Alternating flow characteristics of a regenerator under isothermal conditions

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Article Info

Article history:

Received 25/05/2022
Revised 12/12/2022
Accepted 12/12/2022/

Keyword:

Regenerator, alternating air flow, pressure drop, friction factor, phase shift

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ABSTRACT

An experimental set up have been designed and built to investigate the alternating flow characteristics of a metallic regenerator with straight channels under isothermal conditions. The operational frequency of the system was run between (0.71-4.25 Hz). Detailed experimental data were obtained for the pressure drop. It is found that the pressure drop of alternating flow in regenerator is different than that of steady state unidirectional flow. The effect of frequency and size of the regenerator on the pressure drop is presented and discussed. The current results of the friction factor were compared to other studies from the literature employing alternating air flow in similar metallic materials.

1. INTRODUCTION

A regenerator is a heat exchanger which composed of a porous material through which a fluid flows alternately. Its role is essential in the operation of motor Stirling engines and refrigerators[1,2]. During the thermodynamic cycle engine for example, in the isochoric heating phase, the gas receives heat from the solid matrix of the regenerator and during the isochoric cooling phase, it provides heat to the solid matrix. In the dynamic mode, a regenerator must quickly accumulate heat when traversed in one direction by a hot gas (which itself cools) and gives back the heat during the passage of the cold gas (which itself heats) in the other direction.

The regenerator must possess thermophysical and geometrical properties promoting rapid heat exchange between the solid and gas such as: high heat capacity, high thermal conductivity and high surface area, low pressure drop and high porosity. Some features are contradictory (low pressure drop and high surface area, for example) where the difficulty is the size dimension of a generator for different alternating flow conditions.

A Stirling engine without regenerator should absorb up to five times more heat in the hot heat exchanger to achieve the same performance as a Stirling engine having a regenerator [3]. In the Stirling engines the regenerator are passed through by an alternating flow having different characteristics of continuous flows. The velocity of an alternating flow has a zero value in each half cycle. Between these two values, it increases, reaches a maximum and decreases[4]. This has the effect, according to the classic definition, to obtain a coefficient of friction variable through two infinite theoretical values (at zero velocity) and a limit value (maximum velocity) depending on the nature of the flow (laminar or turbulent) and the geometry of the regenerator (hydraulic diameter of the pores).

Under these conditions, the friction factor can be characterized by equations corresponding to those of a steady flow. Richardson and Tyler [5]demonstrated by experimental measurements in a tube of circular cross section, the existence of an annular effect on the velocity profiles near the wall.

Different authors have proposed equations to estimate the friction coefficient in alternating flow for regenerators of various geometries and operating conditions. They reported that the friction factor of oscillating or alternating flow is different than that of steady flow (Table1). For example, Tanaka et al.[6]proposed friction factor versus Reynolds number for different materials. They found that the friction factor of oscillating flow is higher than that of steady flow. The friction factor is defined by:

$$C_f = \frac{\Delta P D_h}{\frac{1}{2} L \rho u^2} \quad (1)$$

Where ΔP is the pressure drop, D_h is the hydraulic diameter, ρ is the fluid density, L is the length of the regenerator matrix and u is the velocity of the fluid flow.

Gedeon and Wood [7]studied characteristics of oscillating air flow within manufactured regenerators made of metal felts and woven screens. They stated any significant differences between these two flows, except at high frequencies. Miya be et al.[8] found a correlation of the friction factor in steady unidirectional flow in a regenerator filled with wire screen. Zhao and Cheng [9]analyzed experimentally the pressure drop through packed column composed of three different sizes of woven screen. They found that the pressure drop could be 4 and 6 times larger than the pressure drop in steady flow computed from Miyabe's correlation.

Jin and Leong [10]made measurements permitting the study of the friction factor, the velocity of an alternating flow of the Stirling engine, inside a channel filled with aluminum foam (open-cell metal foam) under a constant heat flux for different porosity densities. The tests were conducted at low average velocity (0 m/s-3.5 m/s) under a pressure fluctuation between 0 Pa and 450 Pa and a frequency between 0.5 Hz and 9 Hz. The authors compared their experimental results with the data from experiments of Tanaka et al. [6]for a regenerator made of wire-screens. They showed that maximum pressure drop in aluminum foam that is much lower than that in a channel packed with wire- screen. The large difference of the maximum friction factor between aluminum foam and wire screen is due to difference in structure.

Table 1. Correlations for the friction factors

Authors	Regenerator geometry	Correlation	Conditions	Comments
Miyabe et al.[8]	Woven screens	$C_f = \frac{33.6}{Re_{D_l}} + 0.0337$	$C_f = \frac{\Delta P}{n} / (1/2 \rho u^2)$ $5 < Re_{D_l} = \frac{u D_l}{\text{v}} < 1000$ D_l: $mesh\ distance, n$: $number\ of\ screens$ Porosity: $0.58 < \varepsilon < 0.84$ Unidirectional steady flow	---
Tanaka et al.[6]	Wire netting, sponge metal and sintered metal	$C_f = \frac{175}{Re_h} + 1.6$	$C_f = \frac{\Delta P_{max} D_h}{\frac{1}{2} \rho u_{max}^2}$ Frequency: 1.7-10 Hz $10 < Re_h = \frac{u D_h}{\text{v}} < 4000$ D_h: $hydraulic\ diameter$ Wire netting$0.58 < \varepsilon < 0.84$ Sponge metal $0.702 < \varepsilon < 0.956$ Sintered metal $\varepsilon = 0.372$	The friction factor of alternating or oscillating flow is higher than that of steady flow.
Gedeon and Wood[7]	Wire screens Metal felts	$C_f = \frac{129}{Re_h} + 2.91 Re_h^{-0.103}$ $C_f = \frac{192}{Re_h} + 4.35 Re_h^{-0.067}$	$C_f = \frac{2 D \varepsilon \Delta P}{\rho L (1-\varepsilon) u^2}$ Frequency: 1-120 Hz Wire screens:$0.45 < Re_h < 6100$; $0.62 < \varepsilon < 0.71$ Metal felts: $0.11 < Re_h <$ 2500; $0.68 < \varepsilon < 0.84$	Any significant differences between these two flows, except at high frequencies.
Hsu et al.[11]	Wire screens	$C_f = \frac{109}{Re_h} + \frac{5}{Re_h^{0.5}} + 1$	$C_f = \frac{\Delta P_{max} D_h}{\frac{1}{2} \rho u_{max}^2}$ Frequency: 0-4 Hz $1 < Re_h < 2000$ $0.8 < \varepsilon < 0.9$	No difference if the frequency is below 4Hz.
Dellaliet al.[12]	Array of pillars of lenticular shape	$C_f = 11.88 Re_h^{-0.262}$	$C_f = \frac{\Delta P_{max} D_h}{\frac{1}{2} \rho u_{max}^2}$ Frequency: 2-10 Hz $900 < Re_h < 6000$ $0.8 < \varepsilon < 0.9$	The friction factor of alternating or oscillating flow is higher than that of steady flow.

In this paper, we report results of an experimental investigation, carried out in the Energy Department, dealing with friction factor in alternating flow. The experimental set up designed and built to produce alternating flow conditions found in many industrial applications, such as Stirling machines, compressors and internal combustion engines.

In the first part, a brief describe the exeperimental apparatus and its associated instrumentation are presented. In the second part, experimental results in pressure drop profil are presented to verify the hypothesis which consists of considering the pressure drop of oscillating flow is different than that of unidirectional or steady flow. The effects of the frequency and regenerator length on pressure drop are then presented in alternating flow. Finally, correlation of the friction factor resulting from experiments Tanaka et al. [7] is compared with our experimental results.

2. Experimental setup

2.1. Steady state unidirectional flow test bench

The test bench scheme of the regenerator in unidirectional flow is shown in Figure 1. In this working mode, the regenerator is subjected to a flow of gas (dry air) at ambient temperature T_{amb} = 24°C and under an absolute supplying pressure P_i of 2 bar generated by a system involving a compressor, a dryer and a reservoir. The volumetric flow rate Q_v is measured by a flow meter arranged upstream of the regenerator.

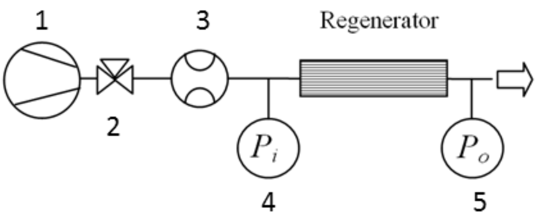

Figure 1. Schematic of experimental apparatus for a steady flow through a regenerator. 1: air compressor, 2: valve, 3: flow meter, 4: inlet pressure sensor, 5: outlet pressure sensor.

2.2. Alternating flow test bench

A schematic diagram of the experimental apparatus to perform experimentsof alternating flow interaction with a porous media is shown in figure 2. The device is similar to an alpha Stirling engine configuration. The alternating flow (maximum frequency of20 Hz) is generated by a mechanism which consists of a compression cylinder, a piston and a crankshaft with adjustable stroke lengths. The travel of the piston is set such that all of the fluid displaced passes completely through the regenerator.

The working gas alternately passes through a hot exchanger (HEX) and a cold exchanger (CEX). The regenerator (REG)is thus alternately subjected to a hot air stream and a cold air stream so that the temperature gradient $\Delta T = T_{hot\text{-}side} - T_{cold\text{-}side}$ along the regenerator is 0 °C ≤ ΔT ≤ 90 °C. The pressure (P), velocity (V) and temperature (T) of the gas are measured at the both ends of the regenerator (Figure 2).

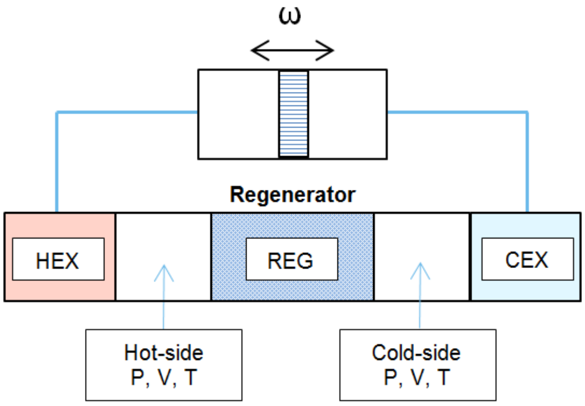

Figure 2. Experimental scheme of the alternating flow test bench. HEX: heat exchanger, CEX: cold exchanger, REG: regenerator.

Three regenerators are constituted of a porous matrix of a stainless steel 316L of a total length L= 60 mm, 70 mm and 80 mm and of an outer diameter D_{reg}= 9.5mm. Regenerator geometries were made from straight channels,porosityε = 35%, hydraulic diameter D_h= 0.237 mm and of square sections (Figure. 3).

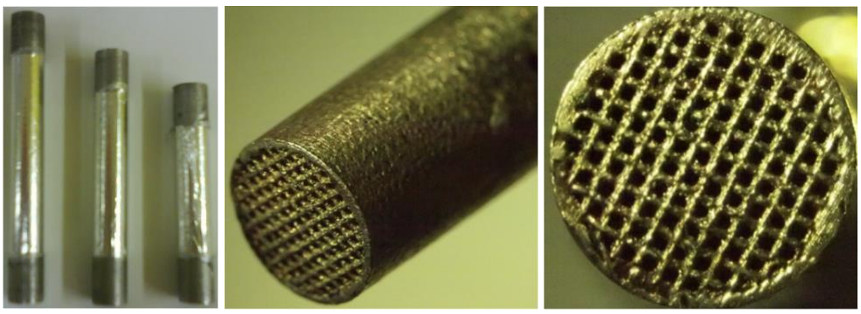

Figure 3. Photograph of a regenerator.

Local instantaneous pressures are measured using Kulite sensors model XTL-140M-5BARA. They were calibrated using a Druck PV621 Pressure Station. The fluids temperatures are measured using two homemade 12.7 µm diameter K type (Chromel-Alumel) micro-thermocouples. They were characterized under static and dynamic states[13]. The instantaneous velocity is achieved with a hot-wire sensor (TSI, Model 1201). They were calibrated by TSI IFA-300 system. Detailed information about the experimental apparatus can be found on our pervious study[14].

3. RESULTS AND ANALYSIS

In this section we present experimental results for the pressure drop ΔP across metallic regenerator subjected to analternating flow and uniderctional flow. Experiments were carried out under isothermal conditions ΔT = 0 °C.

3.1. Isothermal unidirection al flow: ΔT = 0 °C

The pressure drop by regenerator unit length at isothermal conditions for a regenerator of porosity ε=35% is plotted in Figure 4. The analysis of the curve of pressure gradientshows a classical behavior of the form $\Delta P = KQ_v^2$. This curve allows us to compare the values of pressure gradient for alternate flows in the same temperature conditions.

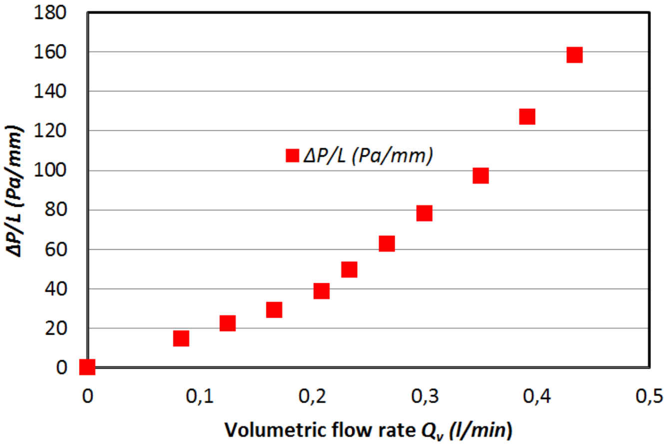

Figure 4. Variation of the pressure drop by regenerator unit length *ΔP/L* of the regenerator as a function of the volumetric flow rate Q_v for steady flow (Pi = 2 bar) and isothermal flow.

3.2. Isothermal alternating flow: ΔT = 0 °C

Different measurement points are marked in Figure2. However, in this paper we only present the results of the pressure drop and friction factor of a regenerator of porosity ε = 35% and length L = 60 mm, without study the impact of roughness on the nature of the flow. The operational frequency of the system was run between (0.71-4.25 Hz). The piston stroke was set at C = 64.4 mm.

The pressure drop by regenerator unit length at isothermal flow conditions for a regenerator is plotted versus the piston crank angleθ in Figure5. We observe a maximum value for θ=168° corresponding to half the period of the cycle for which the gas volume has been scanned by the piston. The magnitude of maximum pressure gradient for θ = 168° is $\Delta P/L$ = 588.71 Pa/mm (Figure 5).

The curves analysis of the linear pressure gradient of Figures4 and 5 shows that we are dealing with completely different phenomena. In alternating flow, velocities, pressures and temperatures vary during the time[14], which is not the case in steady flow.

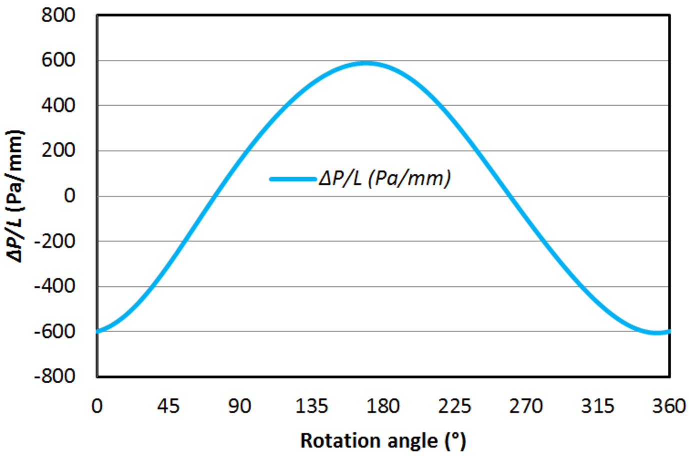

Figure 5. Pressure drop by regenerator unit length for *Fr* = 4.25 Hz and isothermal flow.

Figure 6 shows the variation of pressure drop (ΔP=P_{HEX}-P_{CEX}) versus crank angle for a regenerator of length L = 60 mm, porosity ε = 35%and frequencies 0.71, 1.23 and 4.25 Hz. The amplitude of the pressure drop increases with the increase in frequency. We have a maximum pressure drop ΔP_{max} = 0.036 MPa for a frequency Fr= 4.25 Hz. It may be noted that the attitude of the pressure gradient for the different frequency is nearly sinusoidal. This phenomenon is explained by the sinusoidal movement of the pneumatic cylinder.

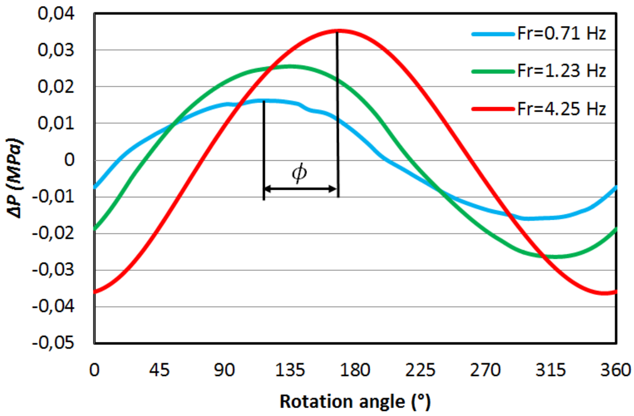

Figure 6. Pressure drop for a regenerator ε =35%, L = 60 mm, Fr= 0.71, 1.23 and 4.25 Hz. Isothermal flow.

We also observe a phase angle shiftϕ of the pressure drop as the frequency increasescompared to the cranck angle sinusoidal variation of the piston(figure7). For a frequency of 4.25 Hz, the phase shift can reach 78.51°. This phenomenon was also observed by Dellali et al.[12]in studies on the pressure drop and friction factor in a regenerator made from an array of pillars in reciprocating flow conditions. This phase shift may be due to the gas inertia.

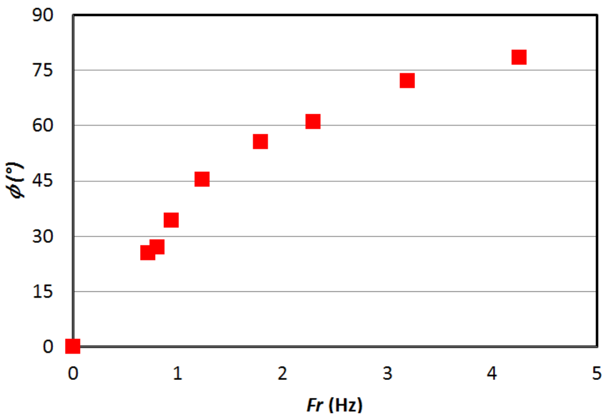

Figure 7. Maximum pressure drop phase angle shift versus frequency.

The maximum pressure drop ΔP_{max} over the cycles are plotted in figure 8a versus Reynolds number Re_h. We observe an increase in the maximum pressure drop with velocity and so with Reynolds number Re_h. The effect of the size of the regenerator on the maximum pressure drop is shown in figure8bfor a frequency Fr = 4.25 Hz. The maximum pressure drop ΔP_{max} increases with the length of the regenerator like in continuous flow. This pressure drop is created at the passage of the fluid through the porous matrix of the regenerator; it is mainly due to friction generated by the gas along the regenerator.

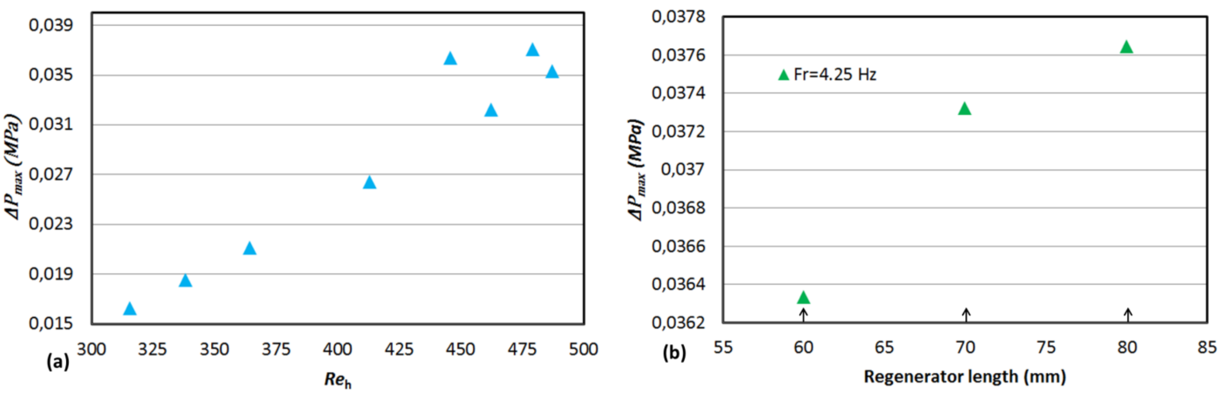

Figure 8. Maximum pressure drop over the cycles versus Reynolds number (a) and regenerator length (b).

3.2. Calculation of the friction factor

Calculating the friction coefficient is based on the following definition[6]:

$$C_f = \frac{\Delta P_{max}}{\frac{1}{2} L \rho u_{max}^2}$$

Where ΔP_{max} is the maximum pressure drop calculated in one cycle.

The maximum velocity $u_{max}$is calculated in function of the hot sidevelocityas well as the porosity ε of the regenerator:

$$u_{max} = \frac{u_{hot\ side}}{\varepsilon}$$

Figure 8 shows the comparison between the values of friction factors obtained from correlation of Tanaka et al. [6]who tested alternating air flow in similar metallic materials having nearly the same porosity with the results of the current study in the range of the Reynolds number validity for the correlation.

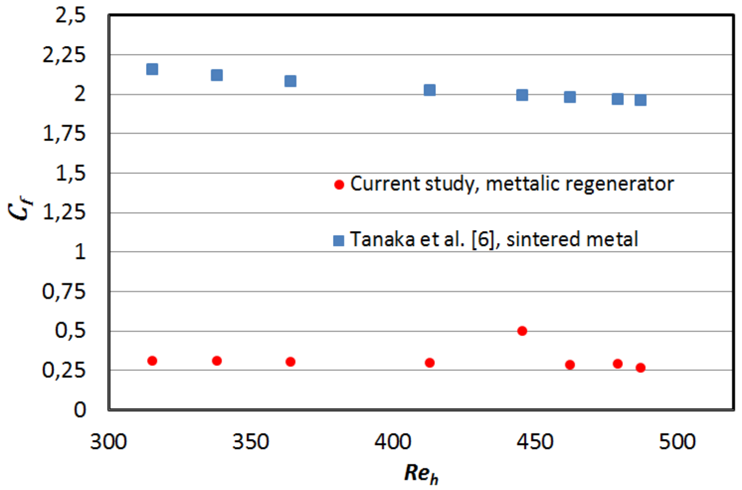

Figure 9.Comparison of friction factor for Tanaka et al.[6] and experimental data for 315.54 <Re_h< 445.76, L=60mm and ε=0.35.

We observe that the experimental values of the friction factor are less than to those of Tanaka [7], on average in the range 315.54 <Re_h< 445.76. This vast difference is probably due to the different geometries or structures of the regenerator lead to different flow fields and probably to a different transition to turbulence.

4. CONCLUSION

In this paper, we present experimental tests involving the alternating flow of air in a regenerator under isothermal conditions. We are varied the frequency of alternating of the flow and the length L of the regenerator. Alternating flow thus present the dynamic characteristics on the pressure drop different from those in steady flow. We show that the pressure drop depends on the frequency and the length of the regenerator. We observe a phase angle shift of the pressure drop as the frequency increases compared to the crank angle sinusoidal variation of the piston. This phase shift may be due to the gas inertia for the case of the alternating flow. We also observe that our experimental values of the friction factor are less than to those of Tanaka et al. [6], on average in the range 315.54 <Re_h< 445.76. These experimental results will be completed in the future by measurements on other porous materials (canvas, open metal foams, channels) and under experimental conditions. CFD numerical modeling of compressible fluid flows will be used to compare these experimental results.

ACKNOWLEDGEMENTS

This work has been supported by the EIPHI Graduate School (contract ANR-17-EURE-0002), and the Region Bourgogne-Franche-Comté, France

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