v. 2, n. 15, 2022

Journal of
Engineering Research

TUNING OF CONTROLLERS IN HEAT EXCHANGERS UNDER THE INFLUENCE OF FOULING

Carolina Borges de Carvalho
Department of Chemical Engineering, State University of Maringá
Maringá - Brazil

Esdras Penêdo de Carvalho
Department of Mathematics, State University of Maringá
Maringá - Brazil

Mauro Antônio da Silva Sá Ravagnani
Department of Chemical Engineering, State University of Maringá
Maringá - Brazil

All content in this magazine is licensed under a Creative Commons Attribution License. Attribution-Non-Commercial-Non-Derivatives 4.0 International (CC BY-NC-ND 4.0).

Abstract: The prediction of important control properties is a challenging task for the design of shell and tube heat exchangers. Dynamic models, which include fouling effects, are still poorly investigated in recent literature. To study the behavior of variable conditions in the team and its influence on the controller adjustment, the model considered in this work is based on the idea of heat exchange cells as basic modeling elements (lumped-parameter model). This type of approach has some advantages over the distributed-parameter model, such as continuous variables in time and discrete in space, leading to ordinary differential equations (ODE) and also providing the possibility to control the complexity of the model by adjusting the number of modeling cells. The influence of the scale on the process control is evaluated considering the thermal resistance of the scale as a function of time. The model was implemented in Simulink/MATLAB and simulations were performed for different operating periods. The results show that the periodic adjustment of the PID parameters is necessary to keep the system quality indicators stable.

Keywords: Dynamic modeling, parameter adjustment, embedding.

INTRODUCTION

The accumulation of scale on the surfaces of heat exchangers (TC) affects almost all chemical plants, introducing additional costs that are mainly related to energy conservation (burning extra fuel to overcome the effects of fouling), operation (loss of production due to planned or unplanned cleanup interventions) and capital investments (oversized equipment). In addition to increasing the resistance to heat transfer and therefore lead to a reduction in the heat exchange capacity of an existing heat exchanger, scale deposition also affects the constraint flow, increasing the fluid velocity and, thus, increasing the operating pressure

Some fouling resistance values (R_f) based on industrial data are provided since 1950 by a compilation published by the Tubular Exchangers Manufacturers Association (TEMA)1 and these values are still the basis for the design of most heat exchangers around the world. world.[2] However, the uncritical use of the fixed fouling resistors provided by TEMA leads to several problems that are mainly related to the dynamic operation of the equipment.

Fouling is a time-dependent phenomenon and changes in the dynamics of heat exchangers can lead to detrimental effects on the overall process. [3] if not properly controlled. Most of the studies are carried out for heat exchangers operating in stationary behavior, focusing on heat recovery in non-dynamic situations. Only a few works focus on transient models for heat exchangers that incorporate fouling resistance as a time-dependent variable..[4]

According to TEMA,[1] five types of fouling mechanisms are recognized, and in most cases the mechanisms do not occur in isolation, increasing the complexity of the model. Are they:

1. Precipitation: it occurs mainly due to the crystallization of the fluid with supersaturated solutions.

2. Particulates: particles of salt, sand or other chemical products are initially suspended in the stream, being later sedimented on the heat exchange surface.

3. Deposition by chemical reaction: as a function of temperature and facilitating agents, certain chemical reactions can be favored.

4. Corrosion: accumulation of iron oxide altering the thermal conductivity of the heat exchanger construction material.

5. Biological deposition: growth of organic material that can occur due to the use of water from rivers, lakes or oceans that have not been adequately treated for operation in heat exchangers .

As presented by, [3] about 80% of industrial heat exchangers are controlled by PID controllers, mainly due to their simplicity and robustness. However, in the vast majority of cases, there is no systematic process of tuning the parameters of the same, which is performed manually or following some general protocols provided by the manufacturer.

Therefore, this work aims to study a more sophisticated and robust method for tuning PID controllers, as Rf varies with the team. For this, a lumped parameters model was implemented to describe the heat exchanger. This approach was initially studied by[5] and proved to be adequate for dynamic studies, being widely used in the literature until the present day.

In the model, the heat exchanger is divided into modeling cells that can be defined as perfectly stirred tanks that only exchange heat among themselves.

This way, the thermo-physical properties in each cell are considered constant. Thus, for each cell, an energy balance is made on the side of the hull, tube and heat exchange wall only in the team domain, obtaining three ordinary differential equations (ODE) per cell. The model complexity is controlled by the number of modeling cells.

In order to obtain equipment as close as possible to the operational reality, a model proposed by,[6]in which a mixed integer nonlinear programming problem (NPLMI) subject to the mechanical constraints of TEMA is formulated. Thus, the construction standard adequate to the current norms is reached.

METHODOLOGY

HEAT EXCHANGER PROJECT

The model developed by [6] was adapted to minimize the total area of the equipment. Since the scope of the present work is to study the dynamic behavior of the heat exchanger, the objective function containing total operating costs, as presented by, 6 did not prove to be the option. to the most suitable. Furthermore, the decision to allocate the hot and cold currents in the hull or tubes is made in advance by the user, in order to maintain the characteristics of the original problem.

The classic Bell-Delaware method is used to formulate the mathematical model involving discrete and continuous variables (PNLMI). All the mechanical parameters of the heat exchanger considered by the TEMA standards are allocated in a table in a GAMS environment containing 565 lines. As a result, the proposed model presents non-linear and non-convex characteristics.

Several algorithms are available for solving this class of problems, however, due to its combinatorial nature, a branch and bound algorithm is more suitable computationally.[6] The method will create a search tree and test only a part of the feasible solutions to the problem.

PID CONTROLLER

Let the general closed-loop system below:

Considerando que $e(t)$ seja a diferença entre o ponto de ajuste desejado (*setpoint*) e a variável de processo medida (erro) e $u(t)$ é a saída do controlador, a forma paralela do algoritmo PID é dada por:

$$u(t) = K_p e(t) + \tau_i \int_0^t e(t)dt + \tau_d \frac{de(t)}{dt} \tag{1}$$

Applying the Laplace transform, Equation (1) can be modified for the frequency domain. The relationship between the output and the input signals leads to the transfer functions of the process.

$$G(s) = \frac{U(s)}{E(s)} = K_p + \frac{\tau_i}{s} + \tau_d s \tag{2}$$

The transfer functions are important for characterizing the dynamic process, as they establish a relationship between the dependent (output) and independent (input) variation.

For the considered system (heat exchanger), the transfer functions are obtained from the energy balances in each cell, as will be shown in the next section.

DYNAMIC SIMULATION

For each modeling cell, an energy balance is performed on the hull (subscript s), tubes (subscript: t) and wall (w). From the principles of Thermodynamics and the perfect mixing considerations, the energy balances obtained for each cell are:

$$\rho_s V_s C_{ps} \frac{dT_{so}}{dt} = M_s Cp_s(T_{si} - T_{so}) + h_{fs} n\pi D_2 l(T_{sw} - T_{so}) \tag{3}$$

$$\rho_t V_t C_{pt} \frac{dT_{to}}{dt} = M_t Cp_t(T_{ti} - T_{to}) + h_{ft} n\pi D_1 l(T_{tw} - T_{to}) \tag{4}$$

$$\rho_w V_w C_{pw} \frac{dT_w}{dt} = h_{ft} n\pi D_1 l(T_{to} - T_{tw}) + h_{fs} n\pi D_2 l(T_{so} - T_{sw}) \tag{5}$$

In Equations (3 - 5), the subscripts o and i refer to the output and input conditions, respectively. M is the mass flow rate of the currents, l is the length of the cell, n is the number of passes in the tubes, V represents the volume of each cell, D1 and D2 are to the inner diameters of the tubes and the tube bundle, respectively.

In order to include the values of thermal resistance due to deposition (R_f) in the heat transfer coefficient (h), the following formulations were used for the tube and housing side, respectively:

$$h_{ft} = \frac{h_t}{h_t R_f + 1} \tag{6}$$

$$h_{fs} = \frac{h_s}{h_s R_f + 1} \tag{7}$$

RESULTS

The presented methodology was applied to a heat exchanger of salt water and methanol, extracted from [7] as the current data are presented in 1.

Current	Location	$\dot{m}$ (kg/s)	$T_i(^\circ C)$	$T_o(^\circ C)$
Methanol	Hull	27,80	95,0	40,0
Salt water	Pipe	68,90	25,0	40,0

Table 1: process data

Source: Adapted from. [7]

For this case, the total thermal load of the equipment is of Q = 4.29MW and the effective logarithmic mean of temperature difference (MLDT) is of 24,89°C.

The mechanical design of the exchanger was carried out according to the method presented by [6], in which the model is formulated using generalized disjunctive programming (PDG) and the optimization is done using nonlinear mixed integer programming (NLPMI). The proposed model strictly follows the TEMA standards,[1] that is, all geometric features are standardized.

The physical properties of fluids (density (ρ), heat capacity (C_p), viscosity (μ) and thermal conductivity (k) are shown in Table 2.

Table 3 presents the main geometric data obtained by the model through the GAMS

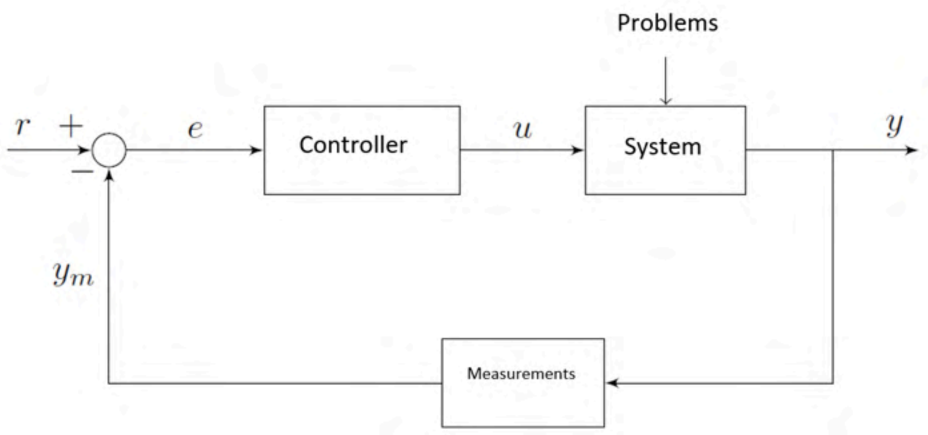

Figure 1: Control loop:PID

Property	Methanol	ASalt water
$\rho(kg/m^3)$	750	995
$C_p(kJ/kg.K)$	2,84	4,20
$\mu(Pa.s)$	0,00034	0,00080
$k(W/m.K)$	0,19	0,59

Table 2: Physical properties of currents

Source: Adapted from.[7]

Project variable	Value obtained
Heat exchange area	272,36 m^2
Tube length	2,438 m
U_s	632,57 $W/m^2.K$
Number of passes in the tubes	2
Amount of chicanery	5
Number of tubes	1400
Tube inner diameter	23 mm
Tube bundle diameter	1,372 m
External hull diameter	1,422 m
Pitch	32 mm

Table 3: Mechanical configuration obtained for the exchanger

software. The maximum allowable pressure drop value of $\Delta P = 68,\ 5kPa$ for both shell and tube sides.

In,[7] the authors obtained a heat exchange area of 243.2m2 for this case, a value 12% lower than that obtained by this work.

The goal of the authors was to minimize the total annual cost of the heat exchanger and not purely the heat exchange area, as discussed in the present work. Furthermore, the exchanger obtained by this model strictly follows the TEMA norms, a fact that does not occur in the compared case.

For the steady-state simulation, the computational package was used: *Aspen Exchanger Design & Rating* (Aspen EDR) of the *software* ASPEN PLUS. Through the simulator, it is possible to find the temperature distribution of the fluids along the hull and tubes (Figure 2) as well as the variation of the physical properties (density, heat capacity, viscosity and conductivity t ‘ermica) at each point of the heat exchanger. The values obtained by the simulation will compose a database for each modeling cell.

With the geometric parameters and thermo-hydraulic properties defined, the transfer functions were generated. The number of modeling cells recommended by [8] for shell and tube heat exchangers is given by:

$$N_c = (N_b + 1)N_p, \qquad (8)$$

in which N_c and the number of cells, N_b represents the number of baffles and N_p ánd the amount of passage through the tubes. For this case, a model with 12 modeling cells was used.

Figure 3 shows the step perturbation response with the tuned controller based on the start of operation.

Table 4 shows the ascent team (t_r), the accommodation team (t_a) and the maximum percentage overshoot, or *overshoot*, (M_p) of the transient response of Figure 3.

Note that the overshoot reaches a value greater than 30% when the fouling factor is at its maximum value, which may not meet the requirements of many processes. The dynamic responses obtained in the different operating periods show that the accumulation of fouling and, consequently, the increase in the resistance to thermal exchange caused by the layer of deposits formed, causes a deterioration of the controller’s performance indicators.

Then, the tuning parameters were adjusted for each value of R_f tested, seeking to compensate for the effects caused by the accumulation of fouling on the heat exchange surface. Figure 4 shows that the controller tuning was able to provide a dynamic response much closer to the initial behavior, eliminating possible adverse effects of fouling in process control.

As it can be seen in Table 5, there were significant changes in the controller gains, showing that periodic adjustments are necessary to ensure the quality of the control system (Figures 4 and 6).

The tuning method employed by Simulink automatically adjusts the PID gains in order to seek a balance between performance and robustness. However, apart from specifying the controller type (P, PI or PID), it is not possible to determine any process requirements., and the designer is responsible for fine-tuning the controller to the process based on experience (trial and error).[9]

This way, a more advanced method of optimizing the controller tuning is needed that minimally depends on user interference. Considering that the process requirements (problem constraints) are $t_r = 14,\ 5s$, $t_a = 75s$ (for an oscillation of ±3%) and $M_p = 12\%$, it is possible to use a numerical algorithm based on Sequential Quadratic Programming (PQS) through the Simulink *Control Design* to obtain

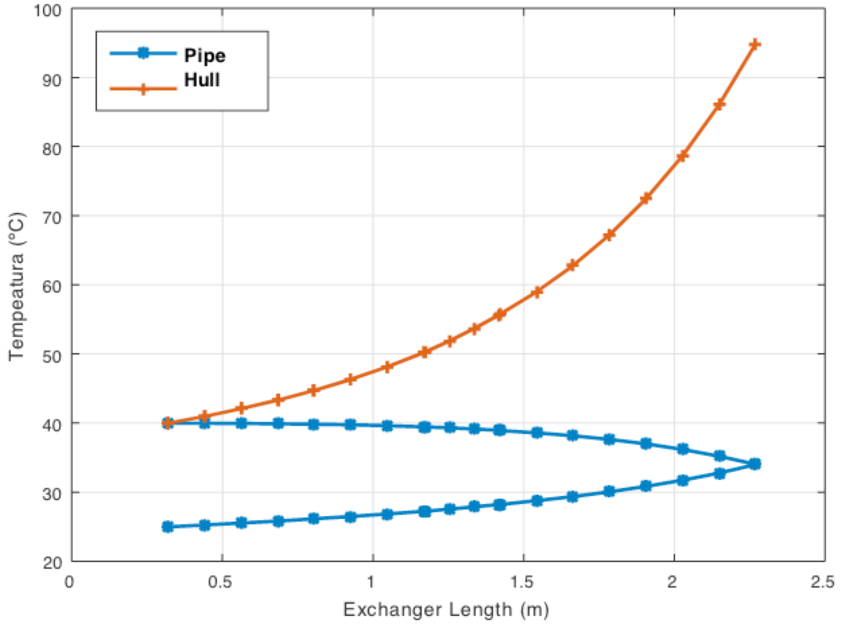

Figure 2: Temperature Distribution

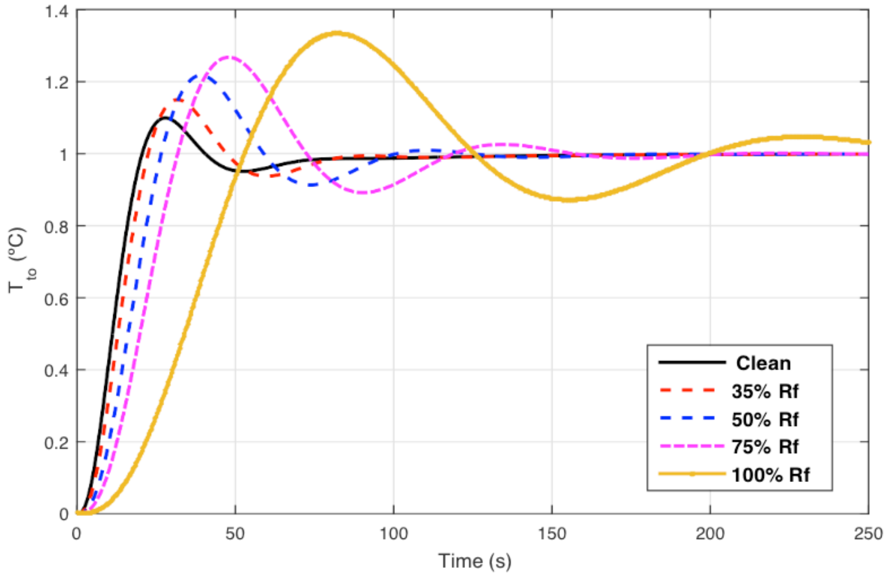

Figure 3: Response to step disturbance before setting controller parameters

Condition,	$t_r(s)$	$t_a(s)$	M_p
Clean	12,8	70,5	10,9%
35% R_f	14,2	78,0	15,1%
50% R_f	16,7	94,1	21,7%
75% R_f	19,7	143	26,8%
100% R_f	32,5	257	33,4%

Table 4: Transient response specifications

Condition	$R_f(m^2K/W)$	K_p	τ_i	τ_d
Clean	$5,3.10^{-4}$	9,609	0,251	3,755
35% R_f	$8,8.10^{-4}$	9,660	0,216	5,276
50% R_f	$1,3.10^{-3}$	10,196	0,206	1,212
75% R_f	$1,9.10^{-3}$	12,693	0,226	1,147
100% R_f	$2,5.10^{-3}$	13,099	0,326	13,040

Table 5: Values of R_f (total) and controller parameters

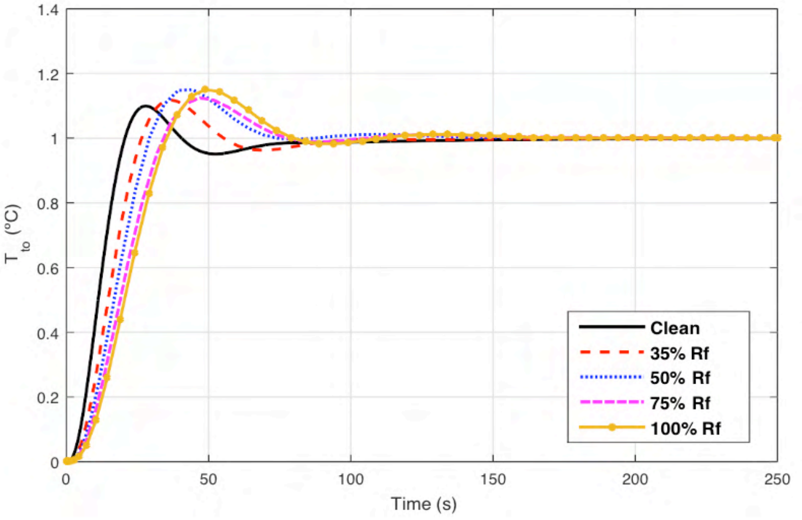

Figure 4: Response to step disturbance after adjusting controller parameters

Condition	$t_r(s)$	$t_a(s)$	M_p
Clean	12,8	70,5	10,9%
35% R_f	16,5	83,8	11,7%
50% R_f	18,4	66,4	15,3%
75% R_f	21,5	74,8	15,0%
100% R_f	21,9	71,4	12,5%

Table 6: Specifications of transient response after tuning

Condition	K_p	τ_i	τ_d	$t_r(s)$	$t_a(s)$	M_p
Clean	15,602	0,203	1,652	14,0	73,5	11,9%
35% R_f	18,103	0,212	2,709	14,1	74,4	12%
50% R_f	22,454	0,228	8,602	14,1	75,0	12%
75% R_f	24,391	0,264	12,765	14,1	75,0	12%
100% R_f	25,231	0,254	13,981	14,1	74,8	12%

Table 7: Optimal controller parameters and performance indices

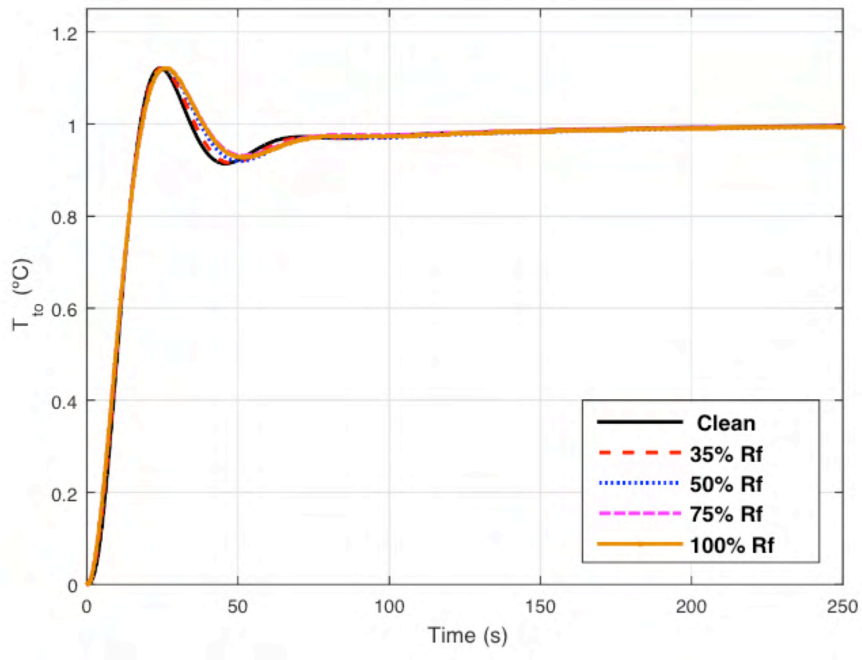

Figure 5: Response to step perturbation with optimized parameters

the optimal values of the gains Kp, τi and τd (variables to be optimized).

The formulated optimization problem seeks to minimize the variations beyond the boundaries stipulated by the constraints. For this first example, the optimal gains found are in Table 7 and the answer illustrated in Figure 5.

CONCLUSIONS

Time-dependent fouling models for modeling heat exchangers can be considered a gap in this field of study. The concentrated parameters model presented in this work is the most popular in the literature due to the simplicity of the ODEs that describe it and the modeling flexibility for any type of geometry or flow arrangement.

The simulated case studies showed that unadjusted PID parameters could lead to step responses outside the process requirements, ie, high oscillation and temperature spikes, which may be unacceptable for certain stages. process, product requirements or operational safety. By adjusting the PID parameters, it was possible to compensate for the fouling effects and stabilize the quality of the indicators.

THANKS

We thank CAPES-Brasil for the financial support.

REFERENCES

TEMA. *Standards of the Tubular Exchangers Manufacturers Association*. 9th. ed. New York, 2007.

Mu¨ller-Steinhagen, H. (2011). Heat Transfer Fouling: 50 Years After the Kern and Seaton Model. *Heat Transfer Engineering*, 32(1), 1–13.

Trafczynski, M., Markowski, M., Alabrudzinski, S., & Urbaniec, K. (2016). The influence of fouling on the dynamic behavior of PID-controlled heat exchangers. *Applied Thermal Engineering*, 109, 727–738.

Coletti, F., & MacChietto, S. (2011). A dynamic, distributed model of shell-and-tube heat exchangers undergoing crude oil fouling. *Industrial and Engineering Chemistry Re- search*, 50(8), 4515–4533.

Varga, E. I., Hangos, K. M., & Szigeti, F. (1995). Controllability and observability of heat exchanger networks in the time-varying parameter case. *Control Engineering Practice*, 3(10), 1409–1419.

Ravagnani, M. A. S. S. & Caballero, J. A. (2007). A MINLP Model for the Shell and Tube Heat Exchangers Design Using TEMA Standards. *Chemical Engineering Research and Design*, 85(A10), 1423–1435.

Patel, V. K., & Rao, R. V. (2010). Design optimization of shell-and-tube heat exchanger using particle swarm optimization technique. *Applied Thermal Engineering*, 30(11–12), 1417–1425.

Mathisen, K. W., Morari, M., & Skogestad, S. (1994). Dynamic models for heat exchang- ers and heat exchanger networks. *Computers & Chemical Engineering*, 18, S459–S463.

Trafczynski, M., Markowski, M., Alabrudzinski, S., & Urbaniec, K. (2015). Tuning pa- rameters of PID controllers for the operation of heat exchangers under fouling conditions. *Chemical Engineering Transactions*, 52, 1237–1242.