Gas Diodes for Thermoacoustic Self-circulating Heat Exchangers

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Abstract. An asymmetrical constriction in a pipe functions as an imperfect gas diode for acoustic oscillations in the gas in the pipe. One or more gas diodes in a resonant loop of pipe create substantial steady flow, which can carry substantial heat between a remote heat exchanger and a thermoacoustic or Stirling engine or refrigerator; the flow is driven directly by the oscillations in the engine or refrigerator itself. This invention gives Stirling and thermoacoustic devices unprecedented flexibility, and may lead to Stirling engines of unprecedented power. We have built two of these resonant self-circulating heat exchangers, one as a fundamental test bed and the other as a demonstration of practical levels of heat transfer. Measurements of flow and heat transfer are in factor-of-two agreement with either of two simple calculation methods. One calculation method treats the oscillating and steady flows as independent and simply superimposed, except in the gas diodes. The other method accounts for the interaction between the oscillating and steady flow with the quasi-steady approximation. The mutual influence of superimposed turbulent oscillating and steady flows is a theoretical challenge.

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INTRODUCTION

A gas diode is an asymmetrical structure that tends to favor one direction of flow over the other. Figures 1(a) and (b) illustrate a simple and common example, a pipe constriction that is abrupt on one side and gently tapered on the other. In Fig. 1(a), the flow passes through the constriction with difficulty, because separation occurs in the abruptly diverging flow and essentially all of the kinetic energy gained in the gradually converging flow is dissipated by turbulence. In Fig. 1(b), the flow passes through the constriction more easily, because separation is avoided. Gas diodes include the vortex diodes described by Mitchell [1], the valvular conduit described by Tesla [2], and the tapered structures sometimes called jet pumps [3,4]. Gas diodes are much less perfect than electronic diodes, often with the ratio of backward and forward flow impedances less than a factor of ten. Nevertheless they can partially convert an oscillating flow into a significant and useful steady flow.

Figures 1(c)-(f) illustrate gas diodes used for external heat transfer in thermoacoustic and Stirling engines and refrigerators [5-7]. Figure 1(c) illustrates the traditional approach, in which geometrically intricate heat exchangers are used to interweave the external fluid (e.g., cooling water or hot combustion products) and the working gas, bringing them into very good thermal contact with no mass exchange. A

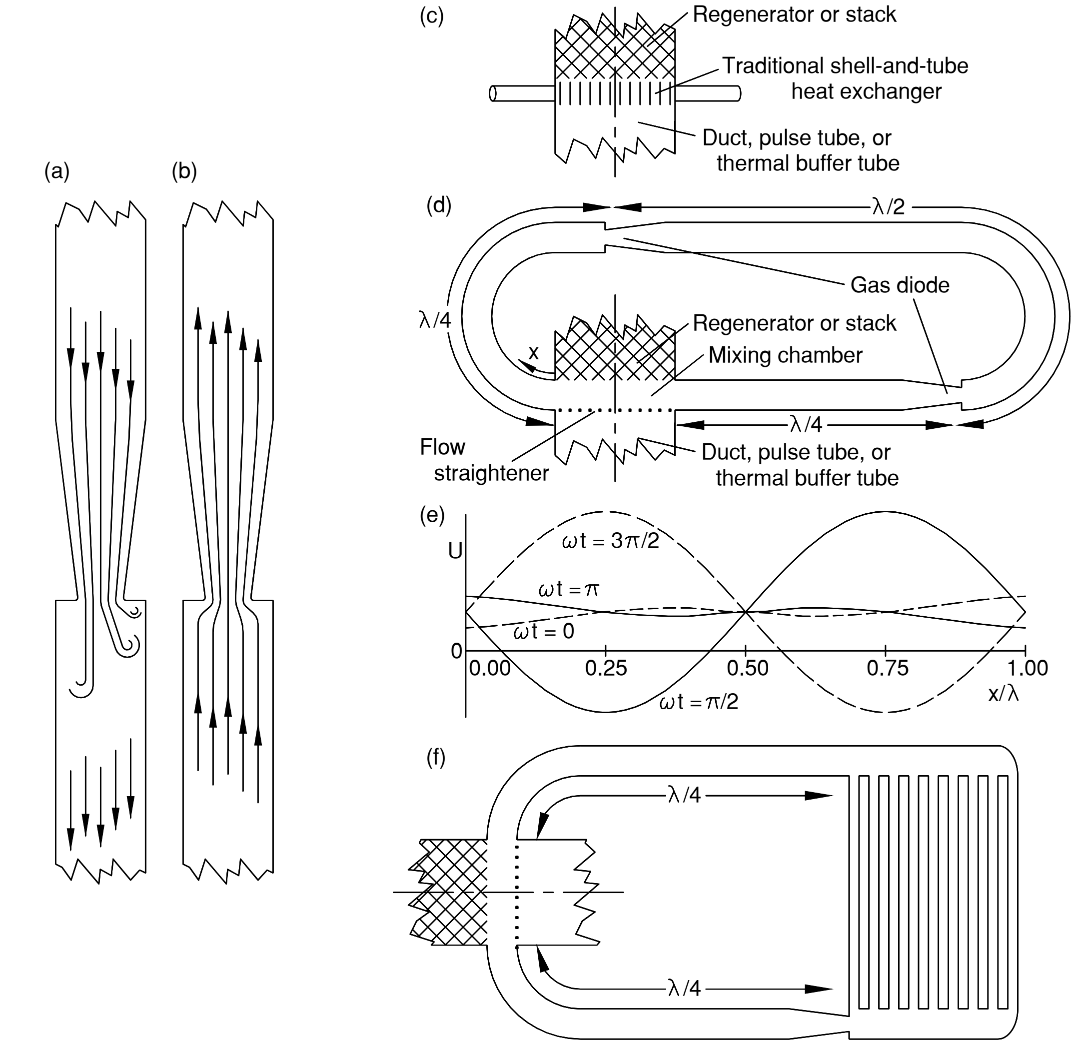


FIGURE 1. (a), (b) Flow through a gas diode. (a) High-impedance direction. (b) Low-impedance direction. (c) Part of a thermoacoustic or Stirling engine or refrigerator, with traditional heat exchanger. (d) Replacement of traditional heat exchanger with two gas diodes and external pipe one wavelength (λ) long. (e) For case (d), volumetric flow U as a function of position x at four different times t, with ω the angular frequency. (f) Alternative geometry with one gas diode, two quarter-wavelength pipes, and one heat exchanger with large surface area.

shell-and-tube heat exchanger or finned-tube heat exchanger is typical. In Fig. 1(c), the thermodynamic working gas oscillates vertically through one set of streamlined passages, while the external fluid flows horizontally through another set of passages.

The nature of such oscillating-gas engines and refrigerators imposes constraints on their traditional heat exchangers as they are scaled up to higher power. Higher power demands more heat-transfer surface area. However, passage lengths cannot be increased in the direction of oscillation of the thermodynamic working gas, because doing so increases viscous dissipation while providing no increase in the effective heat transfer area when the length exceeds the "stroke" of the oscillating gas flow. Hence, the number of passages is usually increased in proportion to the power, keeping the size of each passage constant. Such heat exchangers can have thousands of passages, causing expense and unreliability.

Thermally induced stress poses an additional challenge to reliability when such a geometrically complex heat exchanger is at an extreme temperature, i.e., a red-hot temperature for an engine or a cryogenic temperature for a refrigerator. Yet another shortcoming of the traditional heat exchangers in these engines and refrigerators is that they must be located close to one another, simply because each heat exchanger must be adjacent to one end of the nearest regenerator or stack, and these components themselves are typically short. Hence, intermediate heat-transfer loops must typically be used, adding complexity and expense.

Figures 1(d) and 1(f) show two variations of our alternative approach [8] to heat exchange for thermoacoustic and Stirling engines and refrigerators. In Fig. 1(d), the traditional heat exchanger has been replaced by a mixing chamber and one long loop of pipe. Oscillations of the gas in the loop are caused by those in the mixing chamber. Gas diodes in the loop create nonzero steady flow, so the motion of the working gas in the loop is a superposition of oscillating and steady flows. Most of the extensive outside surface area of the loop is available for thermal contact with the external fluid, which can flow either parallel or perpendicular to the pipes. The steady flow through the loop carries heat between this surface area and the mixing chamber.

The fact that the loop in Fig. 1(d) is one wavelength long leads to beneficial features, shown in Fig. 1(e). The gas diodes are at oscillating-volume-flow maxima, so they can create a large steady flow. Meanwhile, the ends of the loop are at oscillating-volume-flow minima, so the loop perturbs the oscillations in and near the mixing chamber minimally. Figure 1(e) illustrates such minimal perturbation with the pipe ends presenting a real impedance to the mixing chamber, but a slightly shorter pipe would add a positive imaginary part to that impedance, which could cancel unwanted gas compliance in and near the mixing chamber.

Figure 1(f) illustrates a half-wave option with increased heat-transfer surface area and reduced dissipation of acoustic power. The resonant enhancement of oscillating volume flow at the gas diode is still present. However, beyond the gas diode, the pipe is subdivided into many passages in parallel, with a large increase in surface and cross-sectional areas. The higher cross-sectional area causes reduced acoustic velocity and reduced oscillating pressure, decreasing viscous and thermal-hysteresis dissipation of acoustic power despite the increased surface area for heat transfer.

Relative to either Fig. 1(d) or Fig. 1(f), dissipation of acoustic power in the pipes between the mixing chamber and the gas diodes can be reduced further by proper variation of the cross-sectional area of those pipes with x, with smaller area near the gas diodes and larger area near the mixing chamber. The principles of dissipation reduction by this method are described by Hofler [9] for laminar oscillations.

Another variant of the basic idea of a resonant self-circulating heat exchanger is the use of several gas diodes in parallel where one diode is shown in Fig. 1. The use of several diodes in parallel can shorten the diode assembly while keeping the taper angle of the diodes gentle enough to avoid flow separation during the times of gradually diverging flow in the tapered parts.

In this paper, we briefly review two calculational approaches [8] and two experimental investigations [8,10] of this concept. Much more detail is in those references.

CALCULATIONAL APPROACHES

Swift and Backhaus [8] described two approaches for calculating the flows in a self-circulating loop. The "independent-flows" approach treats the oscillating and steady flows as independent and simply superimposed, except at the gas diodes where a quasi-steady approximation treats the two flows jointly. The "coupled-flows" approach applies the quasi-steady approximation to the entire loop.

In the quasi-steady approximation, steady-flow analysis is applied at each instant of time and the results are time-averaged over a full cycle of the sound wave. The quasi-steady approximation is valid if the flow at any instant of time depends only on circumstances of that instant, without memory of recent past history.

In both approaches, the quasi-steady approximation is used to couple the steady and oscillating flows at the small ends of the gas diodes. The irreversible part of the turbulent-flow pressure difference δp across any lumped-element flow impedance can be expressed using the minor-loss coefficient K via $\delta p = K\rho u^2/2$, where ρ is the gas density and u is its velocity at a reference location in the lumped element. Hence, for superimposed steady and oscillating flows, the irreversible part of the time-averaged pressure difference developed by the minor loss at the small end of a gas diode can be estimated using

$$\frac{\omega}{2\pi}\oint_0^{2\pi/\omega} \delta p(t)\, dt = \frac{\omega}{2\pi S^2}\left[\int_{t_0}^{\pi/\omega - t_0} K_+ \frac{1}{2}\rho\left(|U_1|\sin\omega t + \dot{M}/\rho\right)^2 dt - \int_{\pi/\omega - t_0}^{2\pi/\omega + t_0} K_- \frac{1}{2}\rho\left(|U_1|\sin\omega t + \dot{M}/\rho\right)^2 dt\right], \qquad (1)$$

where K_+ and K_- are the minor-loss coefficients for the two directions of flow through the diode, S is the area on which the K's are based (conventionally, the smallest cross-sectional area of the component), $\dot{M}$ is the steady mass flow, $|U_1|$ is the amplitude of the oscillating volume flow, t is time, and t_0 is the time at which the flow crosses zero. Steady-flow values of the K's are tabulated in Ref. 11.

The independent-flows approach for pressure gradients elsewhere in the loop is simple. Standard acoustics expressions are used to calculate acoustic pressure gradients in the loop without consideration of the steady flow. Independently, the time-averaged pressure gradient in the loop due to the steady flow is estimated without consideration of U_1, using standard equations of fluid mechanics.

The coupled-flows approach is much more complex. It includes the Doppler shift of the acoustic phenomena by the steady flow and the influence of the oscillating flow and steady flow on each other via turbulence, which is due to the nonlinear nature of turbulent flow resistance. Where flow resistance is described by $\delta p(t)=R[U(t)]^n$ with $n \neq 1$, the extra pressure difference caused by an increment $+\varepsilon$ of mass flow above average is not canceled when an equal decrement $-\varepsilon$ below average occurs half a cycle later, so the time-averaged pressure gradient depends on both the oscillating and steady flows. Similarly, steady flow shifts the oscillating flow into a regime of higher flow impedance, increasing the acoustic pressure gradient.

Reference 8 gives closed-form results for both calculational approaches.

EXPERIMENTS

We have experimented with two self-circulating loops similar to Figs. 1(d) and 1(f).

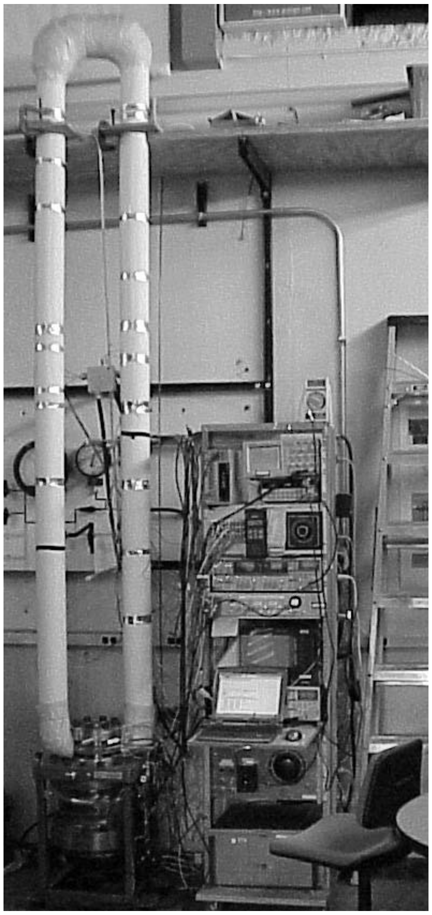
The full-wave loop shown in Fig. 2(a) used 2.4 MPa argon near and slightly above ambient temperature and was piston driven near 50 Hz. The loop had a total length of 6.33 m and was made mostly of stainless-steel pipes with an inside diameter of 2.21 cm. Two gas diodes were located as shown in Fig. 1(d). Heat was applied to the loop with electric heater tapes wrapped around the four straight parts of the pipes between elbows and diodes. In operation, the gas circulating around the loop absorbed heat from the electric heaters, causing the temperature of the gas to rise from one end of the loop to the other, and delivered the heat to the mixing chamber. Further details are given in Ref. 8.

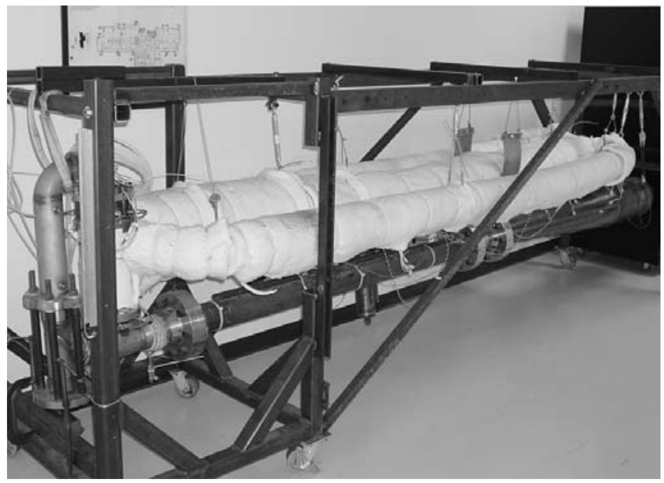
Figure 3(a), adapted from Ref. 8, shows temperature as a function of position around the full-wavelength loop of Fig. 2(a), with a wave amplitude of 190 kPa in the mixing chamber, for four different rates of heating. Filled symbols are measurements on the tube wall, and open symbols are measurements in the gas in the mixing chamber. The four heated regions are indicated by the four sloping straight-line segments in Fig. 4, which show the calculated gas temperature as a function of x for one rate of heating. The deviation of the erect triangles from the line segments is due to velocity-dependent heat transfer.

Figure 2(b) shows a thermoacoustic-Stirling hybrid engine [3] that was modified by replacing its traditional hot heat exchanger with a half-wave loop similar to that of Fig. 1(f). The loop consisted of two 2.2-cm diameter, 6.5-meter long (~ ¼ wavelength) pipes connected by a 10-liter tank to mimic the volume of the extended heat exchanger shown in Fig. 1(f). Approximately 2/3 of the way from the mixing chamber to the tank, each pipe turned 180° [at the right edge of Fig. 2(b)], so the tank was between the pipes. The straight sections of the pipe were heated by electric tube furnaces, and all was surrounded by the white insulation visible in Fig. 2(b). Detailed drawings of the loop and its interface to the engine can be found in Ref. 10. The engine and loop were filled with 3.1-MPa helium gas and oscillated spontaneously at 80 Hz.

In steady operation, the pressure oscillations in the engine drove the oscillations in the loop, so the gas diode created a steady flow around the loop to carry the heat from the tube furnaces to the mixing chamber near the hot end of the engine's regenerator. The oscillating flow through the regenerator drew the heated steady flow into the regenerator, so the engine could maintain the pressure oscillations.

Figure 3(b), adapted from Ref. 10, shows the temperature of the steady flow entering and leaving the mixing chamber vs the regenerator's hot-face temperature. The temperature of the steady flow leaving the mixing chamber matched the regenerator hot-face temperature to within 10°C (perfect equality is indicated by the straight line) while the heat delivered by the steady flow varied over a factor of four and the difference between the inlet and outlet temperatures ranged from 45°C to 100°C. A simple explanation of the interaction between the steady cross flow from the loop and the oscillating flow through the regenerator provides qualitative understanding of this plot. The steady flow is too small to completely flush out the mixing chamber volume in each acoustic cycle, so all of the incoming steady flow is drawn into the regenerator at least once before it leaves again. Thus, the excellent heat transfer in the regenerator assures that the flow bound for its next trip around the loop begins at the regenerator hot-face temperature.





← (a) (b) ↑

FIGURE 2. (a) Full-wavelength and (b) half-wavelength apparatus. White insulation covers both loops.

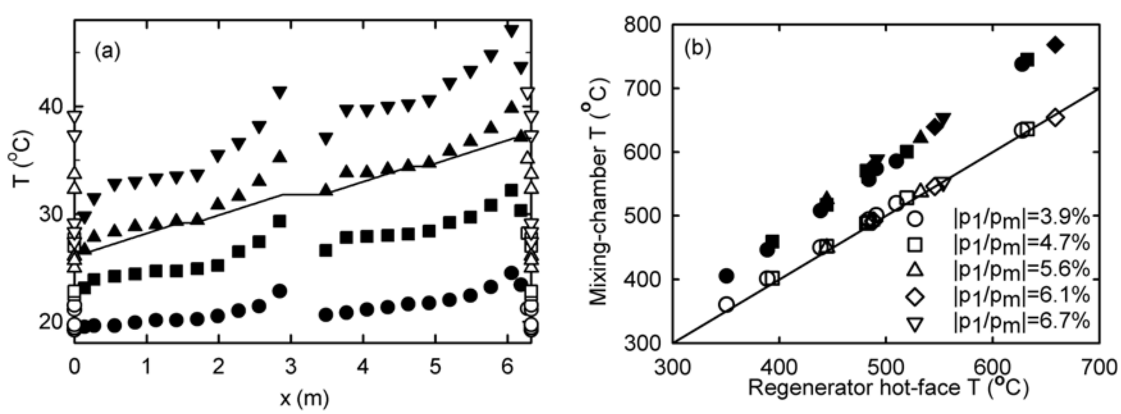


FIGURE 3. (a) Temperature T as a function of position x around the loop of Fig. 2(a), for four different heater powers. (b) Temperature of the steady flow entering (filled symbols) and leaving (open symbols) the mixing chamber of Fig. 2(b), as a function of the regenerator hot-face temperature.

COMPARING CALCULATIONS AND MEASUREMENTS

Figure 4, adapted from Ref. 8, shows some measurements with the full-wave loop of Fig. 2(a), and corresponding calculations.

In Fig. 4(a), the symbols represent measurements of the oscillating pressure at nine locations around the loop ($x = 0$ and $x = 6.33$ m being the same location—the mixing chamber). The solid curves in Fig. 4(a) represent calculations with the independent-flows approach, and the dashed curves represent calculations with the coupled-flows approach. The Doppler effect is largely responsible for the measured values of Im[p_1], and the coupled-flows approach accounts for it well. Figure 4(b) shows that the two calculational approaches give very nearly the same result for $|U_1|$ at the gas diodes, so the Doppler effect in the loop has little effect on the pumping strength of the diodes.

Figure 4(c) shows the steady mass flow around the loop as a function of pressure amplitude in the mixing chamber. The open and filled symbols represent steady flows inferred from data such as those in Fig. 3(a), based on two slightly different ways of using the data to obtain the net temperature gain as the flow goes around the loop. The circles represent measurements with the loop average temperature in the range 43–47 °C and the temperature rise around the loop in the range 13–17 °C. The triangles represent measurements with higher heats and temperatures, with the loop average temperature in the range 57–60 °C and the temperature rise around the loop in the range 20–24 °C. The highest measured mass flow is 50 gm/s. The total mass of argon in the loop is only 95 gm, so the steady flow clears the entire loop every two seconds at the highest acoustic amplitude.

The solid line in Fig. 4(c) is based on the independent-flows approach, and the dashed line is based on the coupled-flows approach. The independent-flow calculation overestimates the steady flow, so it must overestimate the strength of the diodes or underestimate the steady-flow resistance of the rest of the loop. The coupled-flows approach underestimates the steady flow, so it is likely that it overestimates the effect of the oscillations on the resistance to steady flow.

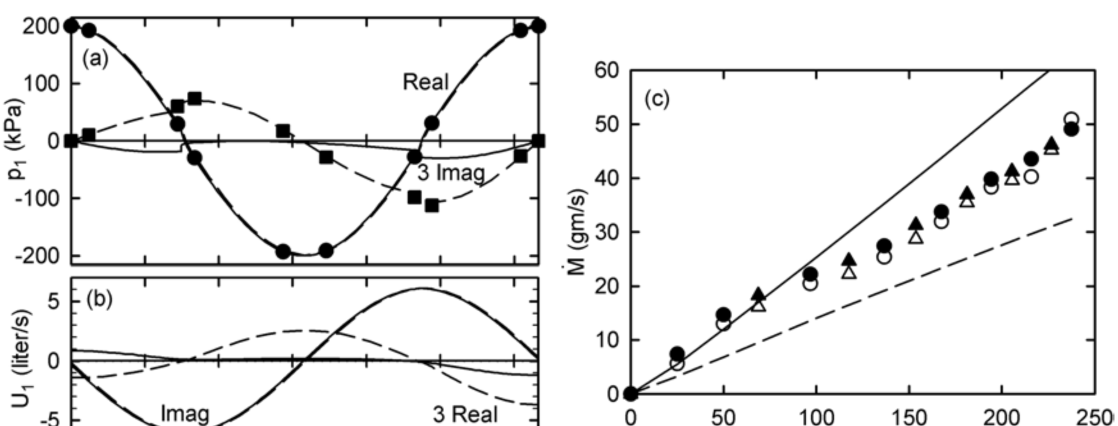


FIGURE 4. (a) Pressure and (b) volume flow in the one-wavelength loop. (c) Mass flow around loop.

In the diodes, for the conditions of Fig. 4, the maximum Reynolds number is 10^6, but the gas displacement amplitude is only about 10 times the gas-diodes' small diameter, perhaps not large enough to justify confidence in the quasi-steady approximation.

CONCLUSIONS

Gas diodes in a resonant loop of piping can cause a substantial steady flow, which in turn can carry substantial heat. They offer unprecedented flexibility in the design of Stirling and thermoacoustic engines and refrigerators.

The steady flow created by gas diodes in a loop can be measured by its effect on the acoustic pressure wave and by the heat that it carries. Two calculational approaches—"independent flows" and "coupled flows" —yield results in *rough* agreement with measurements, but neither approach is in accurate agreement with measurements of steady mass flow. Whether the disagreement is due to shortcomings in the understanding of the gas diodes, the flow in the pipe, or both is not known.

The mutual influence of turbulent steady and oscillating flows in pipes is a significant theoretical challenge; there, the quasi-steady approximation is clearly inadequate for calculating the time-averaged pressure gradient and the heat-transfer coefficient.

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