Combustion heated thermionic systems

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ABSTRACT

Thermionic cogeneration is developing as a new energy conservation option. The characteristics of thermionic systems using combustion of natural gas as heat source are studied. The – until now unused – high temperature potential of fuels can be applied in order to extract work using the high temperature thermionic conversion process. The high temperature of the emitter and the required large heat flux to it demand a high radiation temperature of the burner walls. As natural gas can only just reach these requirements the resulting system efficiency drops to an intolerable low level if the combustion air is not preheated and the heat from the flue gasses is not recuperated. An analytical study shows the effects of using the collector cooling air to preheat the combustion air and the influence of the heat exchanging capacities of the recuperator and the burner walls. It is shown that using the collector cooling air leads to a partial increase of the system performance and that using a recuperator increases the performance to the maximum attainable value. The geometry of the recuperator is discussed.

NOMENCLATURE

cp	specific heat	$[Jkg^{-1}K^{-1}]$
D	diameter	[m]
d	wall thickness	[m]
H	enthalpy	$[Jkg^{-1}]$
kA	heat exchanging capacity	$[WK^{-1}]$
L	length	[m]
M	mass flow	$[kgs^{-1}]$
Nu	Nusselt number	[–]
T	temperature	[K]
Q	heat flow or dissipated power	[W]
V	flow rate	$[m^3s^{-1}]$
ϕ	ratio of mass flows air/gas	[–]
η	viscosity	[Pas]
λ	heat conductivity	$[Wm^{-1}K^{-1}]$

suffix

a	air ambient
ar	air after recuperator
b	burner
c	collector
ca	air after collector cooling
e	emitter
flue	flue gas
fluer	flue gas after recuperator
r	recuperator
rc	recuperator channel

INTRODUCTION

The objective of this study is to obtain insight in the behavior of a combustion heated thermionic system in order to design a cost competitive thermionic cogeneration system.

Thermionic energy conversion evolves from a space power source to a terrestrial energy conservation option. It also fits well in the effort to increase the efficiency of Carnot processes by using the unused high temperature potential of combustion fuels.

Most fuels show an adiabatic combustion temperature of about 2000 °C which is sufficiently high for the thermionic process. The hot side, *emitter*, of a thermionic converter however is isothermal. In combination with the high heat flux to the emitter it is impossible to cool the flue gasses to a low temperature. This implies that only a small temperature range of the combustion gasses can be applied for the thermionic conversion, resulting in poor system efficiencies (electrical output power as a fraction of the fuel enthalpy power).

Solutions to this problem are an increase of the inlet combustion air temperature using the heat of the cold side, *collector*, or a decrease of the flue gas temperature by recuperating the heat to the combustion air and gas. Thermodynamically seen the best results are obtained by leading any heat through the converter and by maximizing combustion and herewith the emitter temperature.

Studies on combustion heated thermionic systems from Fitzpatrick (1977), Miskolczy (1978) and Dick (1980) concentrate on thermionic topped power plants. The thermionic combuster system is evaluated by Dick (1984). This study concentrates on the characteristics of a single thermionic module, including the burner.

To asses the characteristics of a combustion heated thermionic cogeneration system it was modelled as a set of coupled heat exchangers and a stylized thermionic process, allowing to solve the set analytically with the help of symbolic algebra. In this way the pitfalls of a numerical approach are avoided.

As no conclusive figures were found in literature on the adiabatic combustion temperature of natural gas, which is the design fuel, this temperature was approximated by assessing the equilibrium of the most important chemical reactions.

The thermal behavior of a potential recuperator geometry has been calculated in order to assess the approximate sizes.

PREHEATING AND FLAME TEMPERATURE

The objective is to assess the adiabatic flame temperature as function of the preheating of the air. The models proposed by the Dutch Gasunion (1988) and Gaydon and Wolfhard (1979) are used. The difference is that the model of Gaydon and Wolfhard accounts for dissociation, whereas the model of the Gasunion uses the simple heat balance. The assumptions incorporated in this model are:
— the flame is at constant pressure,
— there is no heat loss,
— in the burnt gasses equilibrium is assumed.

At very high flame temperatures the method will not be correct because additional dissociation of the burnt products is not taken into account. The dissociation equilibria considered are:

$$CO_2 \Leftrightarrow CO + 0.5\,O_2$$
$$H_2O \Leftrightarrow H_2 + 0.5\,O_2$$
$$H_2O \Leftrightarrow 0.5\,H_2 + OH$$
$$0.5\,H_2 \Leftrightarrow H$$
$$0.5\,O_2 \Leftrightarrow O$$
$$0.5\,N_2 + 0.5\,O_2 \Leftrightarrow NO$$

The total reaction scheme for the stoichiometric combustion yields in case of methane and air:

$$CH_4 + 2\,O_2 + 7.52\,N_2 \rightarrow CO_2 + 2\,H_2O + x\,NO + x\,N_2$$

In Dutch natural gas the methane concentration is 81.21 %(mol), 3.5 %(mol) other organic gasses, and 14.32 %(mol) N_2. The effect of preheating is introduced in three stages:
— first all components are preheated,
— second only the components in air (O_2, O, N_2, NO) are preheated, as the natural gas is not preheated,
— third together with the second refinement, the reaction heat is corrected according to:

$$H_{preheat} = H_{room} + \int_{T_{room}}^{T_{preheat}} (c_{p,unburnt} - c_{p,burnt})\,dT \qquad (1)$$

In fig. 1 the results are summarized.

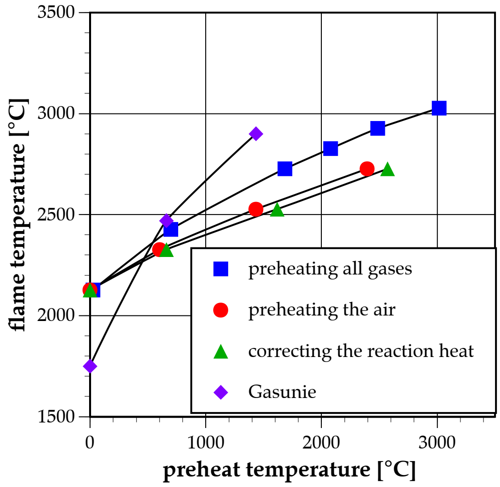

Figure 1 models of flame temperature as function of preheat temperature

The flame temperatures predicted differ markedly. The model of Gaydon, correcting for the reaction heat, is adopted as it is best confirmed with measurements.

The specific heat capacities of burnt natural gas and air are temperature dependent and their differences are important for the system characteristics, see fig. 2.

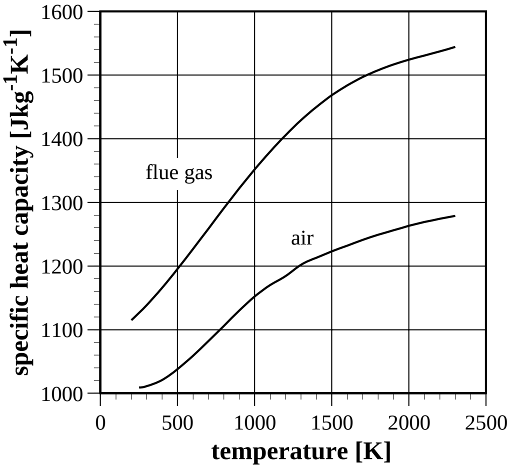

Figure 2 temperature dependency of the specific heat capacity of flue gas and air

MODEL

A scheme of the combination of recuperator, burner and thermionic converter is shown in fig. 3. In this scheme the burner air can be preheated both by using the collector cooling air and by recuperating heat from the flue gasses. The recuperator is a counterflow heat exchanger in order to minimize the exergy losses. In this study the physical properties of the two gas flows are averaged with respect to the temperature. This introduces an error but allows for an analytical relation for the temperatures:

$$T_{ar} = T_a + (T_{flue} - T_a)\frac{1 - e^{\left(-\frac{kA_{rec}}{cp_f(1+\phi)M_a}\left(1-\frac{cp_f}{cp_a}\right)\right)}}{1 - \frac{cp_f}{cp_a}e^{\left(-\frac{kA_{rec}}{cp_f(1+\phi)M_a}\left(1-\frac{cp_f}{cp_a}\right)\right)}} \qquad (2)$$

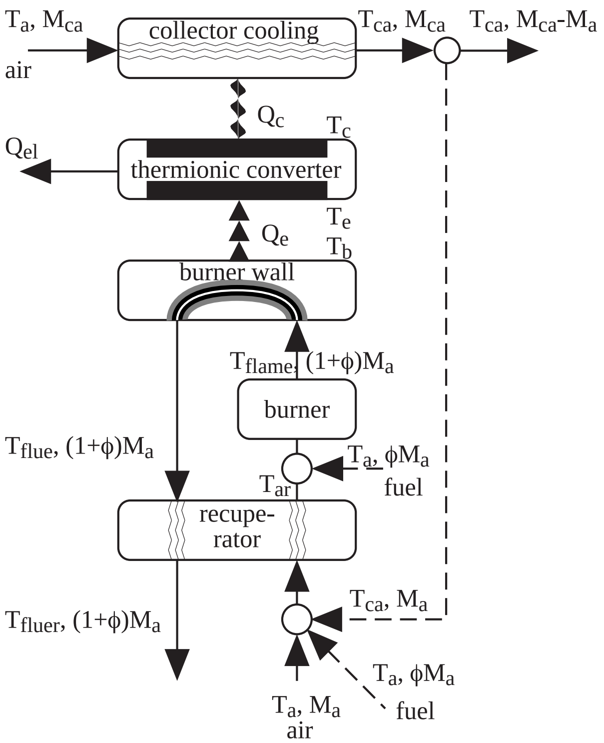

Figure 3 schematic of the system

The heat exchange with the burner wall is modelled as a longitudinal flow heat exchanger of which the burner wall is isothermal. This is a valid approximation as the burner wall radiates to the emitter and itself equalizing the temperature because of the highly nonlinear temperature dependency. The temperatures and flows are related according to:

$$Q_e = cp_{flame}(1+\phi)M_a\,(T_{flame} - T_b)(1 - e^{\frac{-kA_b}{cp_f(1+\phi)M_a}}) \qquad (3)$$

The heat exchanger at the collector cooling is not modelled as the system performance is only weakly dependent on the collector temperature. Instead the collector temperature is used as a free variable.

The heat exchange between burner and emitter is primarily effected by radiation. Convective heat transfer and recombination of radicals at the emitter are small and neglected in this study. The burner wall being porous is nearly black and envelopes the emitter, so:

$$Q_e = A_e\varepsilon_e\sigma\,(T_b^4 - T_e^4) \qquad (4)$$

The thermionic process has been stylized to a second order function of the emitter temperature only. The other variables are set at their optimum value with respect to the electrical output. The performance data are taken from Hatsopoulos (1973). The electrical power output is:

$$Q_{el} = (0.257T_e^2 - 638T_e + 394744)\;A_e \qquad (5)$$

The thermionic efficiency, defined as the electrical output divided by the heat input, is:

$$\eta_{el} = 1.716\;10^{-7}T_e^2 + 8.627\;\;10^{-4}T_e - 0.848 \qquad (6)$$

Equations (2)...(6) are coupled via the energy balance equations for the different subsystems. The solution is constructed using a symbolic algebra program.

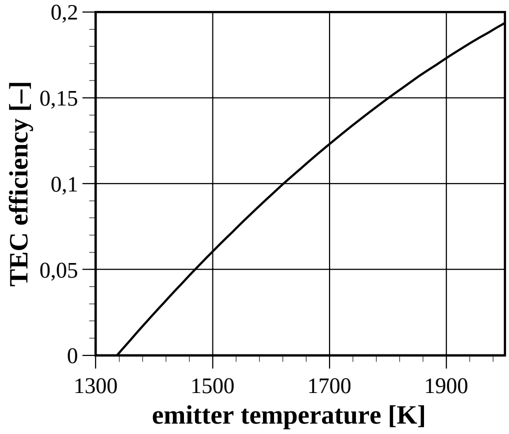

Figure 4 optimized TEC efficiency as a function of the emitter temperature

The resulting equation is a bit too lengthy so only a graphical presentation of the results will be given. If not stated otherwise the results are given for the projected operation point of the converter:

T_e = 1723	[K]
T_c = 873	[K]
A_e = 38 10^{-4}	[m^2]
Q_e = 1750	[W]

cesium reservoir temperature and gap optimized

RESULTS

NO PREHEATING – Natural gas, used as fuel, can deliver the required power and temperature, but at the expense of a very large gas flow. In fig. 5 the burner efficiency, defined as the heat through the TEC divided by the fuel supplied, is drawn.

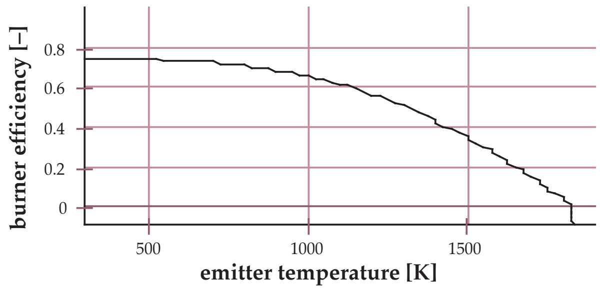

Figure 5 burner efficiency without preheating, kA_b = 10 WK^{-1}

The burner efficiency without preheating reaches zero just beyond the operating point.

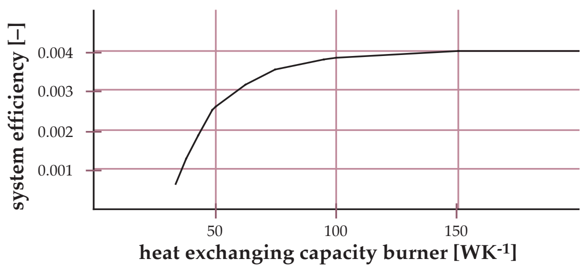

Figure 6 system efficiency as a function of the heat exchanging capacity of the burner wall

Enlarging the burner wall area increases the system efficiency only to the intolerable low level of 4 ‰ despite a large heat exchanging capacity of 150 WK^{-1}, see fig. 6. To obtain a reasonable system efficiency a better burning process is needed. To show the effect of a better caloric value of the gas only the enthalpy of the gas has been varied, see fig. 7.

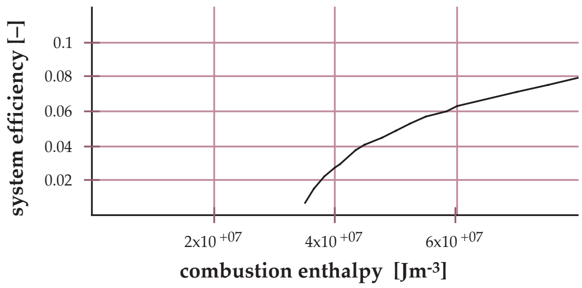

Figure 7 influence of the gas quality on system efficiency, $kA_b = 10\ WK^{-1}$.

The enthalpy of natural gas is less than 32 10^6 Jm^{-3} so the operation point is quite near the edge. This is because the temperature of the burner wall, needed to supply the requested power to the emitter, is quite near the combustion temperature.

As it is uneconomical to change the quality of the gas, two options are left; preheating the air or enriching the air with oxygen. As enriching the air does not seem very practical, we have to preheat the air using the collector cooling or a recuperator for the flue gasses.

PREHEATING BY COLLECTOR COOLING – The collector is cooled to a temperature of 900 K. The cooling can be used to preheat the air for the combustion process. The mass flow of the cooling air is larger than the combustion air flow needed, so only part of this flow can be used.

The collector air heat exchanger has not been included in the model as it does not significantly influence the system performance. Instead the collector cooling air temperature has been taken as a free variable with a maximum temperature of 873 K.

The system efficiency as a function of the emitter temperature shows maxima for each preheat temperature, see fig. 8.

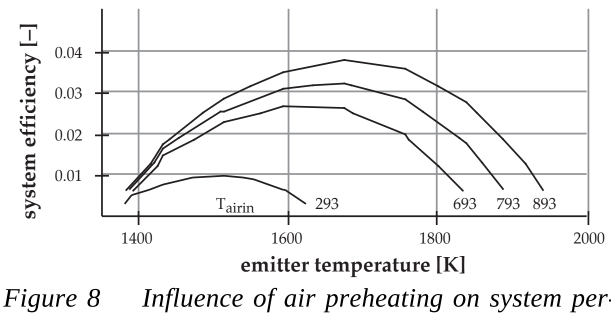

Figure 8 Influence of air preheating on system performance

The maximum system efficiency of 3.8% occurs at an emitter temperature of 1700 K, which is close to the design point of the ECS TEC. The system efficiency without preheating has a maximum of 1% at an emitter temperature of 1525 K. This drop in system efficiency as compared to the efficiency of the converter at the same emitter temperature (more than a factor 3) is caused by the high heat reject temperature, which cannot be used for the thermionic process.

A system with collector preheating only may be economical in a large industrial burner because of the simplicity of the system. For most smaller applications however the efficiency will be too low to justify the initial costs. To increase the performance the reject heat temperature has to be lowered.

BURNER–RECUPERATOR – The highest system efficiency can be obtained if any heat released in the burning process passes the TEC and consequently the flue gas temperature reaches ambient conditions. As the flue gasses cannot be cooled lower than the emitter temperature, a heat exchanger is needed to transfer heat from the flue gasses to the fresh burning air and fuel. To obtain the maximum effectiveness a counterflow heat exchanger is used in the calculations though a regenerator may come close.

Natural gas can be preheated to about 900 K before carbonization starts. In this study a mixture of air and gas in the recuperator is modelled and the maximum temperature of the actual separate gas flow is accounted for by accommodating the heat capacity flow.

As seen from fig. 1 the heat capacity flows of fresh and burnt gas differ. This implies that not all the heat can be transferred. In fig. 9 the nearly maximum burner efficiency is shown as function of the emitter temperature.

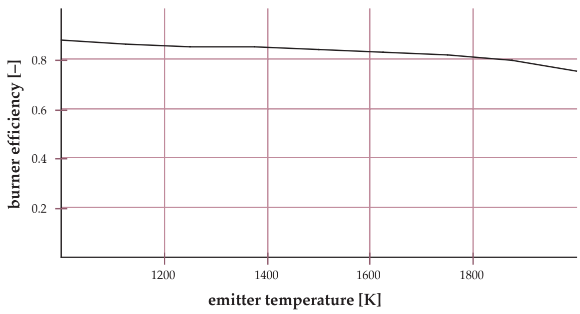

Figure 9 burner efficiency using a recuperator ($kA_r = 40\ WK^{-1}$, $kA_b = 10\ WK^{-1}$)

The burner efficiency is only weakly dependent on the emitter temperature when a recuperator is used. Actually the decrease with increasing emitter temperature is caused by the higher power transport and the consequent effectiveness drop in the recuperator. As compared to the results without a recuperator as shown in fig. 5 it follows that a recuperator stabilizes the burner performance. The fact that the efficiency is 0.8 instead of 1 is caused by the difference in heat capacity flows and not by the limited heat exchanging capacity of the recuperator.

The system efficiency as a function of the emitter temperature, see fig. 10, is nearly conform to the TEC efficiency as shown in fig. 4, be it about 20% lower.

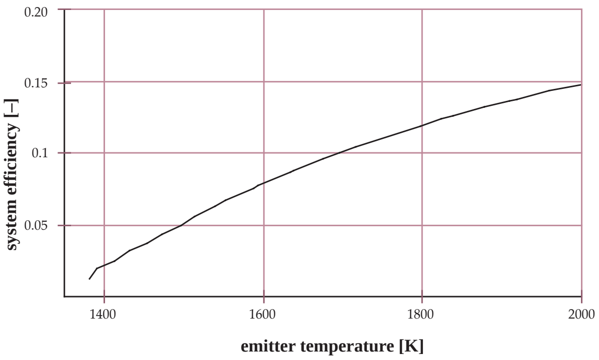

Figure 10 system efficiency using a recuperator (kA_r = 40 WK^{-1}, kA_b = 10 WK^{-1})

At design conditions the influence of the size of the recuperator on the system efficiency is shown in fig. 11.

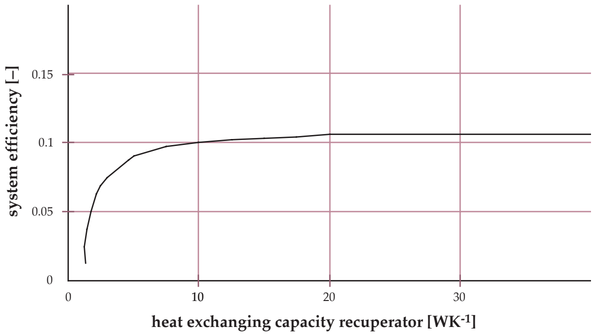

Figure 11 influence of heat exchanging capacity of the recuperator on system efficiency (kA_b = 10 WK^{-1})

For the design conditions it follows that a recuperator of 20 WK^{-1} leads to nearly maximum performance.

The flue gas temperature is lowered by applying a recuperator and reaches its minimum of about 720 K for a recuperator of about 40 WK^{-1}, see fig. 12.

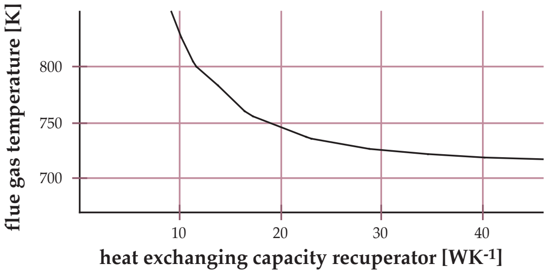

Figure 12 Influence of heat exchanging capacity of the recuperator on flue gas temperature (kA_b = 10 WK^{-1})

The heat exchanging capacity of the burner wall, which may be constructed of porous ceramics, has a similar effect on system performance, see fig. 13. The maximum system efficiency is reached at a heat exchanging capacity of 3 WK^{-1}. Both the heat exchanging capacities of burner and recuperator have an important influence on system performance.

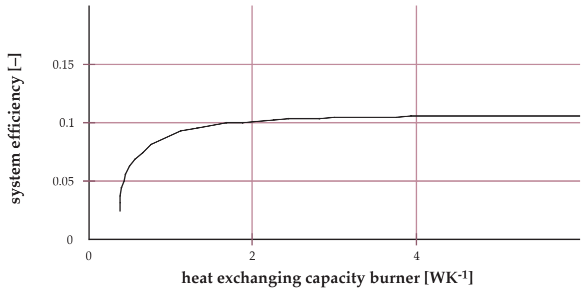

Figure 13 influence of heat exchanging capacity of the burner wall (design point, kA_r = 40 WK^{-1})

The most cost effective combination can be chosen using fig. 14 in which the heat exchanging capacity of the burner is given as a function of that of the recuperator with the system efficiency as parameter.

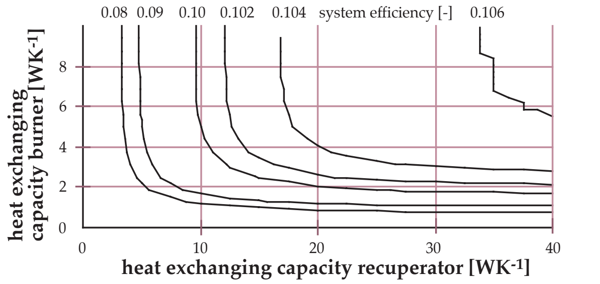

Figure 14 relation between the heat exchanging capacities of burner and recuperator with the system efficiency as parameter.

Another important parameter in the system behavior is the emissivity of the emitter. Together with the emissivity of the burner and the geometry of the emitter–burner combination it represents the heat exchanging capacity by radiation to the TEC. By choosing a porous burner wall, surrounding closely the emitter, the view factor is 1 and the burner emissivity is nearly unity, independent of the material used. The emitter emissivity however may degrade during life time e.g. by the forming of quartz when using SiC as emitter coating. The use of a recuperator tends to offset the degradation of the emissivity by increasing the burner temperature. In fig. 15 it can be seen that the emissivity may drop to less than 0.4 without much effect on system efficiency.

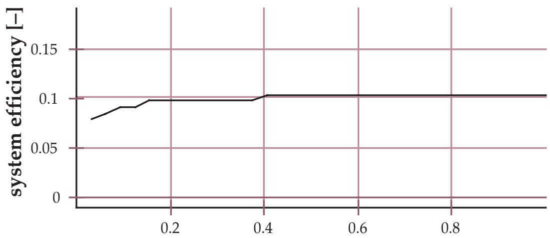

Figure 15 Influence of the emitter emissivity on system efficiency (kA_r = 40 WK^{-1}, kA_b = 10 WK^{-1})

This is quite nice for the designer though a consequence is a higher burner wall temperature.

COMBINATION OF RECUPERATOR AND PREHEATING – Preheating the air with the collector cooling and also using a recuperator may lead to better system performance at lower costs. In fig. 16 the system efficiency of this combination is shown for the design conditions.

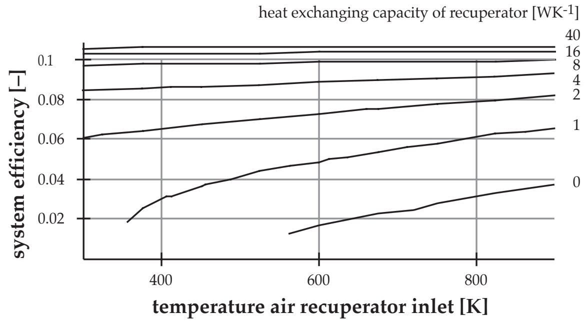

Figure 16 Combining preheating and recuperation at design conditions (kA_b = 10 WK^{-1})

It follows that preheating has effect when the recuperator is not very effective. For recuperators larger than about 15 WK^{-1} the effect of preheating on the system efficiency is very small. The only effect however is an increase in recuperator and flue gas temperature.

In fig. 17 the effect of a recuperator on system efficiency is shown as function of the emitter temperature without preheating, whereas in fig. 18 the air is preheated to the maximum value of 873 K.

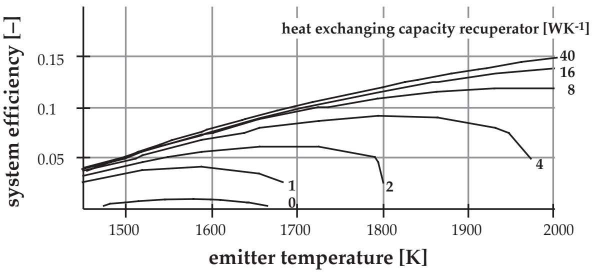

Figure 17 influence of recuperator size on system efficiency without preheating

It is interesting to note that for each recuperator size the efficiency shows an optimum value for the emitter temperature. This optimum emitter temperature increases with recuperator effectiveness.

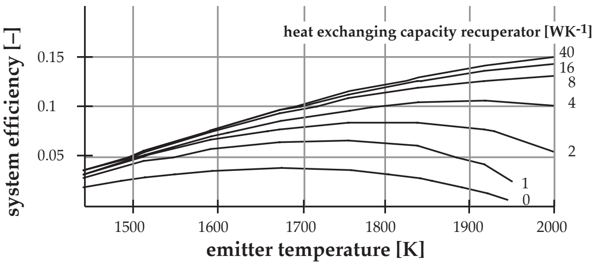

Figure 18 influence of recuperator size on system efficiency with preheating

The optimum system performance as function of the emitter temperature without preheating is higher than with preheating, though a larger recuperator is needed. For a large recuperator the system efficiency goes to the same limiting curve, both with and without preheating.

From this comparison it is clear that the best results are obtained when using a recuperator and that preheating in combination with recuperation decreases the maximum system performance for a given emitter temperature.

Using a recuperator does not increase the burner temperature, only the burnt gas temperature is increased and the flue gas temperature decreased.

RECUPERATOR GEOMETRY AND SIZE

The recuperator is a counterflow heat exchanger. To obtain some idea of the characteristics a simple geometry is chosen; square channels of which each side is formed by the side of a counterflow channel.

As the air plus gas flow density to and from the emitter is rather low (typical 0.2 – 0.3 $kgs^{-1}m^{-2}$) the flow will be laminar if the channel diameter is small. Making the diameter small is attractive in order to increase both the heat transfer from the gas to the wall and the heat exchanging area. The Nusselt number in such a flow with uniform heat flux is 3.6. The pressure drop in such a flow is relatively low. Though heat exchange by radiation can be significant in the channel it is not considered here because of the small channel diameter as compared to the length. This may lead to a conservative estimate of the recuperator size. The heat exchanging capacity of the recuperator, averaging the physical properties, satisfies:

$$kA_r = \frac{2D_{rc}L_r \,\mathrm{round}\left(\frac{1}{4}\frac{\pi D_r^2}{(D_{rc}+d_{rc})^2}\right)}{\frac{8D_{rc}}{\pi Nu\lambda_a}+\frac{d_{rc}}{\lambda_r}} \qquad (7)$$

The recuperator is supposed to be no larger in diameter than the TEC, allowing for a modular design of a TEC integrated with a burner recuperator. As construction material a ceramic is used with a heat conductivity of λ_r = 10 $Wm^{-1}K^{-1}$. The wall thickness is d_{rc} = 0.0005 m.

The channel width D_{rc} versus the channel length L_r with the heat exchanging capacity of the recuperator is given in fig. 19 for design conditions of the TEC.

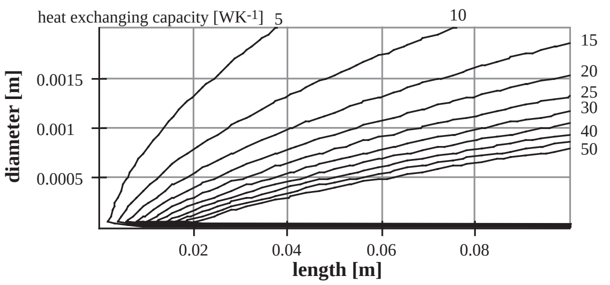

Figure 19 relation between the heat exchanging capacity, channel width and length

As seen in fig. 11 the system efficiency under design conditions is nearly maximum for a heat exchanging capacity of the recuperator of 20 WK^{-1}. This implies e.g. a channel length of 0.06 m at a channel diameter of 1 mm, which is a common size in ceramic burners. Making the channel diameter smaller increases the heat exchanging capacity until, at very low values, the conductivity of the ceramic starts dominating.

The recuperator dissipates power by pressure loss. In a laminar flow the loss is:

$$Q_{vr} = \frac{4\eta_a \pi^2 L_r V_r^2}{D_{rc}^4 \,\mathrm{round}\left(\frac{\pi}{4}\frac{D_r^2}{(D_{rc}+d_{rc})^2}\right)} \qquad (8)$$

The loss is plotted as function of the length and the channel diameter in fig. 20.

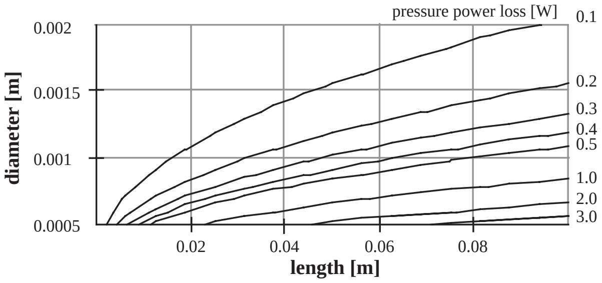

Figure 20 pressure drop loss as a function of channel length and diameter

Combining figs. 19 and 20 it can be seen that a good recuperator may show a low ventilation loss. For the sizes mentioned before of a length of 0.06 m and a diameter of 1 mm the power loss is 0.35 W at a heat exchanging capacity of 20 WK^{-1}. The corresponding pressure difference is 175 Pa, which is in the upper range of usual ventilators.

CONCLUSIONS

Combustion heated thermionic systems without preheating and/or recuperation of the energy of the flue gasses exhibit a low maximum system efficiency of 1% using natural gas. This maximum efficiency can be increased to 3.8% using heat from the collector cooling to preheat the combustion mixture. Using a recuperator to cool the flue gasses, preheating the combustion air, increases the efficiency up to the limiting value which is a function of the emitter temperature. The recuperator has a maximum effectivity of 85% because of the difference in heat capacities of air and flue gas. At design conditions, emitter temperature is 1723 K, the maximum system efficiency is 11%. Using a combination of collector preheating and recuperation leads to the same maximum performance as with a recuperator alone. A combination may lead to a smaller recuperator but an extra channel is needed rendering this solution less viable. The heat exchanging capacities of both burner and recuperator are relatively small for a near maximum system performance. Using a recuperator makes the influence of emitter emissivity on system efficiency negligible. An estimation of the size of the recuperator leads to an acceptable size of less than 0.06 m long at the same diameter as the emitter. The pressure loss of the recuperator for 1 mm channels is shown to be less than 2 ‰ of the electrical power at a pressure drop of 175 Pa.

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