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Use of sophisticated heat exchanger simulation models for investigation of possible design and operational pitfalls in LNG processes

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ABSTRACT

The simulation rating programs S-FIN for PFHE and S-PLATE for PHE have been developed at SINTEF Energy Research. These tools can be incorporated in process simulation environments like PRO/II and Aspen HYSYS®, and thus be used as an integrated part when doing process energy simulation and optimization.

Static flow instabilities that can occur in heat exchangers used in cryogenic services are discussed.

Examples on how to perform, and how to interpret, a Ledinegg instability analysis, are shown using the developed programs.

With the well-known single mixed refrigerant process as a case study, a thermally valid plate-fin heat exchanger was designed that was subjected to Ledinegg instability. Remedies to avoid this and the effect on the process energy consumption are discussed. For the selected case, the compressor power increased by 14% going from an unstable to a stable design/operation.

The examples show that detailed heat exchanger simulations should be performed as a part of process optimization.

1. Introduction

There is an increasing demand for compact heat exchanger equipment, for example within floating LNG in order to keep weight, size, volume and cost at a minimum.

Common for compact heat exchangers is high surface/volume ratio, small footprint and efficient heat transfer capabilities. Plate-fin heat exchangers (PFHE), brazed plate heat exchangers (PHE) and more sophisticated designs like Printed Circuit type heat exchangers are considered used.

Commercial process simulation programs (e.g. Aspen HYSYS®, PRO/II) treat process units like heat exchangers in a simplified manner often using lumped warm and cold composite streams, constant heat transfer values and the assumption that the required cooling/heating is provided. The unit itself is not modeled in detail due to computational time and the added complexity. Therefore process design/optimization and heat exchanger design are often decoupled.

During process energy optimization, minimum temperature differences in heat exchangers are often used as one of the main constraints. Using this criterion alone may result in several unintended operational consequences. Actual heat exchanger designs also have to consider the risk of static instability phenomena like the Ledinegg instability that may occur (especially) in vaporizing services. Heat exchanger units susceptible to such type of instability behaviour need to be investigated in order to avoid potential operational problems, both at design and at off-design conditions. This will be an extra constraint in the energy optimization analysis.

With a detailed heat exchanger model a chosen design can be validated and the process operability can be investigated.

2. Flow instabilities

Flow instability in two-phase flow has been a large research topic over many years. The largest area of research in this field is probably safety aspects of nuclear reactors and flow instabilities in natural circulation boiling loop Nayak and Vijayan (2008). Many of the instability phenomena described in Nayak and Vijayan (2008) can also be present in cryogenic processes and especially in boiling services which will be discussed later. Recent review article by Kakac and Bon (2008) and Tadrist (2007) discuss various categories of flow instabilities.

Instabilities have been categorized by several authors, and an excerpt from a map by Durga Prasad et al. (2007) is shown in Fig. 1.

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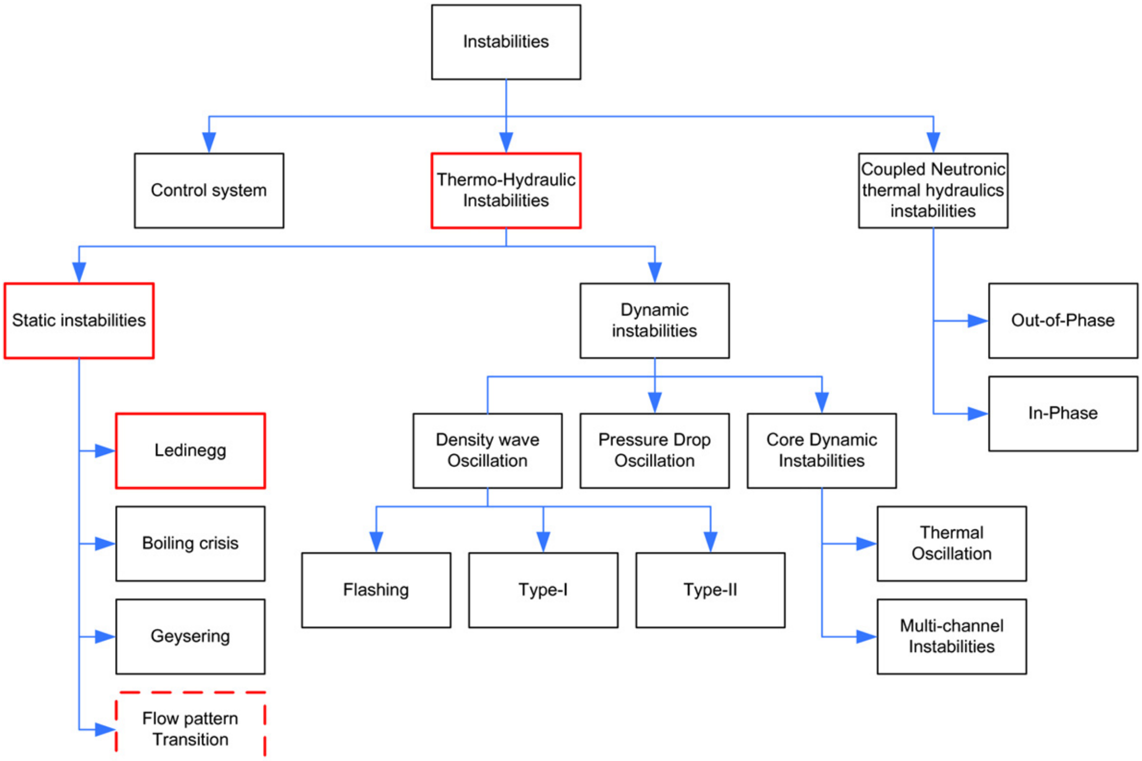

Fig. 1. Instability map by Durga Prasad et al. (2007).

In the branch covering the "Thermo-Hydraulic Instabilities" the two main categories are "static" and "dynamic" instabilities. Dynamic instabilities cover different types of oscillations. Static instabilities are characterized by a system operating at steady state, but when a small disturbance (e.g. flow excursion) occurs the operating point may "jump" to a completely different steady state. The focus of this study is on a sub-class of static instability – the Ledinegg instability.

The Ledinegg instability is not a new observation but is seldom discussed in the open literature regarding design and optimization of LNG heat exchangers. Possibilities of static flow instabilities in parallel heat exchanger networks are briefly discussed by Rolland et al. (2003) and by Sotzek (1999). In spite of this, flow excursion instabilities have been observed in commercial plants that have been built Paradowski (personal communication).

2.1. Ledinegg instability – background theory

Ledinegg (1938) described in his article how natural and forced convective boiling in (vertical) parallel tubes could lead to unstable flow. He showed that depending on heat flux and degrees of subcooling, the total pressure drop vs. flowrate would exhibit different shapes as shown in Fig. 2 to the right. Curve "a" in Fig. 2 has the highest inlet temperature (least subcooled) and shows a monotone increasing pressure drop with increasing flow. This is a stable flow situation. Curve "b" has lower inlet temperature than curve "a" and is on the limit to become unstable, while curve "c" has the lowest inlet temperature of the three and shows an unstable flow situation. This is characterized as the pressure drop passes a maximum and a minimum value as the flowrate increases. As Ledinegg explains: For a given total pressure drop between the inlet and outlet manifold, this system can have three different flowrates. An oscillation between the three is also possible. Ledinegg described this for a water/steam system and discussed how the various contributions from friction, gravitational and acceleration pressure drop determine the shape for these curves. This behaviour has been demonstrated by several authors for different fluids and geometries as shown in the review article by Kakac and Bon (2008).

2.2. Ledinegg instability in heat exchangers

An *N*-shape with decrease in total pressure drop for increasing flowrate in boiling services as shown in Fig. 2 can mainly be the result of two counteracting effects:

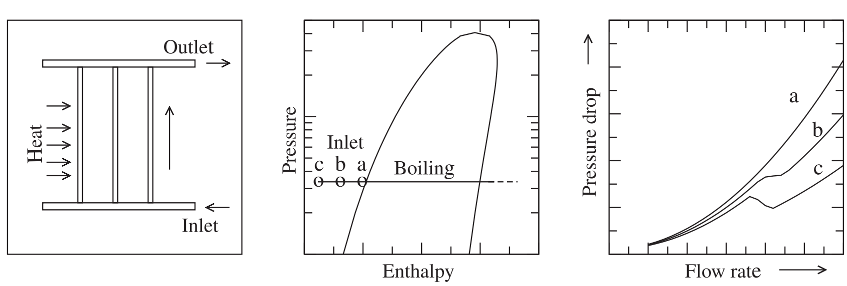

Fig. 2. Definition of Ledinegg instability Ledinegg (1938).

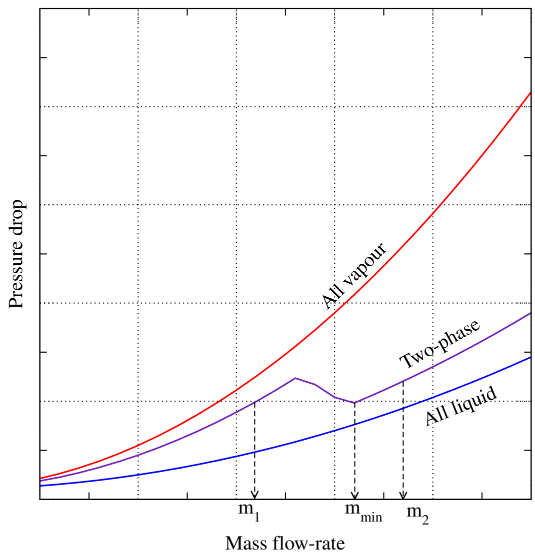

Fig. 3. Ledinegg instability in boiling services.

- Increase in flowrate → higher pressure drop
- Decrease in average void fraction → lower pressure drop

In combination, the increased flowrate will also give a decrease in average void and thus give an *N*-shape. This can be illustrated as in Fig. 3.

For this illustration, horizontal flow with no gravitational pressure drop contribution is used. The upper curve in Fig. 3 is the pressure drop for increasing flowrate for an all vapour flow (infinite heat load where the whole flow evaporizes at the inlet) while the lower curve is the total pressure drop for all liquid flow (no heat load). For a specific heat load the pressure drop at low flowrates will be close to the "all-vapour"-curve since the flow will be fully evaporated after a short distance into the heat exchanger. As the flowrate increases, the total pressure drop will move closer to the "all-liquid"-curve. Whether an *N*-shape will occur or not, depends on the geometry, fluid, inlet condition and heat load (profile). In this illustration, flowrates below m_1 and above m_2 will be in a stable region. Flowrates between m_1 and m_2 are considered unstable. Not shown on this graph is the outlet vapour fraction (shown on Fig. 10). This will often be close to 1.0 in the region between m_1 and m_2 and decrease almost linearly for increasing flowrates. In a practical situation, only flowrates below m_1 would therefore be acceptable, otherwise the outlet condition after evaporation would be in the two-phase region. There will also be a lower limit for acceptable flowrate for avoiding dynamic instabilities.

2.3. Consequences of Ledinegg instability in LNG heat exchangers

An LNG heat exchanger often has several parallel channels (Plate or Plate-Fin types) and often consists of several parallel heat exchanger blocks that are welded together to a heat exchanger assembly. These types of heat exchangers have a built-in parallelism so a Ledinegg instability analysis should be conducted in the system and heat exchanger design phase. If the LNG process is using a hydrocarbon mixture as refrigerant, the temperature glide between bubble and dew-point can be quite substantial, so maldistribution of refrigerant in different parts of a heat exchanger block or between blocks will result in large (local) temperature differences and a reduction in heat exchanger performance and possibly durability.

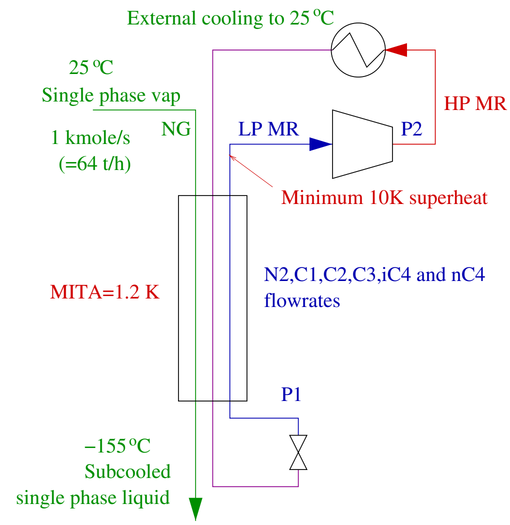

Fig. 4. A simple flow-sheet for the current case study.

If the total flowrate in a parallel heat exchanger system lies in the range between the flowrates indicated as m_1 and $m_{\min}$ in Fig. 3, it is very likely that this will be distributed as a high and low mass flux in various parts of the heat exchanger or of the assembly, driven toward the lowest total pressure drop possible. With uneven heat load distribution, several *N*-shapes could appear, giving an even higher number of possible solutions for the flowrate than discussed by Ledinegg.

To be able to do a Ledinegg instability analysis a realistic heat exchanger model must be used, a model that is physically correct regarding the relation between geometry, pressure drop, heat transfer and void fraction.

3. Ledinegg instability analysis for an optimized single mixed refrigerant process

In the following, a demonstration on how to do a Ledinegg instability analysis will be shown for an optimized Single Mixed Refrigerant process.

Based on the optimized process, the main heat exchanger is analyzed using in-house simulation software for rating of Plate-Fin heat exchangers called S-FIN, integrated as a user added subroutine in PRO/II from SimSci-Esscor.[1]

3.1. Description of the case study

The selected case study is the familiar Single Mixed Refrigerant Process (SMR) often found subjected to optimization in the literature e.g. Jensen and Skogestad (2006).

The process flow-sheet is shown in Fig. 4. 1.0 kmol/s (=64 ton/hr) of Natural Gas (NG) is cooled, condensed and subcooled from 25 °C to −155 °C at 55 bar using a single mixed refrigerant as the cooling fluid.

The specifications were external cooling and partly condensation of the high pressure mixed refrigerant (HPMR) to 25 °C. The composition (mole%) for the lean natural gas was taken from Jensen and Skogestad (2006) and normalized to: Nitrogen (N2/2.8), methane (C1/89.8), ethane (C2/5.5), propane (C3/1.8) and n-butane (nC4/0.1) The components in the mixed refrigerant consists of nitrogen, methane, ethane, propane, *i*-butane and *n*-butane.

[1] A User Added model of S-FIN also exists for HYSYS, but has not yet been developed to the same level as the PRO/II version.

Constant pressure drop for each stream in the main LNG heat exchanger was applied. The values used were 5 bar for the natural gas, 4 bar for the high pressure refrigerant, and 1 bar for the low pressure refrigerant.

The SMR process in this case is run with a single compressor stage since the number of compressor stages only affects the power consumption and not the behaviour of the main cryogenic heat exchanger which is the main focus here.

3.2. Description of S-FIN

S-FIN is a steady state rating tool for Plate-Fin heat exchangers. It requires detailed user-defined geometry, characterized by fin-types and geometry, layer information and stream layout for cold and warm streams. Its intended use is for cooling (condensation) and heating (boiling) of hydrocarbon mixtures. Calculations of heat transfer and pressure drop gradients are done locally in a fixed grid for each stream and integrated to give a total capacity and pressure drop. Heat transfer and frictional factor correlations for various fin types like plain, serrated or perforated are taken from Hesselgreaves (2001). The heat transfer equations are solved simultaneously.

As illustrated in Fig. 5, heat is only transfered in the active zone between the inlet and outlet distribution sections. The inlet and outlet distribution sections and nozzles are represented in S-FIN as an equivalent length having the same fin- type and geometry as for the corresponding stream and a static height. The equivalent lengths are used when calculating the frictional pressure drop, while the static height is used for the gravitational contribution. If measurements or a more detailed distribution section model exist, the equivalent lengths can be "calibrated" against this. By using different equivalent lengths in two or more parallel heat exchangers, this method can also be used to simulate imperfections in the geometry.

In the active zone, each stream can have individual entry and exit locations so the actual number of active streams at a specific location can vary. A simplification, though, is that a common wall temperature is used, meaning that the effect of having different stacking patterns is not taken into account by the program.

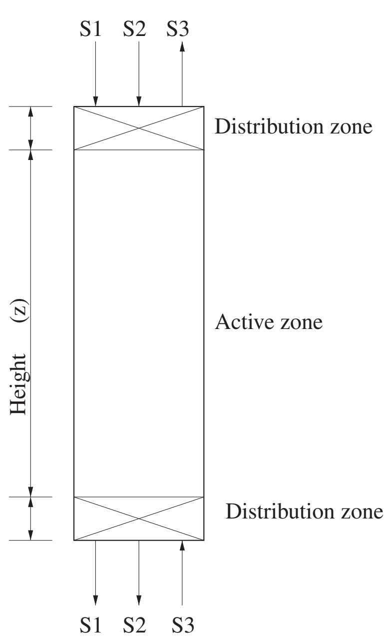

Fig. 5. Principle illustration of S-FIN.

S-FIN is compiled to a Dynamic Link Library (dll) for use in process simulators like HYSYS and PRO/II where several heat exchanger models can be used simultaneously.

3.3. Description of the optimization tool-chain

For optimization, HYSYS v 7.1 has been used in combination with the NLPQLP routine by Schittkowski (2006) and in-house code.

The flow-sheet is evaluated with HYSYS using the included LNG heat exchanger unit model, so no detailed heat exchanger model is used during optimization.

Only standard unit models are used, so the LNG heat exchanger is based on enthalpy differences for the composite curves. No detailed heat exchanger model is used during the optimization. The outlet temperature of the natural gas and the high pressure refrigerant is set to the specified -155 °C while the outlet temperature of the low pressure refrigerant is then calculated by an enthalpy balance for the inlet and outlet streams. The temperature differences inside the heat exchanger are based on the cold and the warm composite curve.

NLPQLP v. 2.21 Schittkowski (2006) is used for the optimization. Some additional code has been added in order to calculate the derivatives and to compile it as a Dynamic Link Library (dll). The in-house code is used for the communication between HYSYS and NLPQLP.

The optimization variables are the molar flows of the mixed refrigerant and the pressures after the expansion valve and after the compressor in Fig. 4.

The objective of the optimization is the compressor power and the restrictions are the minimum temperature approach (MITA) for the LNG heat exchanger (minimum 1.2 K) and the superheating (minimum 10 K) of the feed stream to the compressor.

3.4. Ledinegg instability analysis

The S-FIN program cannot predict the Ledinegg instability in a single simulation. By that, meaning the conditions in one heat exchanger that provide all the possible solutions for mass flowrate for a given pressure drop. The analysis has been done by varying the low pressure refrigerant flowrate and plots the pressure drop vs. flowrate curves. If the characteristic *N*-shaped curve occurs, this heat exchanger can be subjected to Ledinegg instability. This will also reveal whether the actual flowrate is in the "unstable region" or not. When doing an analysis for an actual heat exchanger, two heat exchangers are used in parallel as shown in Fig. 6.

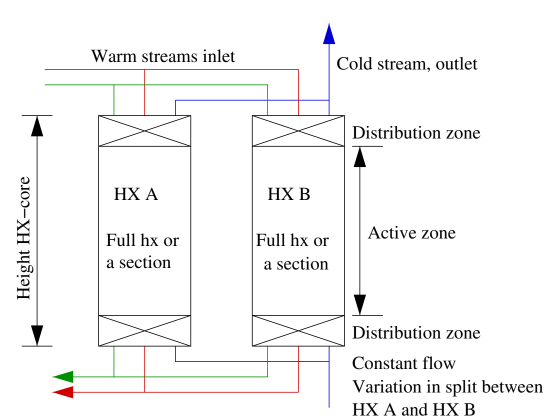

Fig. 6. S-FIN used in the Ledinegg analysis.

Now the total flowrate for the low pressure refrigerant stream is kept constant, at twice design flowrate, while the split factor for flow between HX-A and HX-B is varied between, say from 0.1 to 0.5. When the mass flowrate vs. pressure drop for the two modules are plotted, HX-A will have the relative flowrate from 0.1 to 0.5 while HX-B will have the relative flowrate from 0.5 to 0.9. This will cover the 20–180% of the nominal flowrate. Geometry splits other than 50/50 is of course also possible, meaning that the two heat exchangers can have a different number of layers (1000/200 instead of 600/600 as an example).

One advantage by dividing the heat exchanger in parallel modules is to also use the process simulator's ability to re-distribute flowrate for the warm streams after control criteria like equal pressure drop for each low pressure side flow maldistribution. This is indicated in Fig. 7 where a multivariable controller can be used on the warm streams S2 and S3 to obtain equal outlet pressures $p_{A2} = p_{B2}$ and $p_{A3} = p_{B3}$ In operation, the total pressure drop over these two heat exchangers has to be equal.

The Ledinegg analysis was conducted along the following three steps.

A Start with the optimized process parameters from the flow-sheet optimization using simple heat exchanger models and use of composite stream
B Design a heat exchanger according to the optimized result. At this stage, it may be necessary to alter the process parameters in order to avoid temperature crossings when moving from composite streams to individual streams that calculates local pressure drop.
C Analyze the chosen heat exchanger design from step B in terms of operability. If susceptible to Ledinegg instability, a change of heat exchanger and/or process design conditions will be necessary.

3.4.1. Optimized solution (step A)

Based on the optimization principle described earlier, the result from the optimization on pressure levels and refrigerant composition is shown in Table 1.

As seen, the refrigerant molar flowrate is 3.15 times the natural gas molar flowrate and the temperature difference over the *J*/*T* valve is 2.1 K. The corresponding composite temperature profile for the heat exchanger is shown in Fig. 8.

When re-calculated using PRO/II, the compressor work was calculated to be 18.91 MW for this optimized solution. This will correspond to an energy consumption of 296 kWh/ton LNG.

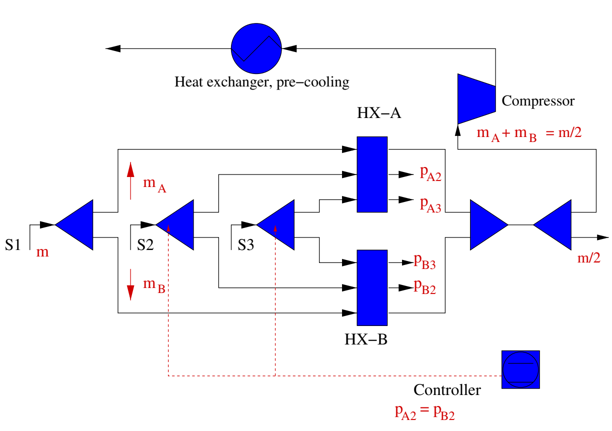

Fig. 7. Ledinegg analysis with S-FIN in a PRO/II environment.

Table 1
Result from the optimization of the SMR process using HYSYS and NLPQLP (re-calculated with PRO/II for the later comparison purpose).

	NG	Refr HP	Refr LP
Inlet temperature (°C)	25	25	−157.1
Outlet temperature (°C)	−155	−155	23.5
Inlet pressure (bar)	55.0	25.88	5.34
Outlet pressure (bar)	50.0	21.88	4.34
Flowrate (kmole/s)	1.0	3.15	3.15
Composition (mole %)			
N2	2.8	9.29	9.29
C1	89.8	29.13	29.13
C2	5.5	38.87	38.87
C3	1.8	–	–
iC4	–	–	–
nC4	0.1	22.71	22.71
Compressor power (MW)	18.9		

3.4.2. Heat exchanger design and validation (step B)

For the actual heat exchanger the following design criteria and constraints were used. The heat exchanger had a vertical orientation and should consist of only one module in the flow orientation – no splitting and mixing of two-phase flow between blocks. The maximum height of the heat exchanger core was set to 6 m which is about the limitation of the brazing furnaces. Otherwise a compact design is desired.

In this example, the refrigerant composition from step A has been kept. The warm streams were run from top to bottom while the cold stream run upward. Gravitational terms for the pressure drop were included.

To avoid unphysical solutions like temperature cross-over when moving from composite to individual warm streams, and to cool the natural gas to the specified −155 °C, the suction pressure has to be lowered to 5.0 bar. Also the discharge pressure was increased from the optimal 25.88 bar to 28.5 bar and the refrigerant flowrate increased to 3.19 kmol/s.

The main results from the selected heat exchanger design are shown in Table 2 and the corresponding temperature profiles in Fig. 9. The minimum temperature approach for this solution is higher than the 1.2 K used in the optimization. With "fine-tuning" of geometry composition and process conditions or re-optimizing

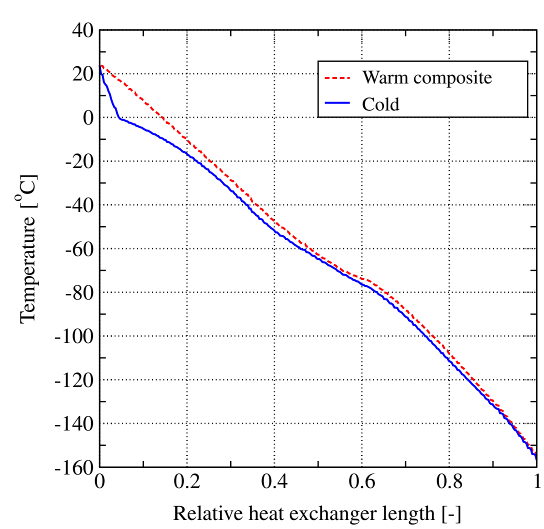

Fig. 8. Warm and cold stream composite temperature profile after optimization with HYSYS.

Table 2
Result from S-FIN/PRO/II calculation with an actual heat exchanger.

	NG	Refr HP	Refr LP
Inlet temperature (°C)	25	25	−158.4
Outlet temperature (°C)	−155.4	−153.8	6.1
Inlet pressure (bar)	55.0	28.5	5.0
Outlet pressure (bar)	54.6	27.1	4.0
Flowrate (kmole/s)	1.0	3.19	3.19
Main heat exchanger data			
Heat transfer surface (m^2)	27106		
Core volume (m^3)	28.5		
Compressor power consumption (MW)	19.8		

with a complex heat exchanger model included, a better result could probably be found.

Process simulation, with PRO/II, using this heat exchanger, that operates with lower evaporating pressure and higher refrigerant condensing pressure, shows that the compressor work increased from 18.9 to 19.8 MW compared to the solution in step A.

3.4.3. Heat exchanger operability validation (step C)

With the heat exchanger design from step B, a Ledinegg analysis has been carried out using the approach outlined in Section 3.4.

When the low pressure refrigerant flowrate is varied in the model, due to different distribution of the total flowrate, this can be considered as representing a range of various massfluxes locally inside a heat exchanger. By generating a mass flow vs. pressure drop curve for a stream, this will show if this stream is subjected to steady state instabilities or not for the current design/operating flowrate.

When varying the low pressure refrigerant flowrate for the selected single mixed refrigerant case, the pressure drop vs. mass flowrate curve shows a characteristic *N*-shape. Simulations using different choices of pressure drop and heat transfer correlations all show an *N*-shape but with differences in the actual pressure drop values and thus the magnitude of the “peak” and “valley” of the curve. Fig. 10 shows how refrigerant flowrate from the optimization step A lies in the negative slope on the curve. This is in an unstable operating point and should be avoided. With parallel heat exchanger blocks, or with several parallel channels like in a plate-fin heat exchanger, it is expected that some blocks, or channels will operate with a high mass flux - be over-refrigerated, while others

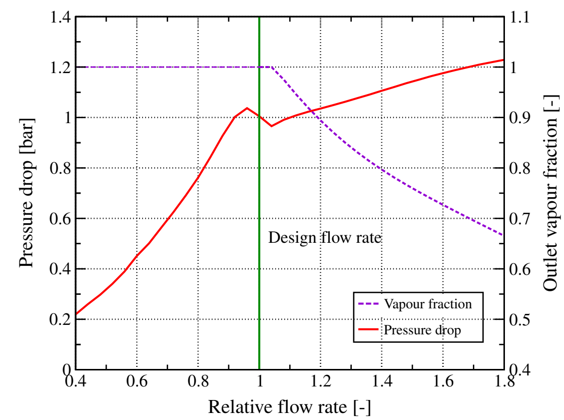

Fig. 10. Evaluation of possible *N*-shape for the pressure drop vs. relative flowrate for the optimized design.

will operate with a low mass flux - be under-refrigerated. The total flowrate will still be the design flowrate. This maldistribution can also occur inside one layer. In Fig. 10, the corresponding outlet vapour fraction after the evaporation of the low pressure refrigerant is shown. At the design flowrate the refrigerant outlet is in the superheated state with a vapour fraction of 1.0. If the relative flowrate is more than 5% higher than design, the refrigerant outlet will move into the two-phase region.

For vertical evaporation there will also be a lower limit for an acceptable flowrate. At too low massfluxes dynamic instabilities with oscillations and reversed flow can occur as discussed by Müller-Menzel and Hecht (1995) and Müller-Menzel and Hecht (1994). This phenomenon has also been discussed by Fu et al. (2008) from experiments for boiling nitrogen in narrow vertical channels.

In Figs. 11–13, results from a simulation of the heat exchanger with a 42/58% relative mass flow distribution between the “under”- and “over”-refrigerated heat exchanger are discussed. This is not directly derived from the *N*-shape curve, because the geometry split between the two modules will not be 50/50, but is shown as an example of a possible “extreme” situation.

The total pressure drop between the inlet and outlet of the heat exchangers are equal, but they follow different profiles due to different local contributions of friction, acceleration and gravitational pressure drop.

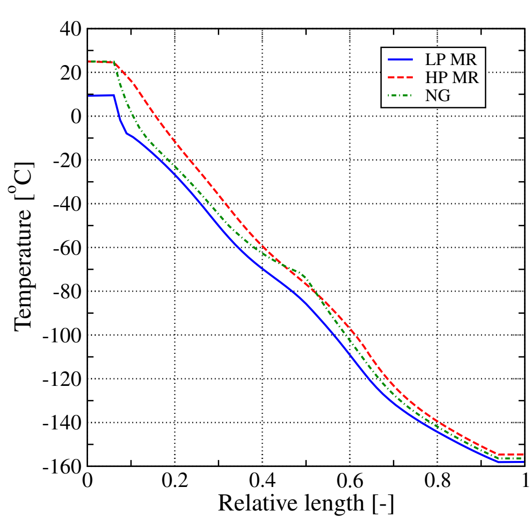

Fig. 9. Design and rating of a plate-fin heat exchanger to comply with the optimized solution from A. Temperature profile for the cold and warm streams.

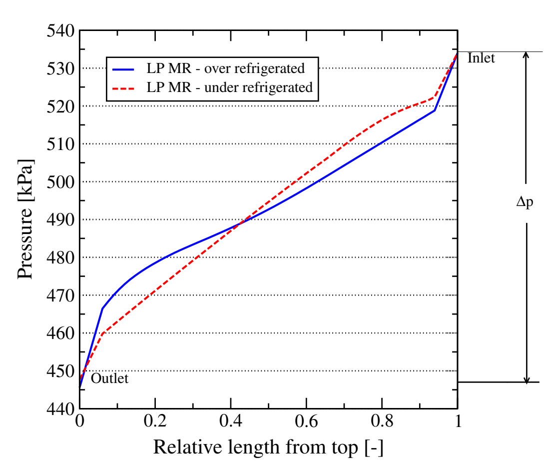

Fig. 11. Pressure profiles for an over-refrigerated and under-refrigerated section of the optimized heat exchanger.

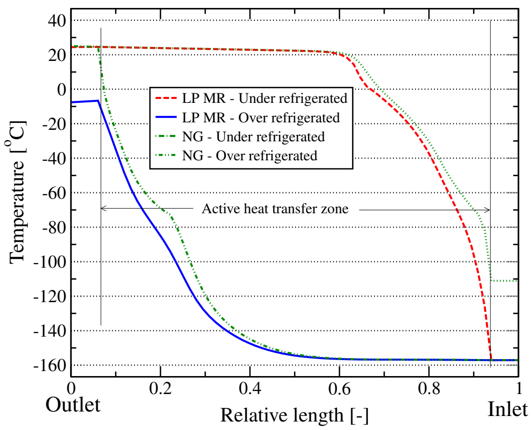

Fig. 12. Temperature profiles for an over-refrigerated and under-refrigerated section of the optimized heat exchanger.

In a plate-fin heat exchanger it is quite common to use serrated or offset strip fins. These are "open" fin structures that may allow vapour to also flow perpendicular to the main flow direction. When looking at the pressure profile from Fig. 11, one can anticipate an amplifying effect of the maldistribution in the cold end of the heat exchanger and a stabilizing effect in the warm end where vapour may "leak" toward the lowest local pressure. This effect is not taken into account in these examples.

The corresponding temperature profile for the two parallel heat exchangers is shown in Fig. 12. Due to the large temperature glide of the refrigerant mixture, the effects from maldistribution of the refrigerant are quite large. This will also be the case for the metal temperatures. If an unstable situation causes oscillations between two steady state operating conditions inside the heat exchanger core, this would mean large metal temperature cyclings that must be avoided in a plate-fin heat exchanger ALPEMA (1994).

The effect on the performance of a heat exchanger that operates as a "mix" of over- and under-refrigerated mode is also shown in Fig. 12. The outlet temperature for the natural gas in the under-refrigerated case reaches about −112° while for the over-refrigerated case it reaches the low pressure refrigerant temperature of −156°. After mixing the temperature is −147 °C, 8° higher than the design requirement. The outlet refrigerant superheat has now been reduced to about 10 K, and compressor power in this model increased from 19.8 to 20.6 MW.

Finally, in Fig. 13 the vapour fraction profiles for the two parallel heat exchangers are shown. Here the over-refrigerated heat exchanger show an exit vapour quality just below 0.8 while the under-refrigerated heat exchanger is superheated the upper 2/3rd the vapour temperature reaches the warm stream temperatures of 25 °C. After mixing, the state is in the superheated region, but in a practical situation it can be anticipated that the vapour flow will contain entrained liquid droplets.

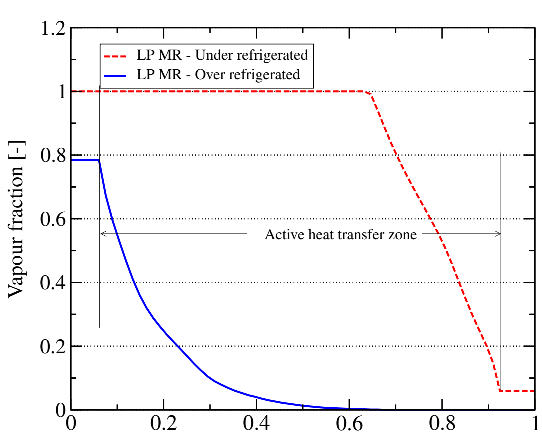

Fig. 13. Vapour fraction profiles for an over-refrigerated and under-refrigerated section of the optimized heat exchanger.

Table 3
Result from S-FIN/PRO/II calculation with increased inlet flow restriction.

	NG	Refr HP	Refr LP
Inlet temperature (°C)	25	25	−156.0
Outlet temperature (°C)	−155.0	−153.0	8.6
Inlet pressure (bar)	55.0	28.5	5.55
Outlet pressure (bar)	54.5	27.2	3.55
Flowrate (kmole/s)	1.0	3.19	3.19
Main heat exchanger data			
Heat transfer surface (m^2)	27106		
Core volume (m^3)	28.5		
Compressor power consumption (MW)	21.63		

4. How to avoid the *N*-shape?

The *N*-shape seen in Fig. 10, show that there is a possibility that a heat exchanger that is designed based on the optimum process operating conditions could be susceptible to Ledinegg instability. Remedies for avoiding Ledinegg instability, are according to Taylor (1987) to:

- increase the inlet flow resistance
- decrease the outlet flow resistance
- increase the cold end temperature difference
- re-design of the heat exchanger

Most of the above steps will necessarily increase the plant energy consumption. Already in the design phase, a heat exchanger should be designed in such a way that, for the design operating condition no *N*-shape of the pressure drop vs. flowrate should occur at all.

In the next section, the effect of increasing the inlet flow restriction and a complete heat exchanger re-design will be shown.

4.1. Increase the inlet flow resistance

When increasing the inlet restriction, the whole *N*-shaped curve will be tilted to a higher total pressure drop, increasingly with

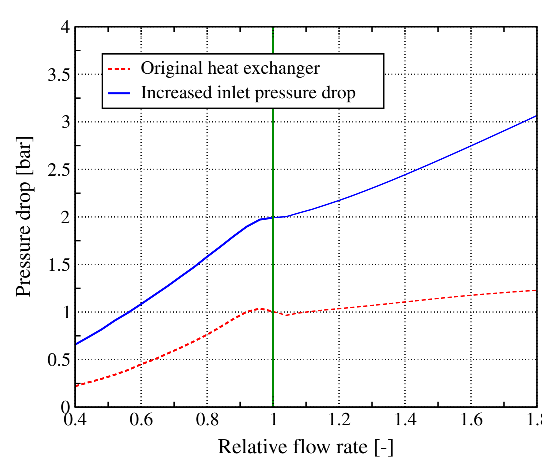

Fig. 14. Pressure drop vs. flowrate for the heat exchanger with increased inlet pressure drop.

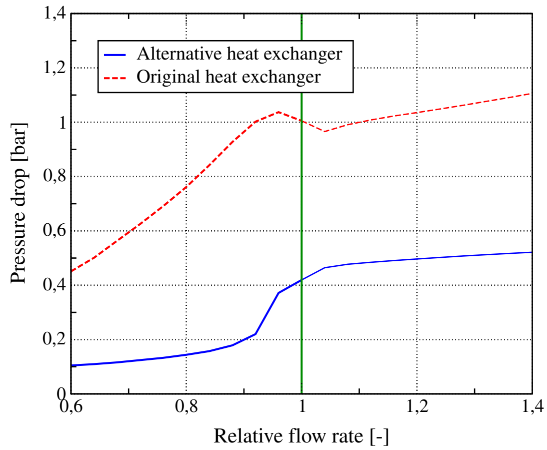

Fig. 15. Evaluation of possible *N*-shape for the pressure drop vs. relative flowrate for the re-designed heat exchanger.

increasing flowrate. This might smoothen out the *N*. The penalty is increased compressor power.

With the heat exchanger from step B, an additional inlet pressure drop of about 1 bar was required to smoothen out the *N*-shape in this case. This doubled the total low pressure refrigerant pressure drop. It was possible to increase the inlet pressure for the low pressure refrigerant to 5.55 bar prior to the inlet restriction. This will reduce the compressor power consumption penalty. The result from the calculations is shown in Table 3. The compressor power increased from 19.8 to 21.6 MW compared to the calculation in the previous step B.

As seen from the graph in Fig. 14 the *N*-shape has been removed, but with almost 10% increase in compressor power.

4.2. Re-design of heat exchanger

In the second example the heat exchanger is re-design in order to smoothen out the *N*-shape.

An example of the pressure drop vs. flowrate for a re-designed heat exchanger is shown in Fig. 15. The basic difference between the original heat exchanger and the re-designed heat exchanger is that the latter one has been designed with a larger heat transfer surface with more parallel channels. This will reduce the heat flux on the boiling flow and dampen the amplitude in an *N*-shape. Since the pressure drop now is considerable lower, the suction pressure in this case can be kept at the "optimum" 5.34 bar from step A and still avoid warm and cold temperatures crossing.

As seen from Table 4 and Fig. 15 the pressure drop over the re-designed heat exchanger is only 40% of the accepted value of 1.0 bar, with no *N*-shape on the pressure drop vs. flowrate curve.

Table 4
Result from S-FIN/PRO/II calculation for the re-designed heat exchanger.

	NG	Refr HP	Refr LP
Inlet temperature (°C)	25	25	−157
Outlet temperature (°C)	−156.8	−156.8	9.80
Inlet pressure (bar)	55.0	25.9	5.34
Outlet pressure (bar)	55.0	25.7	4.93
Flowrate (kmole/s)	1.0	3.19	3.19
Main heat exchanger data			
Heat transfer surface (m^2)	91129		
Core volume (m^3)	111.4		
Compressor power consumption (MW)	17.6		

The penalty here is a heat exchanger 3 times larger compared to the original design.

5. Conclusion

Flow instabilities in heat exchangers have been a large research area for many years. Ledinegg instability is a class of static instability describing a situation where multiple flowrates can result in equal pressure drop in a channel.

According to Taylor (1987), cryogenic systems using fluids with low density and low latent heat, flow instabilities are more likely to occur than for other type of systems.

The risk of ending up with a design operating point in an unstable region is high, if processes are optimized based on composite curves and minimum temperature differences. The pressure drop vs. flowrate curve can exhibit an *N*-shape around the design flowrate.

When equipment with a high degree of parallelism is used, this could have negative consequences on both operability and energy efficiency.

In the analysis of an optimized Single Mixed Refrigerant process using plate-fin heat exchanger a valid design in terms of meeting process specifications and avoiding temperature cross-over was found, using a detailed heat exchanger model. The pressure levels and refrigerant composition and flowrate were derived from the optimized solution where a composite warm temperature curve was used.

This heat exchanger design exhibited an *N*-shape for the pressure drop vs. flowrate curve and the design operating point was located in the unstable region on the *N*-curve. Two remedies for removing the *N*-shape were done:

1. The inlet pressure drop for the low pressure refrigerant was applied with the penalty of increased compressor power.
2. The heat exchanger was re-designed with larger surface to reduce the heat load on the evaporating stream.

These examples show that during optimization of LNG processes, a realistic heat exchanger model need to be used and constraints reflecting the remedies for avoiding Ledinegg instabilities should be included in the model.

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