some embodiments. First the pipes bundle generates the high intensity vertical momentum due to free convection (as shown FIG. 9.1), and secondly the baffles transfer this vertical flow into radial and azimuthal (or circumferential) flows. FIG. 9.2 illustrates a radial flow direction of ambient fluid as described. FIG. 9.3 shows the effect of transverse ambient fluid flow (e.g. radial or circumferential) and the Von Karman vortex development on the low-pressure (downstream) side of the cylindrical pipe. The flow patterns created by the baffles increases the momentum in the pipe vicinity, which amplifies the heat removal and likewise generates Von Karman structures with high turbulence level.

Based on the analysis described supra, a new empirical correlation can be described. According to some embodiments, the heat exchanger is a cooler group with a passive design. That is, no additional powered equipment is used to create forced flow on the ambient side. Nevertheless the interaction between the pipes and baffles is such that free convection flow generated by one pipe tending to act similarly to a forced convection flow source for the neighboring pipes. An expression of the following form can be used describe the mixed convection:

\[ Nu_{mixed} = Nu_{free} \times \frac{Nu_{forced}}{Nu_{free}} \]

It is important to note that analysis and results are dependent of the configuration. Three special cases can be identified: (1) buoyancy induced flow and forced flow parallel with the same direction; (2) buoyancy induced flow and forced flow parallel in opposite directions; (3) buoyancy induced flow and forced flow perpendicular, such as provided by several embodiments described herein.

When analyzing a heat exchanger design, challenges can relate to the fact that there can be two different characteristic lengths representative of the composite mechanism. It hence makes it difficult to establish the combination of the characteristic dimensionless numbers for the free and the forced convections. The characteristic length to build the Nusselt number and quantify the free convection intensity is the total vertical pipe length (L) while the one representative of the forced convection in a staggered bank is the pipe diameter (D). For this reason, the formula proposed above is no longer well suited as the Nusselt magnitude based on different length scale strongly deviates.

According to some embodiments, the following formula can be used to correlate mixed convection heat transfer for the cooler external heat transfer:

\[ Nu_{cooler} = Nu_{SP} + 0.3 \frac{L}{D} Nu_{DP} \]

The Nusselt number for a free convection vertical boundary layer development is:

\[ Nu_{SP} = 0.13 \frac{C_1 C_2 Re_0^2}{Pr^{1.2}} \]

where the constant has the following values according to the geometry configuration: \( C_1 = 0.416 \), \( C_2 = 0.75 \) and \( m = 0.568 \).

Therefore, the external heat transfer for the cooler corresponds to that of the single pipe in addition to a component representing the pseudo forced convection in the vicinity of the baffle. According to the simulation analysis described supra, in the case of a two-horizontal baffle configuration, only one third of this component is included, as only one third of the total pipe area is affected by this horizontal flow pattern. Other amounts of the component should be included for other configurations and numbers of baffles.

The test results presented in the FIG. 6 were compared with the estimated global HTC obtained using the proposed empirical correlation for the external HTC.

Based on these analyses, a new formula for combined free convection and crossflow for a vertical pipe is proposed, supra. The overall heat transfer coefficient obtained from these correlations matches very well with the experimental data.

This described method enables optimization of heat exchanger design and thus improves the heat exchange performances and reducing the weight of the system for similar performances. According to some embodiments, the heat exchange system includes an inlet and an outlet for a first fluid, increases the velocity and the turbulence intensity of a second fluid flow in neighborhood of the heat exchanger with the first fluid. The heat exchange system enables an increase of the heat transfer with the ambient second fluid. According to some embodiments, the heat exchange surface area is substantially reduced for equivalent performance and thus resulting in a substantial decrease in cooling system footprint.

Forced convection causes, as mentioned, higher heat transfer rates than free convection. In some cases, currents and waves will make the cooler acting as a forced convection cooler. These heat transfer rate changes may be challenging when operating a system while trying to maintain a constant cooling performance. It has been found that according to some embodiments, the influence of sea currents and waves on the heat transfer performance can be decreased. The cooling performances will tend to be independent of the horizontal velocity field from the sea current. This benefit can be very useful on the system regulation point of view.

Although many embodiments have been described supra in the context of a heat exchanger system in which parallel pipes are arranged in symmetrical such as shown in FIG. 2, the techniques described herein are applicable to many other types and arrangements of heat exchangers. FIGS. 10.1, 10.2 and 10.3 illustrate using one or more baffles in different configurations for a non-symmetrical arrangement of cooling system pipes, according to some embodiments. In each case, the heat exchanger 1000 includes an inlet manifold 1002, and outlet manifold 1004 and a plurality of parallel pipes 1020. In FIG. 10-1 a single horizontally arranged baffle 1030 is positioned along the exterior of pipes.
1020 as shown. In FIG. 10-2 a series of four horizontally arranged baffles 1032, 1034, 1036 and 1038 are positioned along the exterior of pipes 1020 as shown. In the case of FIG. 10-3, two baffles 1040 and 1042 are mounted in a slanted arrangement with respect to the pipes 1020 of heat exchanger 1000. In each case the baffles have the effect of disturbing the vertical flow of the ambient coolant fluid (the water) and induce a substantial non-vertical component (i.e. non parallel to the pipe axis) that has been found to substantially increase the cooling performance of the heat exchanger.

[0112] The particulars shown herein are by way of example and for purposes of illustrative discussion of the embodiments of the present invention only and are presented in the cause of providing what is believed to be the most useful and readily understood description of the principles and conceptual aspects of the present invention. In this regard, no attempt is made to show structural details of the present invention in more detail than is necessary for the fundamental understanding of the present invention, the description taken with the drawings making apparent to those skilled in the art how the several forms of the present invention may be embodied in practice. Further, like reference numbers and designations in the various drawings indicated like elements.

[0113] Whereas many alterations and modifications of the present invention will no doubt become apparent to a person of ordinary skill in the art after having read the foregoing description, it is to be understood that the particular embodiments shown and described by way of illustration are in no way intended to be considered limiting.

[0114] It is noted that the foregoing examples have been provided merely for the purpose of explanation and are in no way to be construed as limiting of the present invention. While the present invention has been described with reference to exemplary embodiments, it is understood that the words, which have been used herein, are words of description and illustration, rather than words of limitation. Changes may be made, within the purview of the appended claims, as presently stated and as amended, without departing from the scope and spirit of the present invention in its aspects. Although the present invention has been described herein with reference to particular means, materials and embodiments, the present invention is not intended to be limited to the particulars disclosed herein; rather, the present invention extends to all functionally equivalent structures, methods and uses, such as are within the scope of the appended claims.

1. A heat exchanger system for transferring heat between a first fluid and an ambient second fluid, the system comprising:
an inlet configured to accept the first fluid;
an outlet configured to expel the first fluid;
a plurality of vertically oriented parallel conduits positioned between the inlet and the outlet configured to carry the first fluid therein, the conduits each having an exterior surface that is exposed to an ambient second fluid when the system is submerged in said second fluid, wherein heat is transferred between the first fluid flowing through the conduits and said second fluid flowing a vertical direction along the exterior surfaces of and parallel to the conduits by free convection; and
at least one deflector fixedly mounted exterior to the conduits configured to impart non-vertical momentum in said flowing second fluid thereby enhancing heat transfer between said first fluid and said second fluid.

2. A system according to claim 1 wherein said conduits are tubular pipes grouped into one or more groups, the tubular pipes of each group being arranged symmetrically about a central axis of the group.

3. A system according to claim 2 wherein at least one deflector are two horizontally oriented disks.

4. A system according to claim 2 wherein at least one deflector are three or more horizontally oriented disks for each group of tubular pipes.

5. A system according to claim 4 wherein said three or more horizontally oriented disks are five horizontally oriented disks.

6. A system according to claim 3 wherein said group of pipes and said horizontally oriented disks have a large central opening configured to allow free passage of the second through, and said horizontally oriented disks are configured to force the second fluid into and out of the said central opening of the group of pipes thereby enhancing heat transfer.

7. A system according to claim 2 wherein at least one deflector includes at least one non-horizontally oriented structure that are asymmetric with respect to said central axis of the group.

8. A system according to claim 7 wherein said at least one non-horizontally oriented structure is configured to impart momentum in a tangential direction with respect to said central axis of the group.

9. A system according to claim 8 wherein said at least one non-horizontally oriented structures is helical in shape.

10. A system according to claim 1 wherein said conduits are tubular pipes arranged into a rectangular pattern of columns and rows.

11. A system according to claim 10 wherein said one or more deflectors include at least one horizontally arranged baffle.

12. A system according to claim 10 wherein said one or more deflectors includes at least one horizontally arranged baffle.

13. A system according to claim 1 wherein said second fluid is water.

14. A system according to claim 1 wherein said second fluid is seawater.

15. A system according to claim 1 wherein said system does not use powered equipment to force said second fluid to flow past said conduits.

16. A system according to claim 14 wherein said first fluid includes hydrocarbon gas produced from one or more wells penetrating a subterranean rock formation.

17. A system according to claim 1 wherein the at least one deflector is configured to increase turbulent flow in the second fluid thereby enhancing heat transfer.

18. A system according to claim 1 wherein the heat exchanger is configured to cool said first fluid by transferring heat by free convection from the first fluid to the said second fluid.

19. A method of transferring heat between a first fluid and an ambient second fluid surrounding a plurality of conduits through which the first fluid flows, the method comprising:
exposing exterior surfaces of said plurality of conduits to said surrounding second fluid; and
flowing the first fluid through the plurality of conduits, wherein the plurality of conduits are vertically oriented and parallel to each other, and heat is transferred between the first fluid and said second fluid flowing in a vertical direction along the exterior surfaces of an par-
allel to the conduits by free convection and wherein at least one stationary deflector is fixedly mounted exterior to the conduits configured to impart non-vertical momentum in said flowing second fluid thereby enhancing heat transfer between said first fluid and said second.

20. A method according to claim 19 wherein said conduits are tubular pipes grouped into one or more groups, the tubular pipes of each group being arranged symmetrically about a central axis of the group.

21. A method according to claim 20 wherein said at least one deflector are two horizontally oriented disks.

22. A method according to claim 20 wherein said at least one deflector are five horizontally oriented disks.

23. A method according to claim 21 wherein said group of pipes and said horizontally oriented disks have a large central opening configured to allow free passage of the second fluid therethrough, and said horizontally oriented disks are configured to force the second fluid into and out of the said central opening of the group of pipes thereby enhancing heat transfer.

24. A method according to claim 20 wherein said at least one deflector includes at least one non-horizontally oriented structure that is asymmetric with respect to said central axis of the group, said non-horizontally oriented structure being configured to impart momentum in a tangential direction with respect to said central axis of the group.

25. A method according to claim 24 wherein said at least one non-horizontally oriented structures is helical in shape.

26. A method according to claim 19 wherein said second fluid is seawater.

27. A method according to claim 19 wherein said first fluid includes hydrocarbon gas produced from one or more wellbores penetrating a subterranean rock formation.

28. A method according to claim 19 wherein said flowing includes pumping the first fluid through said plurality of pipes.

29. A method according to claim 19 wherein heat is transferred from said first fluid to said second fluid thereby cooling said first fluid.

30. A method according to claim 19 wherein heat is transferred from said second fluid to said first fluid thereby heating said first fluid.

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