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High Temperature Compact Heat Exchangers: Performance of Advanced Metallic Recuperators for Power Plants

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HIGH TEMPERATURE COMPACT HEAT EXCHANGERS: PERFORMANCE OF ADVANCED METALLIC RECUPERATORS FOR POWER PLANTS

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ABSTRACT

An overview is given of recent high temperature compact heat exchangers technology developments, both in the thermo-fluid dynamic innovative solutions and in the materials. Gas turbine recuperative cycles and microturbine systems are the investigated fields where this kind of heat exchangers is used. Some common technological features join all of them, namely: new surface geometry design and burdensome material requirements.

A sizing procedure is developed for a plate-fin type and an innovative printed circuit heat exchangers to be used in a gas turbine cycle. Starting from this calculation, the most important difficulties arising from the specific conditions of each heat exchange system are pointed out, together with their possible solutions. A comparison among them is presented and some suggestions are proposed about expanding the boundaries of the individual research fields, to get a wider understanding of common critical issues.

Finally, the paper focuses on the main current challenges to be faced in the near future in order to get a competitive development of such systems.

1. INTRODUCTION

High temperature heat exchangers (arbitrarily meaning temperatures above 650°C) are required in many industrial systems for process reasons or aiming to attain high efficiency. Latter one is the case of power plants. Among these, the most promising in terms of development chances are:
- Recuperative gas turbine and combined cycles
- Recuperative microturbine systems

Each one of these systems has to comply with specific requirements, related to its function.

In recuperative gas turbine cycles the feasibility of high temperature heat exchangers should be analysed more deeply because of the possibility to reach very high thermal efficiency, up to 46% in simple cycles and 62% in combined ones (GE Power Systems, 2003, Franco and Casarosa, 2002).

The main drawback for the application of recuperative heat exchangers in such cycles lies in the high volumetric flow ratio between the streams, due to the high pressure ratio, generally between 12 and 30, combined with an almost equal massive flow on both sides of the exchanger. In the following the consequences of such different volumetric stream flows will be shown.

Microturbines are a relatively new sort of distributed generation technology being used for stationary energy generation applications. There are no established guidelines to categorize small gas turbines, it is arbitrarily assumed that machines in the 50–500 kW range are classed as microturbines. For simple cycle microturbines, it is difficult to get a thermal efficiency much in excess of 20% based on today's technology, because of the small pressure ratios, thus an exhaust heat recovery exchanger is mandatory to achieve efficiencies values of 30% and higher (McDonald, 2000).

In developing heat exchangers for microturbine systems it should be considered that their application has to comply with space and layout restrictions that markedly affect even the conceptual design of the recuperator. In addition, they have to handle the constant temperature differentials across the surface area, and match an operating life of several thousands hours with hundreds of start-up/stop cycles. It is also very important to balance the need for higher performance and more durable materials with the need to make them as cost-effective as possible.

2. HEAT EXCHANGER CURRENT EQUIPMENT

The industrial systems that have been briefly reviewed can be analysed on the basis of their main technological

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issue. Two major research areas are identified. The first one is related to the development of heat transfer surface geometries, the second to the material issue (namely: high temperature mechanical properties, creep properties, and environmental resistance).

Starting from gas turbine cycles, largely different flow areas on the sides need to be accommodated. A numerical example is reported below to better explain such a problem.

A reference gas turbine combined cycle is considered, whose typical data are listed in Table 1. A thermo-fluid dynamic sizing calculation has been performed, showing the aforementioned trickiness of arranging the two flow streams. Two compact heat exchangers have been selected for this purpose, a printed circuit heat exchanger (PCHE) and a plate-fin (PFHE).

Tab. 1 - Typical reference data for a heat recovery system in a combined gas turbine cycle

Streams mass flow (kg/s)	100
Lower cycle pressure (Pa)	$1.08 \cdot 10^5$
Streams pressure ratio	24.3
Inlet air temperature (°C)	521
Inlet gas temperature (°C)	825
Maximum $\Delta p/p$ allowable (%)	5 - 5 (gas - air)
Thermal duty (W)	$31 \cdot 10^6$
Required effectiveness, min.	0.91

2.1. Sizing high temperature gas-gas heat exchangers

A "sizing" problem means the design of a heat exchanger, selecting an exchanger type, its flow arrangement, the materials and the physical size, in order to meet the specified requirements on heat transfer and pressure drops. When the first three are selected, the problem reduces to the determination of the physical size, e.g. length, width, height and surface areas on each side. In order to get a very high value of efficiency only the counter-current flow configuration and the multipass one should be considered. The single-pass crossflow configuration is not a valid choice, since the heat exchanger would work in such a way that a very large part of it doesn't actually contribute to the overall thermal exchange, because of the small difference between the stream temperatures.

The PCHE is a rather novel heat exchanger type, formed by diffusion bonding of a stack of plates with fluid passages photo-chemically etched on one side of each plate by using a technique derived from that employed for electronic printed circuit boards - hence the name. The diffusion bonding process allows getting an interface-free join between the plates, giving it the base material strength and very high pressure containment capability. The fluid passages have approximately a semicircular section, being typically 0.5 to 2 mm wide (but larger diameters are available, up to 5 mm). There are different etching patterns,

as desired to make a cross flow, counter flow or multipass cross-counter flow exchanger. Besides, the passage shape may be corrugated or straight, depending on a number of factors such as the fluid used, the heat load and the pressure drop requirements. After bonding, several blocks can be welded together to provide flow capacity to any duty. Headers and nozzles are welded directly onto the final core block. Figure 1 illustrates a typical PCHE core configured for a counter-current flow.
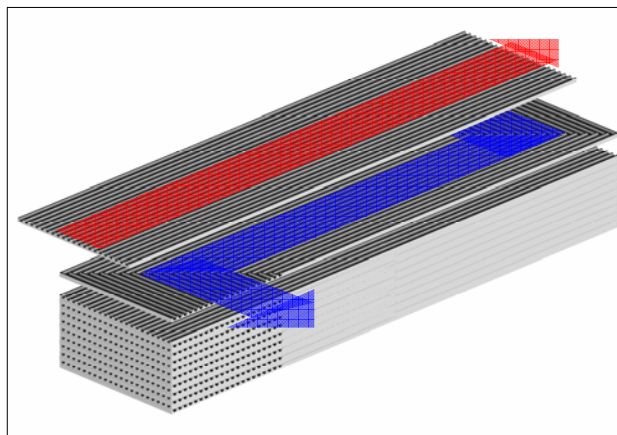

Fig. 1 - Printed circuit heat exchanger core configured for a counter-flow arrangement

In this sizing problem a criterion is followed that is the minimization of the overall frontal area. In a PCHE frame it depends, among other things, on the ratio between the two stream diameters: the greater the latter, the smaller the former. So, the highest ratio between the channels diameters should be chosen to minimize the frontal area. Moreover, the core flow length decreases with the diameters, once their ratio is fixed. But some constraints have to be set, a minimum flow length, equal to 0,8 m, to get a true counter-current flow and a maximum value for the diameters ratio, which here is 2,5. Generally speaking, geometry with different diameters on the two sides is not preferable, for manufacturing reasons, so their ratio should be kept closest to one. (Gezelius, 2004).

The f Fanning factor and the Colburn j factor to be used in the heat transfer and pressure drops calculation are applied, assuming that the streams flow within straight smooth channels.

The sizing procedure is based on the *core mass velocity equation* (Kays and London, 1984). It couples the heat duty and the pressure drops, enabling the computation of the first approximate value of the fluid mass velocity, G. As a result, the iterative solution to the problem converges relatively fast with this estimated value of G.

For duties involving high gas flow rates with pressure drop constraints it's quite logical to expect a short and fat shape. This is the present case: the sizing calculation of a PCHE core gives the results summarized in Table 2.

Tab. 2 - Thermo-fluid dynamic sizing results for a counter-current flow Printed Circuit Heat Exchanger and a Plate Fin Heat Exchanger core

Counter-current flow configuration		
	PCHE	PFHE
Gas channel D (m)	$4 \cdot 10^{-3}$	
Air channel D (m)	$1.6 \cdot 10^{-3}$	
Pitch/Diameter ratio	1.3	
Plate Thick./Diameter	0.75	
Gas side fin surface		Plain 5.3
Air side fin surface		Strip 1/9 – 24.12R [a]
Core cross section (m^2)	29.2	10.3
Stream flow length (m)	0.8	2.5
Effectiveness	0.91	
Outlet T (°C) gas/air	570/798	
Air side h (W/m^2K)	271.3	1061
Gas side h (W/m^2K)	106.7	98.6
Overall U (W/m^2K)	76.6	89.2
Gas pressure drop (%)	5.00	3.88
Air pressure drop (%)	0.123	5.00
Compactness (m^2/m^3)	941.7	876.2

As one could expect, a very large frontal area to comply with the required gas pressure drop characterizes the resulting overall core shape. A practical PCHE counter-flow configuration can be obtained only by allowing a crossflow heat exchange zone near to the exchanger ends, as Figure 1 shows. So, the width should be kept one half the flow length at most. In this case, it would mean to get a (73 x 0.4 x 0.8 m) heat exchanger. Consequently, a true counter-current flow is not feasible.

Regarding the PFHE model, a variety of methods are described in the literature for plate-fin surfaces performance comparison (Shah, 2003). Among these, the Surface Flow Area Goodness Factor, defined as j/f, is used to get the minimum frontal area for the exchanger. According to this principle, the Plain 5.3 surface is adopted for the gas side, as it's the critical one. A modified air side surface is selected, by increasing the fin thickness, to allow the withstanding of the fluid pressure at the maximum operating temperature. By this way a quite cheap material as Alloy 800H could be used, its cost being only 2.5 times the AISI 347, while the others heat resistant alloys are 4÷5 times more expensive. The f and j factors are expressed as polynomial functions of Re, whose coefficient were deduced from a fit of available literature data (Kays and London, 1984).

A PFHE type would be possible, if carefully designed distribution headers are assembled that could guarantee against excessive flow maldistribution and avoid unacceptable pressure drops. Usually, external header tanks with nozzles welded into them are applied to the exchanger. Internal distributors are required, since one stream must be

taken in/out in at least partial crossflow (Hesselgreaves, 2001).

All of the listed drawbacks arise from the high volumetric flow ratio between the two streams. It yields a too much burdensome constraint on the gas side pressure drop, due to its low pressure, compared to the air one.

2.2. Feasible solutions

A two-pass cross-counter flow model with two identical modules would be a valid alternative, for two main reasons:
- The single cross-flow modules have an efficiency value close to the maximum practically allowable for such a configuration.
- Their inlet and outlet headers can be made and installed as in most heat exchangers.

A sizing calculation for a two-modules core, both PFHE and PCHE, leads to the results shown in Table 3, with reference to the single module. The surfaces are the same previously used.

Tab. 3 - Thermo-fluid dynamic sizing results for a PFHE and a PCHE core in cross-counter flow

Cross-Countercurrent configuration, 2 identical modules		
	PCHE	PFHE
Gas flow length (m)	0.45	1.89
Air flow length (m)	2.00	1.75
Core height (m)	17.05	6.66
Single module effectiveness	0.789	
Overall effectiveness	0.91	
Outlet T (°C) gas/air	570/798	
Air side h (W/m^2K) [b]	739.5	931.3
Gas side h (W/m^2K) [b]	103.7	87.5
Overall U (W/m^2K) [b]	91.0	80.9
Gas pressure drop (%)	5.00	5.00
Air pressure drop (%)	5.00	5.00

As is obvious, the resulting overall volume increases compared to the counter-current one. But the proportional rise is not the same for the two heat exchangers. It's about 31.4 percent for the PCHE model and 71 percent for the PFHE. This could be explained by looking at the air side heat transfer coefficient. In the cross-current PCHE its value almost triples, as the fluid conditions are changed, from laminar to turbulent, due to the reduction in the free-flow area.

2.3. Microturbine systems

In microturbine systems the two parameters that have potential for efficiency advancement are increased values of turbine inlet temperature and higher recuperator

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[a] The fin thickness is increased by a factor 1.5 to withstand the fluid pressure at the maximum temperature

[b] relating to the coldest module

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effectiveness. Both of them have a strong impact on the recuperator. A higher turbine inlet temperature means that the recuperator hot gas inlet temperature increases, this necessitating the use of higher cost superalloys for a component that is already the most expensive in the system.

Recuperator technology is not new - recuperators are used in larger industrial gas turbines - but it is also not perfected, particularly at this new size level. One of the chief challenges in designing recuperators is to obtain a surface area that is both large enough to absorb the heat and able to absorb the thermal shock due to temperature change. This is particularly difficult to obtain given the space restrictions of a microturbine. Recuperators also operate in a hostile environment, undergoing thermal stress as they exchange heat varying from room temperature to approximately 600-650°C.

Differential pressure ratios inside the recuperator also contribute to weaken it: compressed air and high-temperature exhaust gases push against the inner surfaces of the recuperator from opposite directions, deforming the metal components. These effects combine to make the recuperator susceptible to heat leakage and to creep.

In manufacturing microturbine recuperators two main design have entered into use: primary surface design (PSR) and plate-fin design (PFHE).

The first one attempts to mitigate the heat exchange stress by maintaining a constant surface area. This is achieved through alternating layers of thin (less than 0.1 mm thick) metal sheets folded into a corrugated pattern, which provides the requisite surface area without the large number of joints (Figure 2a). Pairs of these sheets are welded together around the perimeter to form air cells, which are then pressure checked before being welded into the recuperator core assembly. There are no internal welds or joints within the air cell. Heat flows through the corrugated layers such that hot air proceeds in a cross-counter flow through the recuperator.

The PSR layouts are divided into annular recuperators, which are wrapped around the turbine, and rectangular recuperators installed behind the rotating machinery. Each of them may be attractive for different applications.

The Plate Fin heat exchangers use thin sheets of metal lined up parallel to one another, like the gills of a fish. This provides a large surface area, but there are corresponding disadvantages. The most important one is its minor reliability, compared to a PSR.

The brazed plate-fin recuperator is a rigid structure. As the size of the plate-fin recuperator increases, the structure becomes less capable of handling thermal gradients resulting from sudden changes in operating conditions, including cold starts and shutdowns. The continuous brazed joints between the two plates and air fins and between adjacent plates and exhaust gas fins, lock the structure firmly together (Figure 2b). It is too rigid to accommodate the temperature gradient between the sides from cell to cell and across the entire recuperator structure.

The brazed joints distort to comply with high thermal strains, leading to individual failures that, in turn, connect and propagate from the brazed joint into the parent material of the plate. This cell failure results in leakage of the pressurised air into the exhaust gas stream, reducing the turbine power and increasing the fuel consumption.

Fig. 2 - Compact heat exchangers surfaces:
a) Primary surface sheets
b) Brazed Plate-Fin Recuperator
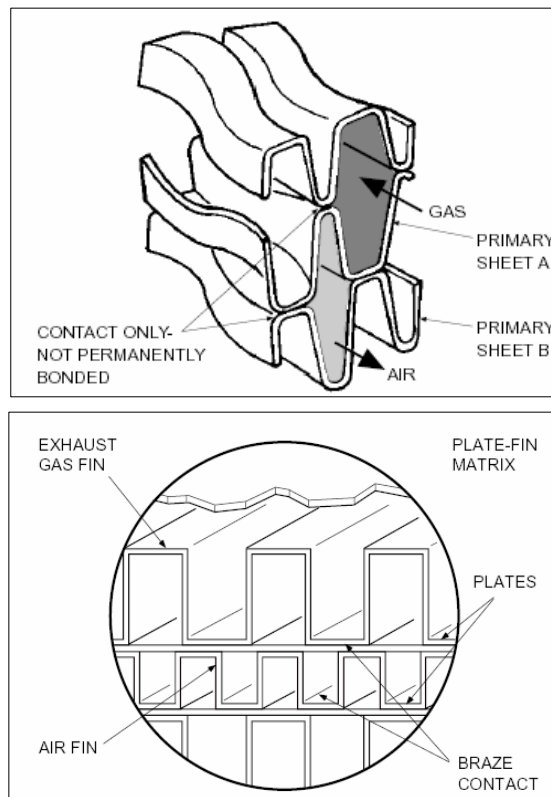
PSR design is inherently resistant to cyclic thermal fatigue because it can flex to relieve thermal and mechanical strains. Its construction allows the individual sheets to move relative to each other and relieve the thermal strain. An additional benefit is the intrinsic natural damping characteristics derived from stacking cells together with multiple friction interfaces. These several contact points absorb displacements from vibration sources (Ward, 1995).

3. MATERIALS ISSUES

All the previously investigated high temperature heat transfer systems have in common high materials performance requirements.

Currently, for the majority of gas turbines operating at temperature lower than 650°C, PSR and PFHE are made of thin sheet of Type 347 stainless steel (18Cr-10Ni-1Nb), which provides excellent oxidation resistance. Its performance up to this temperature is derived from the presence of a thin, adherent chromium oxide (Cr_2O_3) film,

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which forms in service at a predictable rate. It inhibits further transport of oxygen to the underlying material, thus avoiding excessive oxidation damage. At higher temperature, namely above 700°C, film instabilities lead to high oxidation rates and possibly cracking with spallation and Cr depletion, which could breach the protective layer and accelerate oxidation (Ward, 1995). For that reason, at temperatures higher than 650°C or where severe corrosion may be anticipated, alternative materials with improved properties meeting these specific needs are to be used.

Before an alloy can be adopted for the fabrication of a PSR or a PFHE surface, several questions must be answered. First, the alloy must be capable of being rolled into thin (<0.1 mm) foils that have adequate uniform elongation so that they can be formed into the required corrugated pattern. The foil-forming process must lend itself to continuous operation in order to minimize cost. Second, the fine-grained alloy foil must possess the necessary creep strength at the maximum operating temperature. The creep strength is only one measure of lifetime of the alloy; the other is its resistance to oxidation and other corrosive species present in the exhaust gas. Third, the final product must be the most cost-effective combination of the two previous properties (Omatete et al., 2000).

Intensive research activity has been carried out during last ten years, especially by Oak Ridge National Laboratory (ORNL), that allows getting a wide overview of the candidate alloys for meeting the performance requirements (Stinton and Raschke, 2003).

Type 347 stainless steel has been shown capable of being modified to operate up to 750°C with creep resistance superior to the standard commercial material, at a cost less than twice the standard 347 SS. Nevertheless, at temperatures above 700°C, corrosion resistance and the effects of moisture remain the dominant limiting factors for Type 347 steels. A breakaway oxidation in water vapour could still happen, but the delay in such attack would provide a substantial added benefit of these modified alloys for recuperator applications (Pint et al., 2001, Pint and Peraldi, 2003).

Coming to better, and also more expensive alloys the following two groups could be identified, with Alloy 740 representing a sort of dividing line:

- Alloys comparable with 347 stainless steel: modified Alloy 803, Alloy 602 CA, Haynes HR120 and Haynes HR230.
- Alloy considerably stronger than 347 SS: Alloy 625, Haynes HR214, Hastelloy X and Plansee alloy PM2000 (ODS alloy).

Table 4 shows their nominal composition, while Figures 3a and 3b illustrate the creep strains versus time for many of the alloys. The creep data refer to creep-rupture tests in air at 750°C and 100 MPa. This aggressive creep testing was chosen as an accelerated screening condition for advanced alloys, as most commercial recuperators made from standard Type 347 SS operate at low stress below 700°C.

Tab. 4 - Composition of some heat resistant austenitic stainless alloys processed into foils (Maziasz and Swindeman, 2003)

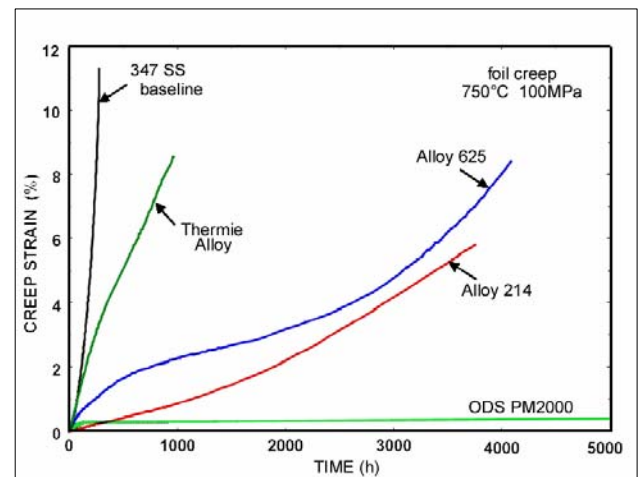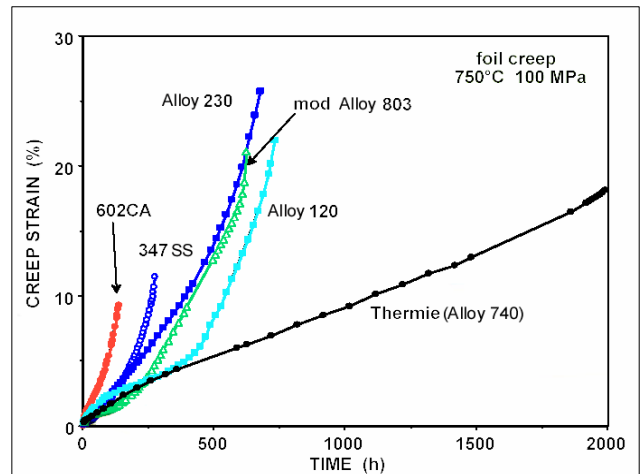|	347 SS	All. 803	All. 740	HR 120	HR 230	HR 214	All. 625	602 CA
Fe	68.7	40	2	33	3 *	3	3.2	9.5
Cr	18.3	25	24	25	22	16	22.2	25
Ni	11.2	35	48	32.3	52.7	76.5	61.2	63
Mo	0.3	n.a.	0.5	2.5 *	2		9.1	
Nb	0.64	n.a.	2	0.7			3.6	
C	0.03	0.05	0.1	0.05	0.1		0.02	0.18
Si	0.6	n.a.	0.5	0.6	0.4		0.2	
Ti	10^{-3}	n.a.	2	0.1			0.23	0.15
Al	$3 \cdot 10^{-3}$	n.a.	0.8	0.1	0.3	4.5	0.16	2.2
Oth.	0.2 Co	n.a.	20 Co	3 Co, 3 W, 0.2 N	5 Co *, 14 W, La	Y		

* means maximum allowable element content

Fig. 3 - Creep strain versus time for some foils of heat resistant alloys in air at 750°C and 100 MPa. a) alloys comparable with Type 347 SS; b) Stronger alloys (Maziasz and Swindeman, 2003, Pint et al., 2001)

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It should be pointed out that the results apply to foil creep, where foil is a few tenths of millimetre thickness sheet. At such a reduced thickness, a maximum value of 10 μm for the grains diameter should be expected, compared to 50-100 μm for typical tubular and plate products (Maziasz *et al.*, 1999). As the creep resistance is dependent on the grain size, (a greater one providing a higher strength) a different behaviour will be obtained for the same materials depending on their form. That's the reason why some alloys, for example HR230, show a worse creep resistance than others that usually are weaker.

It's clear that there is a wide range of more heat resistant and corrosion resistant alloys and superalloys commercially available, but many of them cost significantly more than Type 347 stainless steel. That could be seen in Table 5, where some relative costs are compared setting the 347 SS as a baseline.

From these limited data, Alloy 625 is one of the most cost effective improvements relative to 347 steel, but at a substantially greater initial cost. The same is true for HR214, which has a longer rupture life and a much higher cost too. By contrast, alloys HR230 and 602CA cost about 7 and 9 times more than Type 347 steel respectively, and HR230 lasts only 1.6 times longer to the 10% strain limit, while 602CA shows less creep resistance than Type 347 steel. These alloys are not cost effective replacements for 347 steel, simply based of their creep behaviour.

Tab. 5 - Relative costs for some alloys using Type 347 stainless steel as a baseline (Maziasz and Swindeman., 2003, Pint et al., 2001)

347 SS	Alloy 803	HR 120	Alloy 625	HR 230	Alloy 740	HR 214	ODS PM2000
1	3	3.5	5	7	9	9	10

The stainless alloys with 25%Cr and 35%Ni, modified Alloy 803 and HR120, have about twice the creep resistance of 347 steel, and cost about 3-3.5 times greater. Therefore, both of them would represent possible performance upgrades relative to Type 347 steel for using to about 750°C or slightly more, while Alloy 625 and HR214 would be the very high performance alternative up to 800°C and possibly higher.

3.1. Environmental resistance performances

As previously said, a fundamental requirement for the materials is the oxidation and corrosion resistance at high temperatures. The oxidation of alloys protected by the formation of a chromium oxide (Cr_2O_3) film (such as SS 347, 20/25/Nb, 253 MA and HR230) has been compared with alloys protected by the formation of an aluminium oxide (Al_2O_3) such as HR214 and ODS alloy PM2000.

Figure 4 shows the total mass gain in foils of alloys heated at 900°C in laboratory air for greater than 10,000 hours. The Cr_2O_3-forming foils show excessive weight

gained in less than 4,000 hours, whereas the thinner (50 μm) Al_2O_3-forming foils show minimal weight gained after 12,000 hours. This is due to the limited reservoir of Cr in those stainless steel foils and the fast Cr consumption (oxidation) at 900°C compared with the rate of formation of Al_2O_3 on the alloys that contain aluminium (Pint *et al.*, 2001).
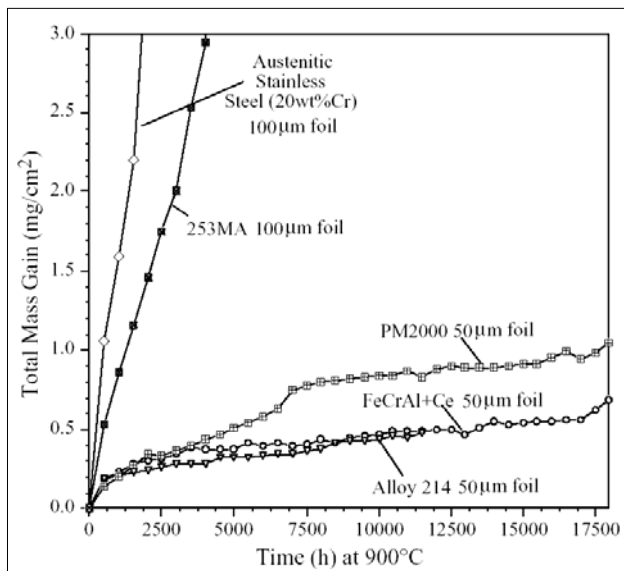

Fig. 4 - Total mass gain during 500-hour cycles at 900°C in laboratory air for Cr_2O_3 and Al_2O_3 forming specimens (Pint et al., 2001)

This indicates that alloys that are protected by Cr_2O_3 formation can be used roughly up to 800°C; beyond that, only alloys protected by Al_2O_3 formation will survive.

In many power plant combustion systems the combination of high temperature and aggressive environment, which may contain oxidizing, sulphidizing and chlorinating elements, obliges to give up the typical solution strengthened alloys, either Fe or Ni based, starting from 750-800°C. Compared to such environments, up to now two ways only have been followed: ODS alloys and ceramics.

ODS means Oxide Dispersion Strengthened alloy, referring to the distribution of fine particles of an inert oxide throughout the alloy matrix and the development of a microstructure in which large grains are developed. Typically, ODS alloys contain 0.5 % of an oxide, such as Y_2O_3 or ThO_2, dispersed through the material. Table 6 shows the nominal compositions of some ferritic ODS alloys.

The reason for the interest in ODS alloys is summarized in Figure 5, which compares the average 100000 hours stress rupture strength for all available wrought or cast high-temperature alloys with the theoretical maximum rupture strength that could be obtained if the strengthening mechanisms used were fully optimised. The typical longitudinal stress rupture strength for ODS-FeCrAl alloys, also shown on the diagram, illustrates their strength

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advantage. Essentially, ODS alloys have potential for use to temperatures where otherwise ceramic materials would have to be considered (Wright *et al*., 2003).

Tab. 6 - Composition of some ferritic ODS alloys (Wright et al., 2003)

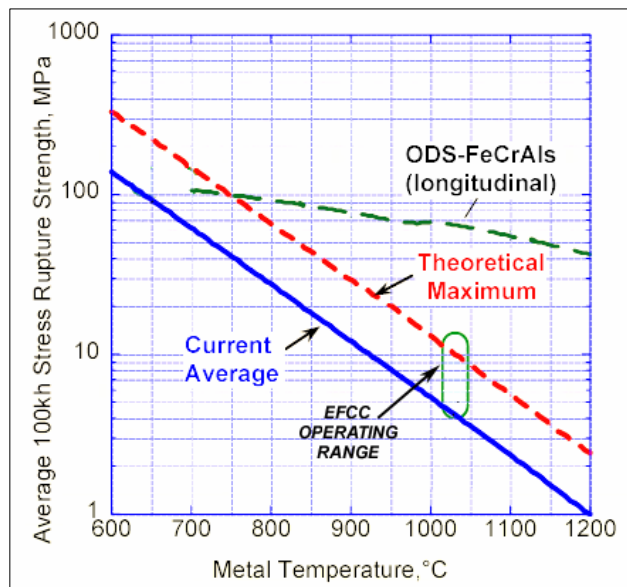| ALLOY	Fe	Al	Cr	Mo	Ti	Dispersoid
ODS-Fe$_3$Al	Bal.	15.9	2.2		0.07	Y$_2$O$_3$
MA956H	Bal.	5.77	21.66		0.4	Y$_2$O$_3$
MA956	Bal.	4.46	19.64		0.39	Y$_2$O$_3$
PM2000	Bal.	5.5	20		0.5	Y$_2$O$_3$
ODM751	Bal.	4.5	16.5	1.5	0.6	Y$_2$O$_3$

Fig. 5 - Schematic comparison of the current average and theoretical maximum creep strength of some high-temperature alloys (Wright et al., 2003)

The dispersed oxide particles help to pin grain boundaries reducing their ability to creep at high temperature and stabilize oxide scales. In fact, they are much more stable than the typical strengthening particles such as γ' and carbides, present in conventional high temperature alloys and the oxide scales possess better spallation resistance (Quadakkers *et al*., 1989).

It is expected that ODS alloys in service will be subjected to mechanical loading such that their rate of creep would be quite slow, therefore their service life is likely to be determined by the oxidation behaviour. Currently, it limits the maximum use temperature at 1050-1150°C, depending on the specific environment aggressive constituents (Wright *et al*., 2003).

The ODS microstructure requires a powder metallurgical production process to be used and also poses problems in joining these alloys. Further drawbacks are the cost and a high directionality.

ODS tubes have been developed for the US HiPPS heat exchanger with MA754, a Ni-Cr based alloy, whereas the European EFCC program heat exchanger is planned to be built with ceramic tubes, hence the difference between the two projects in the maximum temperature achievable through the heat exchanger. Unfortunately, up to now no laminar products have been manufactured with ODS alloys, so their application in the field of the extended surfaces is unlikely to take place in the near future.

4. CONCLUSIONS

A general review of high temperature compact heat exchangers has been performed. They are required in many industrial processes for inherent reasons or in order to improve the efficiency. This is particularly true in the case of the power production plants.

A representative group of high temperature systems has been critically analysed and some common technological issues have been pointed out.

The difficulty of managing highly different massive flows in high-pressure gas turbine recuperators has been shown through a numerical example, suggesting some possible solutions.

As regards the material issue three reference ranges have been identified, depending on the temperature.

In order, one could consider current developments in microturbine recuperator materials. A modified version of Alloy 803 and HR120 alloy have been found the most suitable from a cost effective standpoint up to about 750°C. In succession, the nickel-based superalloys are the most promising up to 800-850°C and also above, if a properly aimed design of microstructure and mechanical properties is planned. Finally, the more suitable classes of materials for using in the high temperature indirect firing have been identified: ODS alloys and ceramic materials. Many differences could be highlighted between latter two, for example the maximum allowable temperature, which for ceramics may be up to 200-250°C higher than ODS alloy. It should be considered that, when properly applied, ODS-FeCrAl alloys could be usable to temperatures up to 1150°C in components at the lower end of the temperature range where ceramics typically are thought to be required.

As concerns the main current challenges to be faced in the near future, the followings should be recalled:

- Even if it's currently possible to find suitable metallic materials on the entire temperature range between 700°C and 1150°C, each one of them has to comply with manufacturing limitations. For instance, it is not easy to process ODS in foils, and it's still quite difficult to join them.
- Many alloys are at a developmental stage yet, which causes their projected cost to be very high.

It's essential that such items be deeply investigated, in order to get a successful development of high temperature

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compact heat exchange systems. In this perspective, a stronger connection between the investigations regarding seemingly different fields would be probably very profitable.

NOMENCLATURE AND ACRONYMS

D	Tube or channel diameter, m;
f	Fanning friction factor, dimensionless;
G	Fluid mass velocity, kg/m^2s
h, U	Heat transfer coefficient, W/m^2K;
j	Colburn factor, dimensionless;
$\Delta p/p$	Pressure drop, %, dimensionless;
GE	General Electric Company
HiPPS	High Performance Power System
ODS	Oxide Dispersion Strengthened (alloy)
PCHE	Printed Circuit Heat Exchanger
PFHE	Plate-fin Heat Exchanger
PSR	Primary Surface Recuperator.

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