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Improved trace gas flux estimation through IRGA sampling optimization

R.J. Clement [a,*], G.G. Burba [b], A. Grelle [c], D.J. Anderson [b], J.B. Moncrieff [a]

[a] School of GeoSciences, The University of Edinburgh, Edinburgh EH9 3JN, Scotland, United Kingdom
[b] LI-COR Biosciences, Lincoln, NE 68504, USA
[c] Department of Ecology and Environmental Research, Swedish University of Agricultural Sciences, Uppsala, Sweden

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ABSTRACT

We examine the theoretical and practical aspects of improving the sampling methods of spectroscopic trace gas sensors of Eddy covariance flux measurement systems. Theory is developed based on non-ideal ventilation devices and existing equations for tube flow and attenuation of non-reactive trace gases and temperature. Model results indicate an optimum design exists which can be expressed in relation to intake tube diameter and which depends upon the ventilation device employed. Field experiment results (employing modified open path IRGAs) show that the use of short intake tubes can reduce flux losses by trace gas signal attenuation while minimizing the adjustments required for density fluctuations, with additional benefits of increased data capture under adverse environmental conditions.

1. Introduction

The use of spectroscopic methods for environmental measurement of trace gases has expanded greatly over the past two decades. Improvements in sensor design and stability have made this sensor type a keystone in the deployment of Eddy covariance measurements for the determination of ecosystem trace gas exchanges. Trace gas sensors can be classified as being either closed path or open path sensors. The continuing use of both configurations points to the presence of positive and negative aspects of each.

A closed path sensor must have its air sample transferred from the point of collection to the point of measurement. The time required to transport the air sample allows for manipulation of the sample—such as elimination of fluctuations in temperature, pressure and water vapour, or the removal of unwanted constituents such as particulates, all of which can affect the accuracy of the desired measurement. Unfortunately, the process of transport to the sample cell also degrades the quality of the signal we wish to measure. This degradation is commonly observed as an attenuation of the higher frequency fluctuations in the measured signal. Description of, and corrections for, this effect have received extensive coverage in the literature (Philip, 1963; Massman, 1991; Lenschow and Raupach, 1990; Leuning and Moncrieff, 1990).

* Corresponding author.
E-mail address: robert.clement@ed.ac.uk (R.J. Clement).

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As their name implies, open path trace gas sensors make their measurements in situ using an exposed sample cell. This configuration eliminates high frequency sample signal loss caused by tube attenuation but also makes sample modification impossible. The inability to modify the air sample results in the need to adjust fluxes for air density fluctuations, caused primarily by fluctuations in air temperature and humidity (Webb et al., 1980). This adjustment exposes a potential source of inaccuracy. Open path sensors may also incur a considerable amount of data loss when deployed in environments with frequent precipitation or other large particulates impinging on the sensing path.

Of a more practical concern for closed path systems is the additional power required to transport the air sample to the sensing cell. This transportation can be from a few, to tens of meters and, because of flow resistance, may require a large proportion of the power needed by the Eddy covariance system. This fact may easily sway the choice of an open or closed path sensor for experiments in which available power is a contributing factor to the quality and quantity of data that is collected.

In response to this dilemma a compromise is needed. A closed path sensor with low power requirements, minimal attenuation of trace gas species and strong attenuation of temperature is needed. The authors examine theoretical and experimental evidence for such a system and present the results of this analysis.

2. Optimization of sample intake design

The objective of this theoretical analysis is to ascertain if there exists an optimum in the relationship of the minimization of both power consumption and trace gas signal attenuation (CO_2, H_2O) to the maximization of temperature signal attenuation. We attempt to determine, for a given ventilator, the optimum intake dimension to maximize the ratio of temperature to CO_2 attenuation. Because of the nature of the component equations, we do not attempt to define the true attenuation ratios but only a response to variations in tube diameter.

To obtain this optimized configuration we employ three sets of equations: a model for attenuation of scalars flowing through a tube, a model for attenuation of temperature flowing through a tube and a model for tube flow created by a mechanical ventilator. The equations for scalar tube attenuation transfer functions are those of Leuning and Moncrieff (1990) for laminar flow and of Massman (1991) for turbulent flow. For temperature we employ the 'damping distance' equation of Rannik et al. (1997).

In this analysis we first determine the relationship between tube diameter and flow rate for non-ideal ventilation devices. Using this relationship we then establish the rates of attenuation for temperature and CO_2 and then examine the relationship between these two rates of attenuation.

Devices used to ventilate an enclosure are less efficient when working against a flow resistance. This flow resistance can be observed as a pressure drop between the inlet and outlet of the flow system. The flow rate, Q, of most ventilation devices can be approximated as a near linear relationship to the pressure drop, ΔP, caused by flow resistance:

$$Q = a + b\Delta P \tag{1}$$

In this analysis, we have obtained coefficients a and b for two example ventilation devices, an axial fan ($a = 645.6\ \mathrm{l\,min^{-1}}$, $b = -1.8807\ \mathrm{l\,min^{-1}\,Pa^{-1}}$ (D484, Micronel, USA)) and a diaphragm pump ($a = 17\ \mathrm{l\,min^{-1}}$, $b = -0.000224\ \mathrm{l\,min^{-1}\,Pa^{-1}}$ (Capex 2V, Charles Austen Pumps, UK)).

The pressure drop caused by flow through the tube can be defined as a function of Q tube diameter, d tube length, L, air density, ρ, and a friction coefficient, f Duncan et al. (1960):

$$\Delta P = \frac{8\rho Q^2}{\pi^2 d^4}\frac{fL}{d} \tag{2}$$

The value of f is typically a discontinuous function of Reynolds number, Re. For smooth walled tubes, f may be defined empirically as (Landau and Lifshitz, 1959):

$$f = \left| \begin{array}{l} \dfrac{64}{Re}, \quad Re < 3000 \\ \dfrac{1}{\sqrt{\lambda}} = 0.85\log\left(Re\sqrt{\lambda}\right) - 0.55, \quad Re > 3000 \end{array} \right. \tag{3}$$

For our purposes f was approximated using a power function:

$$f = 0.505Re^{-0.384} \tag{4}$$

This assumption introduces inaccuracy in f at low Reynolds numbers (<2000). This assumption is justified on the basis that the results of this exercise are governed by turbulent flow conditions, under which the errors for the assumptions surrounding the value of f are small (<10%) and consistent with respect to tube diameter.

The Reynolds number, which defines the state of turbulence of the flow, is calculated using d, ρ, flow velocity, U, and the kinematic viscosity of air, μ ($0.017\ \mathrm{g\,m^{-1}\,s^{-1}}$).

$$Re = \frac{\rho d U}{\mu} \tag{5}$$

The mean velocity of air flowing through the sampling tube, U, can in turn be defined in terms of the tube diameter d and flow rate Q, Duncan et al. (1960):

$$U = \frac{Q}{\pi(d/2)^2} \tag{6}$$

The relationship of flow rate on pressure drop produces an interdependency of Eqs. (1)–(6). By working backwards from Eq. (6) and making substitutions we can remove ΔP from Eq. (1) and state it in terms of flow rate Q and tube diameter d:

$$Q = a + BQ^{1.616}d^{-4.616} \tag{7}$$

where the coefficient B is defined as

$$B = 0.154bL\rho^{0.616}\mu^{0.384} \tag{8}$$

The value of B is approximately constant if the pressure drop is not so severe as to strongly affect air density. Again this assumption is reasonable for the larger tube diameters and more turbulent flow conditions that govern the results of this exercise.