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4/8-4/11: Swisspack 86 — 7th International Packaging Exhibition. Basel, Switzerland. Contact: Schweizer Mustermesse Basel, CH-4021 Basel, Messeplastz, Switzerland.

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5/12-5/15: XVIII International Symposium, International Federation of Fruit Juice Producers. The Hague, Netherlands. Contact: Congrex (Holland) B. V., Keizersgracht 610, 1017 EP Amsterdam, The Netherlands.

5/13-5/15: Powder & Bulk Solids Conference/Exhibition. O'Hare Exposition Center, Rosemont, Ill. Contact: Cahners Exposition Group, Cahners Plaza, 1350 E. Touhy Ave. Des Plaines, IL 60017-5060.

THE EFFECT OF AGE ON THE GASTIGHTNESS OF ISO FREIGHT CONTAINERS

A. K. SHARP[1],
H. J. BANKS[2]
and
A. R. IRVING[1]

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ABSTRACT

For many transport applications, it is desirable to limit the interchange of gas between the atmospheres inside and outside a container. Such applications include the transport of produce prone to moisture uptake, in-container fumigation and controlled- or modified-atmosphere transport, either with or without refrigeration. In this study, the steady-state pressure test was used to measure the gastightness of 143 containers with ages ranging up to eight years. The survey included general-purpose containers of steel, glass-fibre reinforced-plywood and aluminum-panel construction and bulk and refrigerated containers. Within each type of container, the gastightness varied over a wide range, but appeared to depend more on the type of construction than on age. The survey shows that well-constructed containers can retain high levels of gastightness at ages of up to 8 years, and presumably throughout their service lives.

INTRODUCTION

Freight containers have been used in shipping for many years, but have achieved world-wide acceptance only in the last fifteen years with the adoption of ISO specifications. Conceived as a standard stowage module, having the imperial dimensions of 8 ft wide × 8 ft high × 10 ft, 20 ft, 30 ft or 40 ft long, ISO containers of 20 or 40 ft length, carrying up to 20 or 30 tonne, respectively, now constitute the standard method on most routes of carrying all but bulk cargoes.

Containers in general use, although weather-tight, are not hermetically sealed, so there can always be an interchange between the atmospheres

[1]CSIRO, Division of Food Research, N. Ryde, NSW, Australia.
[2]CSIRO, Division of Entomology, Canberra, ACT, Australia.

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inside and outside the container. This may or may not be desirable. In ventilated containers, air interchange is encouraged as a simple means of temperature control or to prevent the accumulation of undesirable gases or vapors (Sharp and Irving 1984; Sharp 1984). More commonly, however, gas interchange should be limited to minimize the entry of water vapor which, in general-purpose containers, causes corrosion damage to metal products (Stokoe 1974) and, in refrigerated containers, increases the rate of frosting and the heat load (Metz 1970). Air leakage must also be limited when a modified atmosphere is created within a container to extend the storage life of horticultural produce (Irving 1984), or when the container is to be used for the disinfestation of grain or other durable foodstuffs (Sharp and Banks 1980).

The rate of gas interchange between a container or other enclosure and the surrounding atmosphere depends on the gastightness of the container (i.e., the sizes, shapes and distribution of all openings in the container through which gas can flow), and also on the prevailing weather and transport conditions (Banks *et al.* 1975; Sharp *et al.* 1976). Therefore, it is necessary to incorporate a measure of gastightness into calculations of gas concentration during fumigation or transport under a modified atmosphere, and into the calculation of the heat load on a refrigerated container. In this paper, results of a survey of the gastightness of various types of container after various periods in service are presented. These data provide typical values of gastightness for use in such calculations, and also indicate the rate at which gastightness can be expected to degrade.

The gastightness of an enclosure commonly is specified in terms of a pressure test. Two types of pressure test are in common use: (1) the equilibrium pressure-flow, or steady-state, test ('P-Q' test) and (2) the pressure-decay test ('P-t' test) (Banks 1984). In this paper, the P-Q test has been used as a basis for comparison, but the two types of tests have been shown theoretically (Sharp 1982) and experimentally (Sharp and Cousins 1982) to be interchangable.

In the P-Q test, a specified pressure difference is maintained by means of a metered air stream into or out of the container; the air flow rate is the measure of gastightness. This process is performed for one or more pressures. The observations follow the expression given by Eq. 1 (Sharp *et al.* 1976).

$$Q = b\Delta P^n \quad (1)$$

where Q = steady-state flow rate of air (m^3s^{-1})

ΔP = applied pressure (Pa)

b = constant (= air flow rate equivalent to an applied pressure of 1 Pa)

n = constant, $0.5 < n < 1.0$ always

$0.6 < n < 0.7$ usually

$n = 0.65$ typically

This expression provides a convenient means of interpolation or extrapolation to estimate the air flow rate, under test conditions, corresponding to any given applied pressure. When only a single pressure-flow observation is made or reported, Eq. 1 can be used, by assuming a value for n (i.e., 0.65), to estimate the flow rate at other pressures. Thus, pressure test data quoted at various pressures can be compared by converting them to a common basis. Equation 1 can also be used when, for practical reasons, it is not possible to generate the reference pressure, either because the enclosure is not sufficiently strong, or because its gastightness is too low.

Although important for many applications, the gastightness of general-purpose freight containers is not specified by standards governing their construction. Gastightness levels specified for refrigerated containers by various organizations are summarized in Table 1. The last entry in this table, for containers intended to retain modified atmospheres, is very stringent, and usually can be met only by sealing the door opening internally with a plastic membrane.

Table 1. Maximum permitted air leakage (litres per second) for refrigerated containers during P-Q pressure test with applied pressure of 250 Pa.

Reference	20 ft "Port-hole" container ($Vol.\ 24\ m^3$)	20 ft Integral container ($Vol.\ 27\ m^3$)	40 ft Integral container ($Vol.\ 52\ m^3$)
International Standards Organization (ISO 1979)	2.2	4.4	8.3
Lloyds Register of Shipping (Lloyds 1968)	2.3	2.3	-
American Bureau of Shipping (Guilfoy 1973)	3.8*	3.8*	7.6*
TransFresh Inc. (for modified atmospheres) (Lugg 1969)	0.22*	0.25*	0.47*

*Calculated from values specified at 125 Pa applied pressure, using Eq. 1 with n = 0.65.

Gastightness standards are also applied to some types of unrefrigerated vehicles and containers such as those in which grain or other material is to be fumigated. The Japanese Plant Protection Service operates a certification scheme (priv. comm.) for bulk containers intended for fumigation with methyl bromide or hydrogen cyanide using a recirculation system. The gastightness specification (that the container, when empty, must retain at least 70% of an initial application of 500 g of methyl bromide after 48 h) is equivalent to an air exchange rate of 0.18 container volumes per

day. The gastightness specification for containers suitable for an in-container disinfestation with carbon dioxide generated from dry ice (6 Ls^{-1} at 250 Pa, Sharp and Banks 1979), corresponds to the same air exchange rate (Banks 1984). This level of gastightness is specified by Australian standard AS2511 (1981) for bulk containers.

EXPERIMENTAL PROCEDURE

Containers of various types of construction and various ages were assessed for gastightness using the P-Q test. The following containers were included in this survey: 45 general-purpose containers of welded-steel, 51 general-purpose glass-fibre-reinforced plywood (GRP) containers, 15 general-purpose aluminum-clad containers, 17 bulk containers, and 15 refrigerated containers of both porthole and integral types.

Before the test, each container was inspected for damage, and any vents or drains were sealed. The gastightness of each container was then determined by measuring the flow rate of air required to maintain each of a number of values of pressure difference.

The container was pressurized with air, supplied from cylinders of compressed air via a double-stage pressure regulator, or from an industrial vacuum cleaner (Fig. 1). The air-flow rate was measured with a bank of variable-area flow meters (''Rotameter'') making due correction for the pressure in the air-delivery hose. The pressure difference between the interior and exterior of the container was measured with an inclined manometer. To increase the accuracy of determination of gastightness, instead of taking only a single reading of air leakage rate at 250 Pa pressure difference several readings were made for each container, over a range of pressures, and the air leakage rate at 250 Pa was found using Eq. 1. Whenever possible, pressures above and below 250 Pa were used, and the air leakage rate at 250 Pa was found by interpolation. With containers of very low gastightness, however, the equipment used was unable to supply sufficient air to reach a pressure of 250 Pa, and the equivalent flow rate at this pressure was found by extrapolation.

RESULTS AND DISCUSSION

Many of the results presented for general-purpose containers were obtained while seeking containers of high gastightness for studies of in-container fumigation of wheat. Containers selected for testing were clean and exhibited no gross current damage, but many bore evidence of previous

FIG. 1. GENERAL-PURPOSE, ALUMINUM-CLAD CONTAINERS BEING PRESSURE TESTED USING VACUUM CLEANER TO SUPPLY AIR, BANK OF ROTAMETERS TO MEASURE FLOW RATE AND INCLINED MANOMETER TO MEASURE INTERNAL PRESSURE.

repair work. It was quickly found, however, that the appearance of the container gave no indication of its gastightness.

Although the pressures used in pressure testing are small, there is the possibility that they could produce forces on the surfaces of the container sufficient to alter the dimensions of the apertures through which the air leaves (or enters) the container, i.e., although reproducible relationships of the form of Eq. 1 may be obtained for each container, it remains possible that the intrinsic gastightness of the container has varied during the test. To investigate this possibility, preliminary tests were made on two general-purpose containers (GRP ceiling and walls, plywood floor, 15 months old), and a new integral refrigerated container. Testing each over a range of positive internal pressures extending to or above 250 Pa and applying Eq. 1 gave air leakage rates at 250 Pa of 9.3, 9.2 and 1.1 Ls^{-1}, respectively. Re-testing each over a similar range of negative pressures gave similar air leakage rates at 250 Pa with, values 8.2, 8.8 and 1.3 Ls^{-1}, respectively. Since positive and negative pressures would effect the dimensions of openings in opposite ways, it can be concluded that apertures in containers remain uneffected by the pressures applied during pressure testing.

During investigations into in-transit fumigation, many general-purpose and bulk containers were pressure-tested empty, and again after loading with grain. Loading generally increased the gastightness of containers of low gastightness (leakage rate greater than 100 Ls^{-1} at 250 Pa), and generally caused little change in containers of high gastightness (leakage rate less than 10 Ls^{-1} at 250 Pa). Thus a pressure-test of a container made before loading provides a conservative estimate of its gastightness in service.

As previously observed (Sharp *et al.* 1976), the pressure test results for each container were described well by expressions having the form of Eq. 1, and gave straight lines when plotted on log-log coordinates. Typical plots (Fig. 2) show that the relationship holds well for containers varying in gastightness over a range of 30,000:1.

The gastightness of each container tested, expressed as the air leakage rate at an applied pressure of 250 Pa, is plotted against the age of the container at the time of testing in Fig. 3 (refrigerated), Fig. 4 and 5 (general-purpose) and Fig. 6 (bulk containers, for granular solids such as grains). Also shown in these figures are the levels of gastightness specified for various purposes. All gastightness values given in these figures were obtained from tests of empty containers. Each point plotted represents an individual container.

The levels of gastightness ranged 5000-fold, with leakage rates at 250 Pa ranging from less than 0.1 Ls^{-1} to greater than 500 Ls^{-1}. Even containers of the same type of construction varied in gastightness over a 100-fold range. Now the gastightness of a new container will be deter-

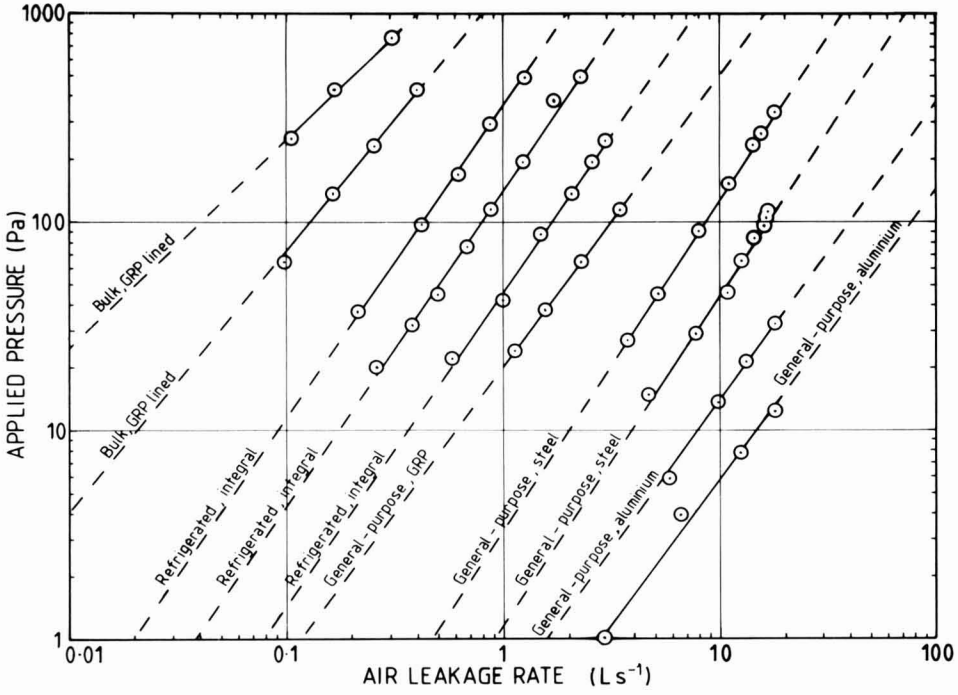

FIG. 2. TYPICAL PLOTS SHOWING THE RELATIONSHIPS, DURING STEADY-STATE PRESSURE TESTING, BETWEEN APPLIED PRESSURE AND AIR LEAKAGE RATE FOR TEN DIFFERENT CONTAINERS, OF VARIOUS TYPES OF CONSTRUCTION, SHOWING WIDELY DIFFERENT DEGREES OF GASTIGHTNESS

mined by the clearances left during construction, and so will depend on the design, and also on the skill and care exercised during construction and might be expected to deteriorate with age. However, Figs. 3-6 show no evidence of any consistent trend in gastightness with age. The reason for this may lie with the rigorous inspection procedures applied by shipping companies to their containers, and to the high standards applied to repair work. Presumably, those events that decrease gastightness also cause visible damage that is detected during the routine pre-trip inspection and repaired before the container is used. In this way, although containers are manufactured with a wide range of gastightness, and may suffer damage and be repaired, the gastightness of containers remaining in or restored to service does not decline with age.

Most of the refrigerated containers tested were within the ISO specifications and meet the gastightness suggested for in-container fumigation (Sharp and Banks 1980), but only one of these reached the level of gastightness specified by TransFresh for a "one-shot" modified atmosphere treatment (Lugg 1969).

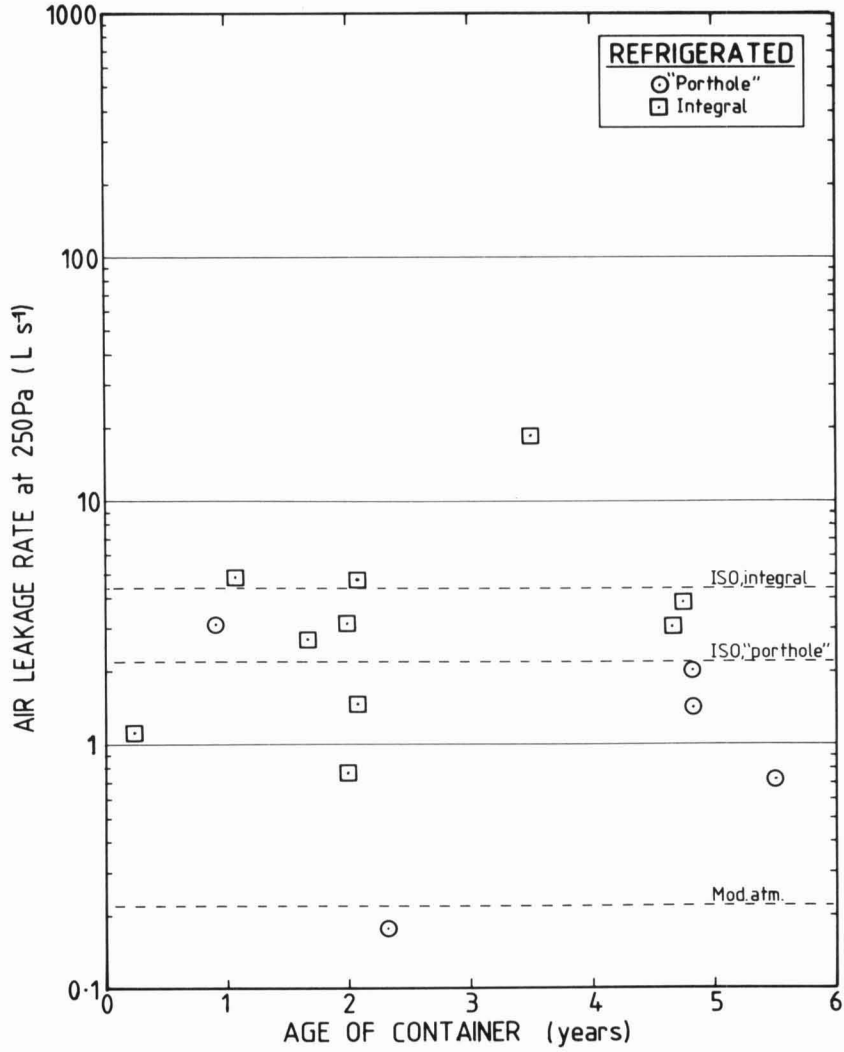

FIG. 3. RELATIONSHIP BETWEEN GASTIGHTNESS AND AGE OF 20 FT REFRIGERATED CONTAINERS. ISO specifications for integral and porthole containers, and TransFresh specification for modified atmosphere containers also shown.

The general-purpose containers tested covered several forms of construction, a particularly wide range of ages and almost the entire range of gastightness. Containers clad with rivetted panels were the least gastight, due presumably to leakage around the rivets, at the joins between panels and at the joins between floor planks. The actual difference between containers with planked and those with plywood floors is greater than is apparent from Fig. 5 since the gastightness of many containers with planked floors was too low to be evaluated using our equipment. The GRP con-

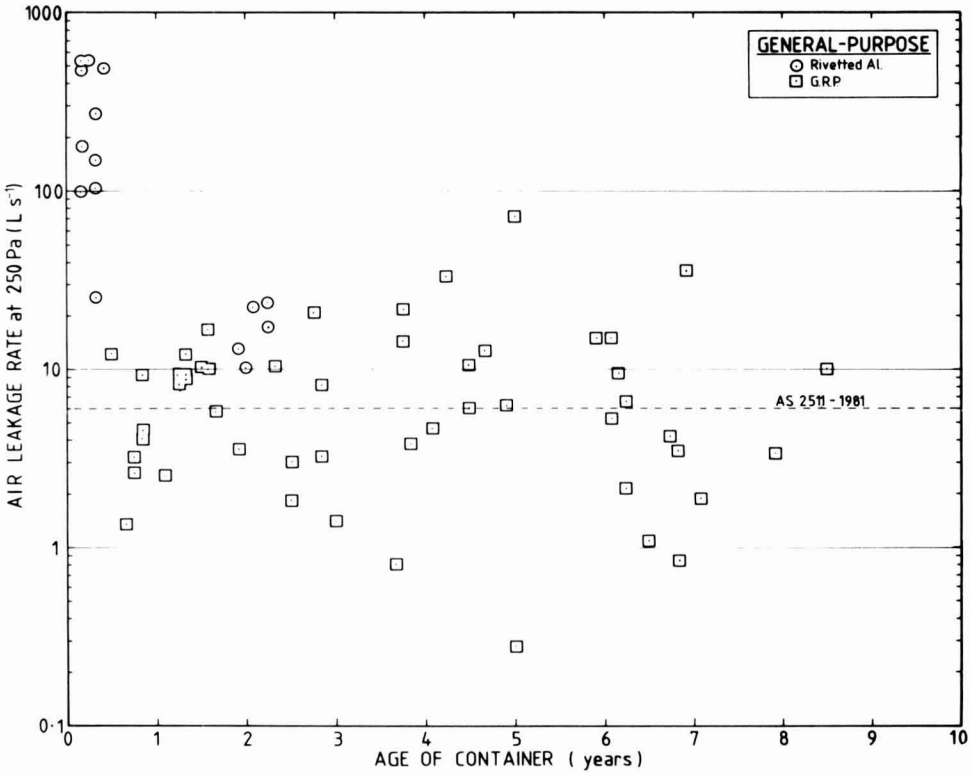

FIG. 4. RELATIONSHIP BETWEEN GASTIGHTNESS AND AGE, OF 20 FT GENERAL-PURPOSE CONTAINERS CONSTRUCTED OF GLASS REINFORCED PLYWOOD, AND OF GENERAL-PURPOSE CONTAINERS CLAD WITH RIVETTED SHEETS OF ALUMINUM. Australian standard for bulk containers also shown.

tainers represented in Fig. 4 were all very similar in design, mostly manufactured in Germany and Holland, and were all fitted with plywood floors and compression-type door seals. Although they varied greatly in gastightness, there was no apparent dependence on age. The steel containers fitted with plywood floors (Fig. 5) were generally of similar gastightness to those made of GRP (Fig. 4).

Apart from one unlined steel container, the bulk containers tested were built in Japan and were in service between Australia and Japan. These containers were fitted with thick compression-type door seals of the type normally used on refrigerated containers, and they were found to be very gastight. One had been certified to the Japanese fumigation standard, but several others were considerably more gastight. The steel bulk container tested was of similar construction, and also of similar gastightness, to steel general-purpose containers.

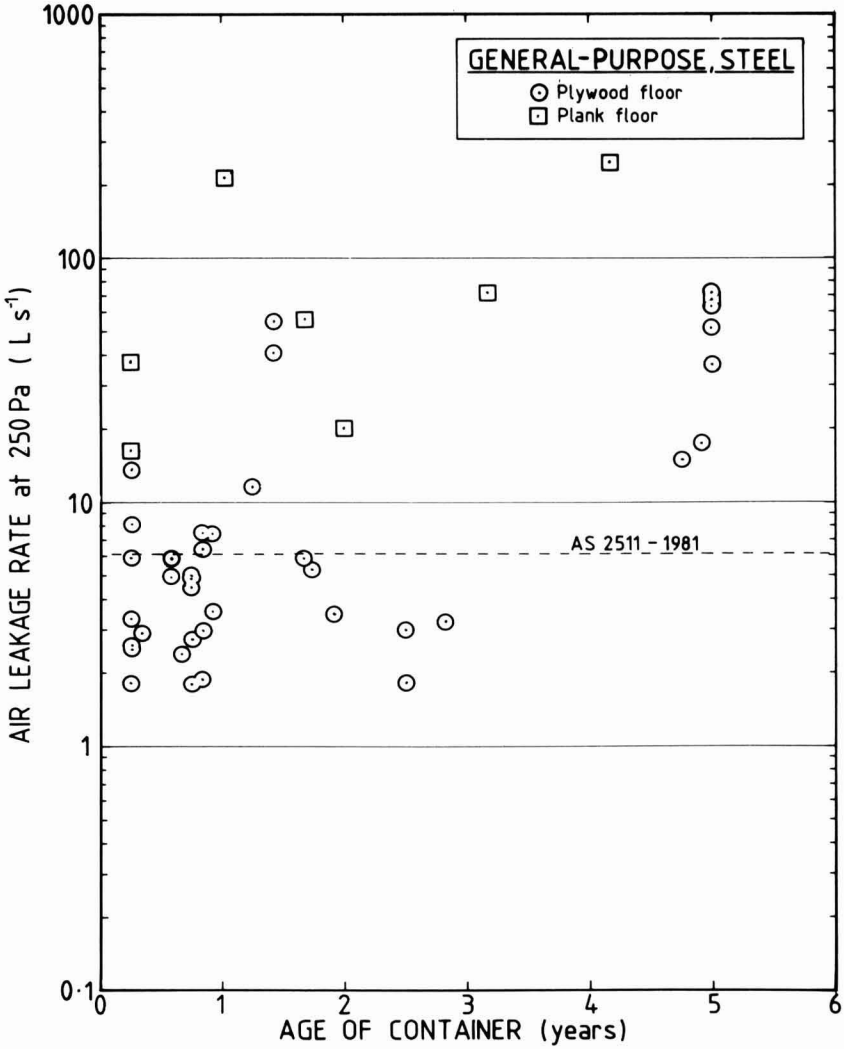

FIG. 5. RELATIONSHIP BETWEEN GASTIGHTNESS AND AGE OF 20 FT GENERAL-PURPOSE CONTAINERS CONSTRUCTED OF WELDED STEEL, WITH PLYWOOD OR PLANKED FLOORS Australian standard for bulk containers also shown.

There is little published information regarding either variation in the gastightness of containers at time of manufacture, or the change in gastightness with age, and what information is available relates only to refrigerated containers and truck bodies. In an American survey of refrigerated trailers and containers (Bodenheimer 1978), the gastightness of 20 trailers of identical design was measured immediately after manufacture. The results are shown in Fig. 7, and give a mean value of 16.8 Ls^{-1} air leakage at 250 Pa, with a standard deviation of 4.6 Ls^{-1}. Although

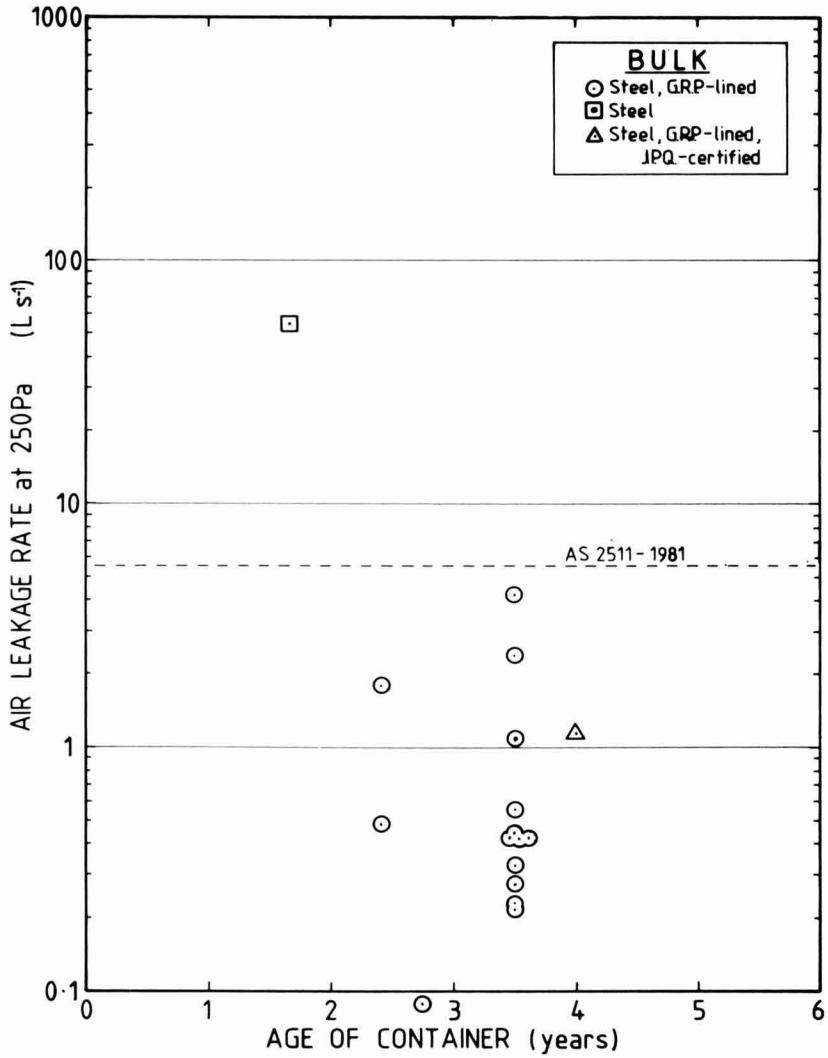

FIG. 6. RELATIONSHIP BETWEEN GASTIGHTNESS AND AGE OF 20 FT BULK CONTAINERS Australian standard for bulk containers also shown.

Bodenheimer does not describe the design of these trailers, these results give an indication of the range of gastightness that can be expected for insulated bodies built under standardized conditions. Bodenheimer also measured the gastightness of 9 integral-refrigerated containers (presumably of 40 ft length) of ages 2 to 9 years from two manufacturers and of 8 trailers of ages 2 to 13 years. These results, also shown in Fig. 7, suggest a steady loss of gastightness with age, and a range of gastightness at a given age similar to that observed in new trailers. Bodenheimer noted that the loss

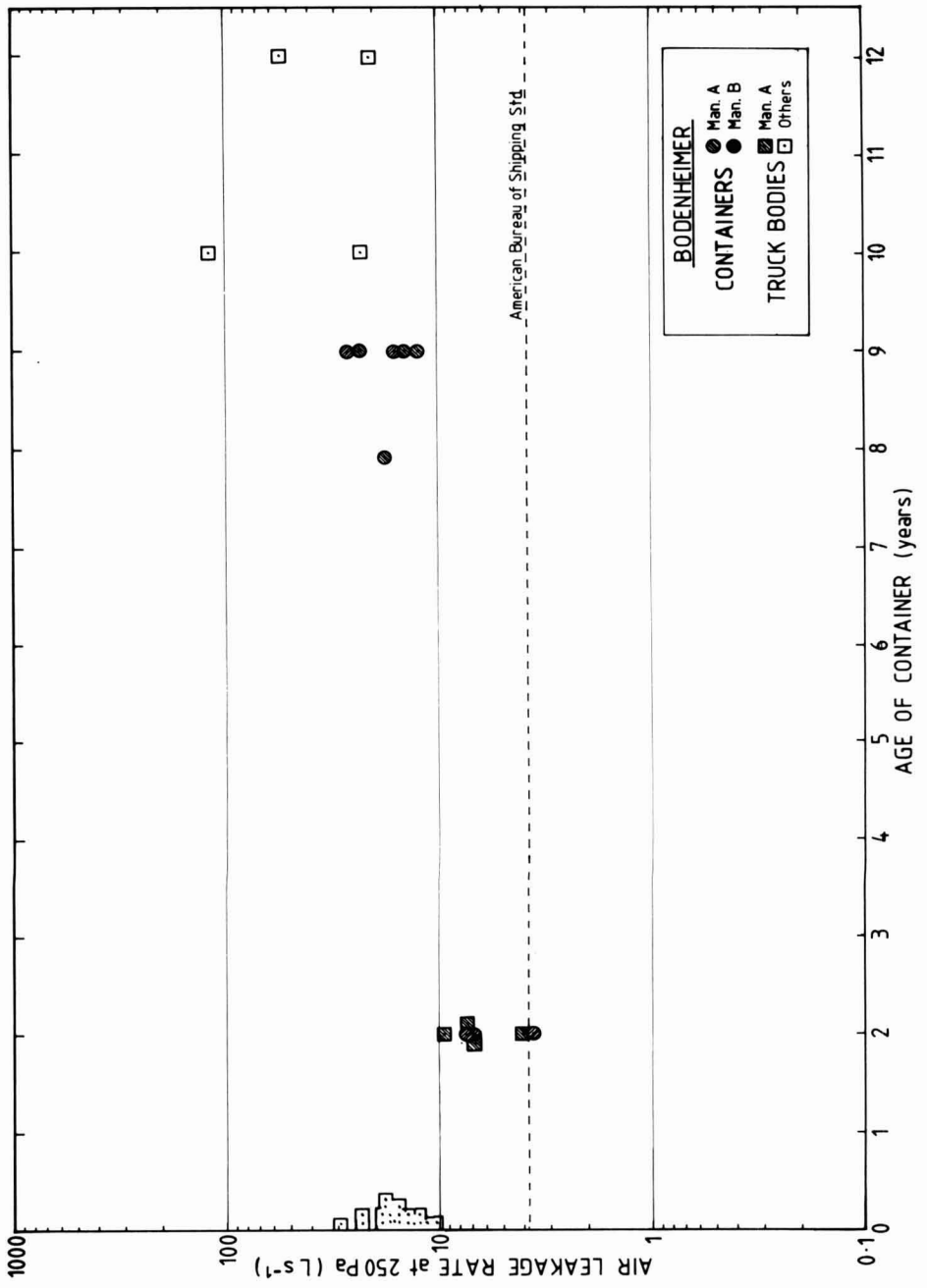

FIG. 7. RELATIONSHIP BETWEEN GASTIGHTNESS AND AGE OF 40 FT REFRIGERATED CONTAINERS AND TRAILERS (Bodenheimer 1978).
American Bureau of Shipping standard for containers also shown. The Truck Trailer Manufacturers Association (USA) specifies P-Q pressure testing for road trailers, but does not stipulate a level of gastightness.

could be accounted for by deterioration with age of the door seals and of the seal between the refrigeration unit and the body.

Gastightness measurements were made on 153 porthole-type 20 ft refrigerated containers of six different forms of construction tested in England by SRCRA (Hales *et al.* 1978, 1979 and priv. comm.) after various periods in service. These workers found, as shown in Fig. 8, that most new containers tested reached, or almost reached, the stipulated level of gastightness, but that once put into service the gastightness rapidly deteriorated. The degree of deterioration, however, was limited, rarely exceeding five times the pressure-test leakage rate when new. Within this range of gastightness there was considerable scatter and, beyond the initial deterioration, no significant trend with age, either for containers of each type of construction, or for the entire sample of containers tested.

As observed by Bodenheimer, one would expect a degradation in the gastightness of a structure with age. These results and those of Hales *et al.* indicate no such trend, even when the containers from one manufacturer were considered alone. It is suggested, therefore, that the gastightness of a container consists of two components. One component is the leakage of air through the rigid structure of the container at apertures which result from manufacturing tolerances or from occasional mechanical damage. This type of leakage may be either increased or reduced during the repair of damage. The other component of gastightness, the leakage of air through flexible seals, would be expected to increase with age, but it appears that the routine inspection procedure, to which containers in the Australian trade are subjected before each trip, is sufficiently rigorous to detect and replace defective components before they have a significant effect on the gastightness of the containers.

During the author' in-transit fumigation investigations, attempts were made to improve the gastightness of general-purpose containers, and to identify the sites of air leakage, by covering selected portions of the interior of the container with plastic film, adhesive tape and building sealants. The main sites of air leakage, in order of importance, were found to be gaps between the floor boards of planked floors, damaged or ill-fitting door seals, gaps between the panels of plywood floors and nail-holes in the floor (presumably left from cargo nailed to the floor!) Sealing these defects individually, or taping a large sheet of plastic film over the whole floor, usually increased the gastightness markedly (reductions in the leakage rate at 250 Pa were as large as twenty-fold) but was very labor intensive. Moreover, it was found that a plastic film floor covering is not sufficiently durable to ensure this improved gastightness is maintained while the container is being loaded. Likewise, attempts to create a gastight interior by fitting a complete welded plastic film container-liner were also unsuccessful because the liner always developed slight tears during loading. Pro-

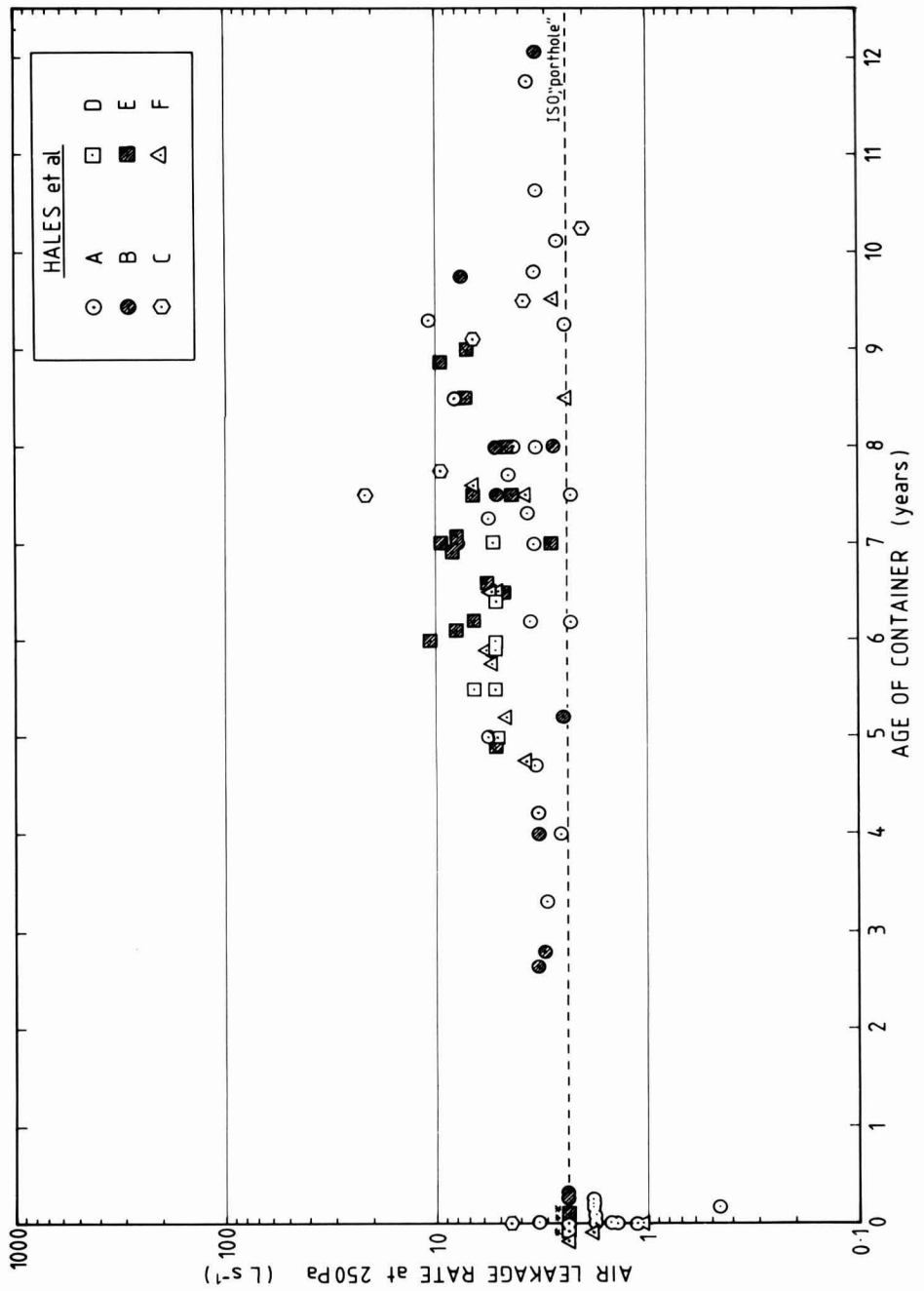

FIG. 8. RELATIONSHIP BETWEEN GASTIGHTNESS AND AGE OF 20 FT "PORTHOLE" TYPE INSULATED CONTAINERS (Hales *et al.* 1978, 1979 and priv. comm.). ISO specification for "porthole" containers also shown.

bably the only effective methods of improving the gastightness for commercial purposes are to replace defective door seals, and to treat the whole floor with a fibre-reinforced, gap-filling, tear-resistant sealant. Extensive trials would be necessary to prove the durability of such treatments and it may be thought more realistic to incorporate gastightness into the design of new containers, and to manufacture to gastightness specifications.

CONCLUSIONS

From this investigation into the gastightness of freight containers, measured by steady-state pressure tests if may be concluded that: (1) the gastightness of containers does not deteriorate in a predictable way with age, (2) it is possible to build and maintain general-purpose and bulk containers having the high levels of gastightness usually associated only with refrigerated containers, (3) gastightness can be determined only by testing, not by visual inspection, and (4) most bulk and refrigerated containers, and selected general-purpose containers, are sufficiently gastight to be used for in-container disinfestation of grain.

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A REVIEW OF FUNDAMENTAL PROCESS ASPECTS FOR THE PRODUCTION OF MUSHROOM MYCELIUM

ANTONIO M. MARTIN

Food Science Program, Department of Biochemistry
Memorial University of Newfoundland
St. John's, Newfoundland
Canada A1B 3X9

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ABSTRACT

A review of fundamental process aspects to produce mushroom mycelium in submerged culture is presented. With peat hydrolysate as the main nutrient source, the operations of substrate preparation, cultivation and product recovery are explained. Based on the growth of Agaricus *spp. mushroom mycelium, the conditions for the production of the desired mycelial pellets are analyzed.*

INTRODUCTION

The production of mushroom mycelium in submerged culture process has been studied for several years. Extensive reviews of the works in this field have been presented by Litchfield (1967a) and Worgan (1968). Fungal biomass can be grown in submerged culture as a single-cell protein source (Litchfield 1968), but the main focus of the interest has been with the use of the mycelium as food or flavoring agents (Solomons 1975). Given the increasing market for mushrooms, work was undertaken utilizing different species, in an attempt to develop an industrial process where mushroom mycelium could be grown. The first commercially orientated studies were conducted by Humfeld (1948) utilizing the species *Agaricus campestris*, which is considered a wild species of the cultivated mushroom *Agaricus bisporus*, although confusion between both has been common in the literature (Singer 1961). In addition, the identity of Humfeld's original strains has been questioned (Molitoris 1962). Besides the *Agaricus* species and because of the pleasant flavor typical of the morel mushrooms, several

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members of the genera *Morchella* have been studied. A commercial operation to produce *Morchella* spp. mushroom mycelium was reported in the sixties (Kliss 1963) based on the so-called Szuecs process (Szuecs 1956). Recent studies involved the utilization of spent sulphite liquor (Kosaric *et al.* 1973) and cheese whey (Kosaric and Miyata 1981) to produce mycelium from several morel mushrooms; peat hydrolysates (Martin 1983a,b; Martin and Bailey 1985a) have also been utilized in the growth of *Agaricus campestris* and *Morchella esculenta*. In addition, other types of mushrooms such as the *Pleurotus* spp. (Zadrazil 1975) and *Volvariella* spp. (Ghosh and Sengupta 1977) have been grown in submerged culture.

Different substrate sources, in particular food plant wastes, have been investigated in regard to using them as a nutrient source in the mycelial growth. At Memorial University of Newfoundland, it has been of interest to study the potential of peat as a feedstock in industrial productions, specifically as a source of nutrients for biological processes. The use of peat and peat hydrolysates in several microbial processes have been reported (LeDuy 1979). The hydrolysis of peat is a necessary step in some of the processes aimed at dewatering raw peat and to increase its caloric value, one method of accomplishing this is the "Wet Carbonization" process (Fuschman 1980). This hydrolysis produces a liquid fraction which is rich in both organic and inorganic substances (Zommers *et al.* 1974) and, if untreated, could be a source of pollution. In Canada there are several areas, specially in the northern regions, with abundant and practically non-utilized peat resources. Therefore, peat and peat hydrolysates could be an inexpensive substrate for producing food stuffs, especially where agricultural output is limited due to climatic factors. The two main envisaged uses of mushroom mycelium is as a flavoring agent in a powdered form for use in soups and sauces, or as flavored pellets which could be consumed in place of the fruiting body of the mushrooms cultivated in solid medium. Several studies have been made on the nutritional and flavor characteristics of different mushrooms (Humfeld and Sugihara 1952; Reusser *et al.* 1958; Moustafa 1960; Litchfield 1967b; LeDuy *et al.* 1974), including their growth in peat hydrolysates (Martin and Bailey 1985a) and the production of mycelial pellets (Martin and Bailey 1985b). Based on the basic research done in the production of mushroom mycelium utilizing peat hydrolysates as the main substrate source, this paper presents an approach for the design of a process for this production.

TYPE OF MUSHROOM

As has been stated previously, several species of mushroom can be grown in submerged fermentation to produce mycelium, the flavor characteristics,

morphology and acceptance of the product will depend on the species, the medium and other process considerations. This paper will deal with the mushroom *Agaricus bisporus*, which is practically the only species cultivated and consumed in North America. Tissue culture of sporophores (fruiting body) of fresh *A. bisporus* were conducted following a method similar to that reported by Cirillo (1960) and subsequently adapted to peat hydrolysates employing the same technique to that utilized with *A. campestris* (Martin 1983c).

CULTURE MEDIUM

Peat Hydrolysates

One objective of the hydrolysis of peat is to convert the cellulose, which can account for up to 14% of the peat (Smith *et al.* 1958) to glucose, which can be metabolized by most microorganisms. The process will also extract other carbohydrates in addition to several organic substances some of which may be utilized as nutrients. Several operating variables are involved in the hydrolysis of peat, such as temperature, retention time, type of catalytic agent (generally, an acid) and peat-acid concentration ratio. In addition, peat particle size and original moisture content could influence the process (Martin and Bailey 1984). Normally, temperatures of 150 °C or higher have been employed. However, if the liquid hydrolysate is intended to be used as a nutrient source for biological processes, milder conditions could be preferred, although the time required for the hydrolysis could be lengthened and the process yield lower.

The use of debituminized peat, the removal of humic acids, the use of HCl as hydrolysis agent and a low temperature-low pressure process have been proposed as alternative ways for the peat hydrolysis process (Fuchsman 1983; Chang 1985). Those methods are worthy of being tested with the aim of removing potential inhibiting substances for the mushroom mycelium growth.

Preparation of the Substrate

After filtration, the low pH of the peat hydrolysate (1.5) needed to be adjusted to the optimum value required by the fermentation process. In addition, and if required, nutrient supplements should be added to compensate for deficiencies in the basic peat hydrolysate composition. In the basic studies with mushroom mycelium grown in peat hydrolysate (Martin 1983a,b), NaOH was utilized to adjust the pH. NaOH was chosen in the interest of finding the potential of peat hydrolysate to wholly support the fungal biomass. However, in a scaled-up production of mushroom

mycelium, the use of NH_4OH is recommended because peat hydrolysate have been found to lack sufficient nitrogen to support an active growth of microbial biomass (Quierzy *et al.* 1979). Peat hydrolysates have been also found deficient in phosphorus. Supplementing the media with $(NH_4)_2HPO_4$ or yeast extract produced a better growth in the production of *Agaricus* spp. mushroom mycelium (Martin and Bailey 1985a). Utilization of dilute peat hydrolysates and supplementation of this with yeast extract was required to produce mycelial pellets, the diluting being necessary to reduce the assumed growth inhibitors in the peat hydrolysate (Martin and Bailey 1985b).

PROCESSING METHODS

A general diagram for the sequence of operations in the process of producing mushroom mycelium from peat hydrolysates is shown in Fig. 1.

Hydrolysis of Peat

The hydrolysis of peat is conducted in a reactor chamber where raw peat is mixed with an acid solution and subsequently steamed. In the hydrolysis of peat for the production of substrate for the mushroom mycelium growth, the following conditions have been generally employed: temperature, 121 °C (15 psig); time, 2 hours; acid concentration, 1.5% (v/v) H_2SO_4 mixed with peat in the ratio of 5–6 parts solution to 1 part peat (calculated on a dry basis). In Fig. 2, a diagram for the material for the hydrolysis process and the subsequent solid-liquid separation is presented.

The following are the total mass balances for the processes:

In the reactor unit: $A + P + S = Z + V$

In the separation unit: $Z = R + E$

For all the system:

$$A + P + S = V + R + E$$

Defining:

X_p : % solid in P
X_r : % solid in R
Y_p : % liquid in P
Y_r : % liquid in R

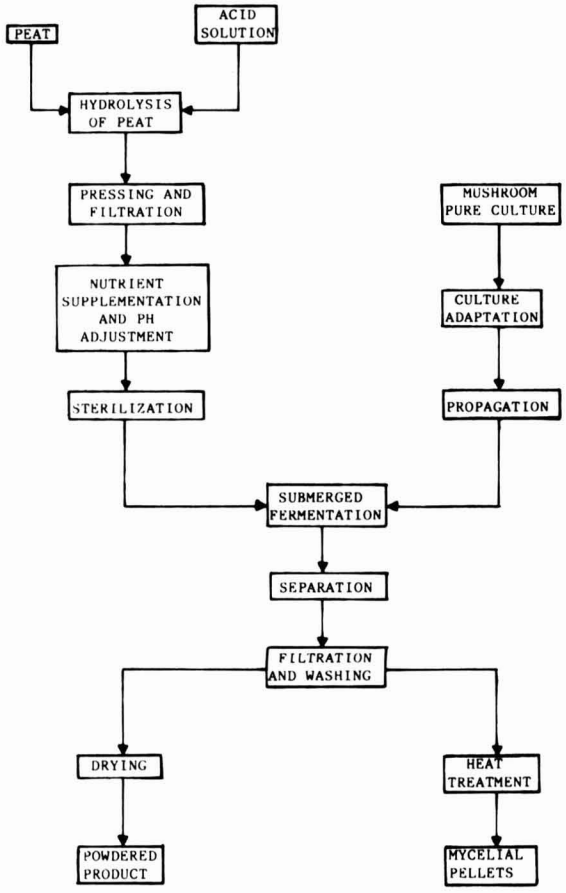

FIG. 1. OPERATION SEQUENCE IN THE PRODUCTION OF SUBMERGED MUSHROOM MYCELIUM FROM PEAT HYDROLYSATES.

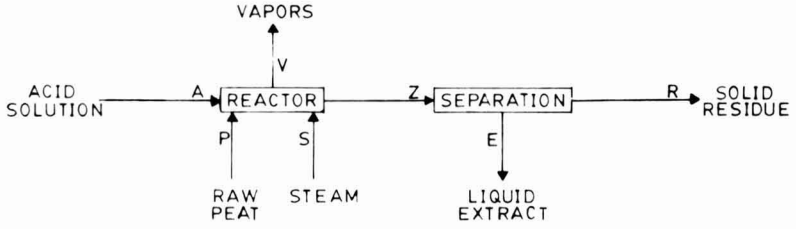

FIG. 2. DIAGRAM SHOWING PROCESS FLOW FOR THE ACID HYDROLYSIS OF PEAT AND THE RECOVERY OF THE PEAT HYDROLYSATE.

The component mass balances for both the solid and the liquid fractions of the peat, are:

Solid:

$$(X_p)(P) - (W)(X_p)(P) = (X_r)(R)$$

Liquid:

$$A + (Y_p)(P) + (W)(X_p)(P) = V + (Y_r)(R) + E$$

Where: W = yield of hydrolysis:

$$W = \frac{\text{mass of solid peat solubilized}}{\text{total mass of solid peat}}$$

Cultivation

The peat hydrolysate based substrate is fed to a batch fermentor and inoculated with a starter culture of the desired mushroom species. The previous stage of culture propagation must be able to provide the required amount of inoculum as specified in the predetermined inoculum ratio, and it should be grown with the same conditions as those operating in the fermentor, in order to avoid a growth lag phase in the latter. In the case under study, *Agaricus bisporus* mycelium was cultivated under the optimized conditions found for the growth of *Agaricus* spp. in peat hydrolysate (Martin 1983a), which are: 24 °C, pH:6 and 4% (v/v) inoculum ratio.

In the production of mushroom mycelium, mechanical agitation and aeration acquire special significance for the process, in addition to their intrinsic importance in the overall reaction rate and mass transfer patterns. It is a recognized fact that the morphology of the mycelium is affected by the rates of aeration and the medium turbulence (Litchfield 1967a). In the growth of *Agaricus* spp. in peat hydrolysates, mycelial pellet formation was reported (Martin and Bailey 1985b) using no mechanical agitation and 1.0 vvm (volume of air per volume of medium per minute), and maximum filamentous mycelium production was found at 200 rpm and 1.5 vvm. In addition, 8 days were considered the appropriate fermentation time. These were the parameters adopted to obtain the results presented in this review.

Figure 3 shows a diagram for the cultivation process.

With the following definitions:

X: dry mycelial concentration

S: substrate (total carbohydrate) concentration

V: volume: $V = V_I + V_S$

Y: yield (dry mass of mycelium produced/total carbohydrate consumed).

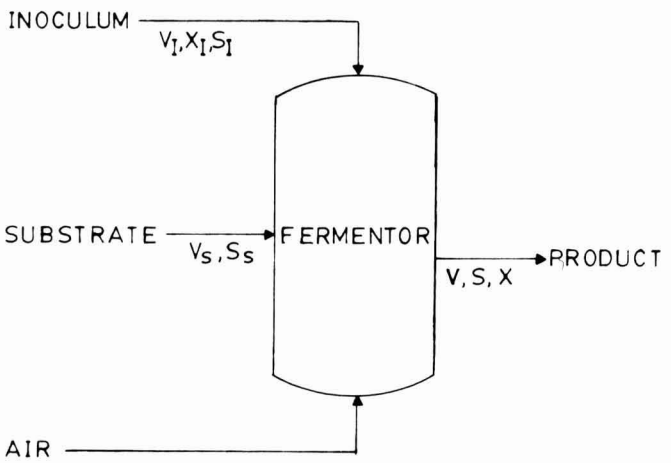

FIG. 3. DIAGRAM SHOWING PROCESS FLOW FOR THE SUBMERGED GROWTH OF MUSHROOM MYCELIUM.

The mass balance for the mycelial concentration in a batch process is:

$$(V_I)(X_I) + (V_S)(S_S)(Y) + (V_I)(S_I)(Y) = (V)(X)$$

The corresponding mass balance for the substrate concentration is:

$$(V_I)(S_I) + (V_S)(S_S) = (V)(S) + (V)(X)/Y$$

DISCUSSION

Substrate Preparation

Carbohydrate, a carbon source, and nitrogen, which is essential for protein synthesis, are considered the main nutrients found in peat hydrolysates. The carbohydrate and nitrogen composition of the utilized peat hydrolysates is presented in Table 1, which also presents data from other published works. As far as the production of mushroom mycelium is concerned,

Table 1. Carbohydrate and nitrogen composition of peat hydrolysates (g/L).

Carbohydrates	Nitrogen	References
37-43	*	Evdokimova et al. 1974
43	1	Zommers et al. 1974
34	*	Quierzy et al. 1979
30.6±0.8**	0.4±0.03**	This work.

*Data not reported.
**Mean values ± standard deviation.

the objective of an improved hydrolysis should not necessarily be higher extract concentrations as represented by the TCH value, because of the inhibitory effects reported, which make it necessary to dilute the hydrolysates. However, the process yield (mg TCH produced/g dry peat utilized) could be enhanced if appropriate equipment for the hydrolysis process is designed. Because of the characteristics of the peat:acid solution mixture, it has been stated that incomplete mixing could be hindering a more complete hydrolysis (Quierzy *et al.* 1979; Martin and Bailey 1984) therefore, an agitated reactor would seem to be an appropriate choice in the design to improve the yields in this step.

After hydrolysis, separation of the liquid hydrolysate from the remaining peat solid should be conducted. The solids are not in suspension but embedded with liquid, and pressure should be applied. A filter press appears to be a practical solution. After separation, the liquid could be fed to the fermentor, where pH adjustment and nutrient supplementation is conducted. Usually, a sterilization process should precede the cultivation stage. However, savings in energy can be obtained if it is considered that the hydrolysate has been sterilized because of the thermal treatment required by the hydrolysis reaction. The final low pH value of the liquid (around 1.5) will preserve the product at a great extent. If care is taken in the transportation of the liquid and in the filtration, implementing aseptic conditions where possible, the hydrolysate could be spared a second thermal treatment. Instead, the fermentor would only need to be sterilized with the water required for the dilution of the hydrolysate plus the pre-calculated requirements for supplementation and pH adjustments. After cooling, the peat hydrolysates could be added and the system subsequently inoculated.

Mycelium Production

The design of an appropriate fermentor for the culture of mushroom mycelium is an important step in the overall process. As was stated previously, mechanical agitation could be detrimental to the mycelial growth, in either filamentous or pellet form. Several specific fermentor designs for the growth of molds have been reported (Means *et al.* 1962; Rowley and Bull 1973). Those designs apply mostly to the production of filamentous mycelium where, despite the fact that the mycelium tends to wrap around the impeller, the shearing action between agitator blades and baffle plates tends to avoid the accumulation of a mycelial deposit on the exposed surfaces of the tank walls. However, the agitation speed must not exceed the limit above which the mycelium metabolism is affected.

Air-lift fermentors could be a good design for mold biomass growth (Malfait *et al.* 1981). However, in mushroom mycelium production the

intensity of aeration is also an important factor in the development of an adequate morphology and flavor (Litchfield 1967a). This means that higher aeration rates, which favors biomass production, are not necessarily appropriate if the mushroom mycelium is to be used as a food product. Taking the previous considerations into account plus the fact that in the case under study the nutrient concentrations are low because of dilute substrates, it can be concluded that the culture of mushroom mycelium is a low-density biomass process. This affects the productivity of the system but in the case of a product with the potential high economic value that a mushroom substitute could have, the low productivity is of less importance than it would be in other biomass productions, such as single-cell protein.

The production of pellets has been reported as the most likely to produce a flavored product (Litchfield 1967a). In addition, the viscosity of the medium is practically unaffected, which may not be the case in filamentous growth. These facts suggest that a simple design consisting of a baffled aerated reactor (without mechanical agitation) as being adequate for this process. However, studies on mushroom mycelium growth in airlift and tower fermentors, which have been scarce in the past, should be conducted. Because the low growth rate does not warrant the utilization of continuous culture, the only required controls built in to the system are aeration, temperature and pH. Foam, when occurring, can be easily controlled by an appropriate anti-foam agent.

Table 2 summarizes the results of our experience in the growth of *A. bisporus* mycelium utilizing peat hydrolysates as the main nutrient source,

Table 2. Growth parameters of *Agaricus* spp. grown in submerged fermentation.

Species	Substrate	Substrate Conc. (%)	Mycelium Conc. (%)	Yield[1] (%)	Efficiency[2] (%)	References
A. blazei	Glucose	5.0	15.2	42.0	30.0	Block et al. 1953
A. campestris NRRL 2334	Glucose	4.0	11.4	38.0	28.5	Reusser et al. 1958
A. campestris NRRL 2334	Peat hydrolysate[3]	3.0	7.1	49.0	23.4	Martin and Bailey 1985a
A. campestris NRRL 2334	Peat hydrolysate[4]	2.5	5.5	37.0	22.0	Martin and Bailey 1985b
A. bisporus	Peat hydrolysate[4]	2.5	5.0	33.0	20.0	This work.

[1]Yield: weight dry mycelium produced/weight total carbohydrates consumed.
[2]Efficiency: weight dry mycelium produced/weight total initial carbohydrate.
[3]Nondiluted, supplemented with 0.5% yeast extract. Filamentous growth.
[4]Diluted and supplemented with 0.5% yeast extract and 1.0% glucose. Pelleted growth.

and compares them to other results from *Agaricus* spp. reported in the literature.

Pellet Growth

The growth of a mold biomass, specifically in the case of filamentous pellet cultivation, is not only related to the increase in the number of cells, but to the change in size and morphology of the mold pellet. It has been reported that in submerged culture the filamentous biomass increases at a slower rate than unicellular organisms, whose rate of growth is exponential (Bailey and Ollis 1977). Several researchers have found that fungal growth in homogenous culture is best fitted to a cube root law (Marshall and Alexander 1960). Instead of the logarithmic function commonly used to represent the growth of microorganisms, they proposed the following model (Eq. 1),

$$m^{1/3} = m_o^{1/3} + \propto t \tag{1}$$

Where m and m_0 denote the total biomass and initial biomass, respectively, t is time and ∞ is the cube root growth rate constant. Pirt (1966) provided the explanation for the cube root growth theory, identifying it with the growth of mycelium in pelleted form. Because the pellet mass grows proportionally to the cube of its radius and the surface area for substrate intaking increases in proportion to the square of the radius, only a peripheral shell of the pellet would have sufficient substrate to maintain exponential growth.

The biomass of a pellet of density p and radius r is given by (Eq. 2),

$$m = \frac{4}{3} (\pi) (\rho) (r^3) \tag{2}$$

Defining w as the depth of the peripheral growth zone of the pellet and μ as the specific biomass growth rate: $\mu = \frac{dm}{dt} \cdot \frac{1}{m}$, the following equation is presented for the growth of the pellet biomass (Righelato 1975) (Eq. 3),

$$m^{1/3} = (3/4\pi P)^{-1/3} \mu .w.t + m_o^{1/3} \tag{3}$$

If r is smaller than w, exponential growth could occur. However, it is predicted then the growth will follow a cube root pattern if r is greater than w. Since m_0 is usually small relative to m, the biomass increases with a cubic dependence on time (Bailey and Ollis 1979). In our research in

the growth of *Agaricus* spp. pellets, a rate of growth with this kind of relationship between biomass and time was observed after the first two days of fermentation, lasting for three days, while the total fermentation time was eight days. Agglomeration of mycelium was observed during the first two days, developing in the pellet form, which increased in diameter for six days, to approximately 6 mm (Martin and Bailey 1985b). It has been pointed out that the limitation in the diffusion of nutrients and oxygen to the pellet interior produces the autolysis of the biomass located in the center of the pellet. This autolysis has been accounted as one of the main factors in the flavor development of pelleted mushroom mycelium (Litchfield 1967a). Our experiences in the production of *Agaricus* spp. pellets confirmed these observations. Figure 4 shows the actual growth of mycelial pellets in a laboratory fermentor.

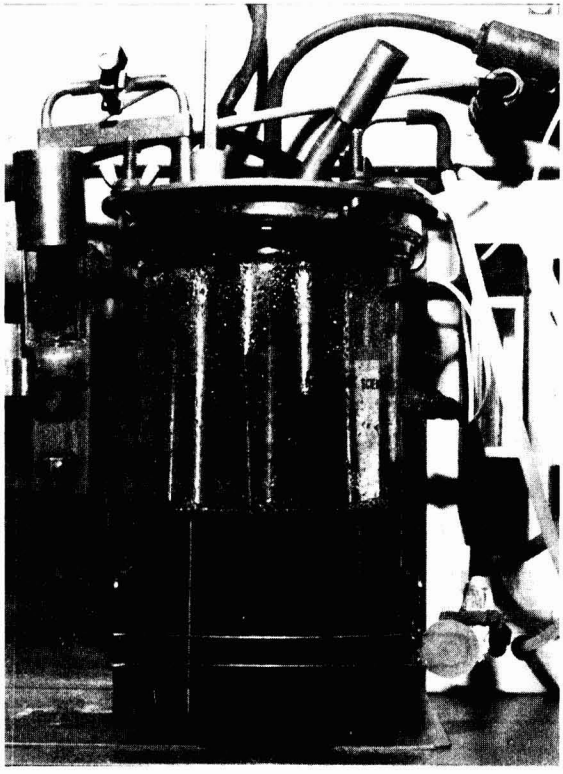

FIG. 4. PELLETS OF *Agaricus* spp. MUSHROOM MYCELIUM CULTIVATED IN A PEAT HYDROLYSATE-BASED MEDIUM.

Post-cultivation Product Processing

After the pellet formation and growth has ended, the contents of the fermentor are extracted and the mycelium separated from the particular medium used. One of the advantages of pellet production is that the pellets are easily separated by simply filtering through an appropriate net, following this the pellets should be washed with water to remove any peat hydrolysate medium left wetting their surfaces. A further mild heat treatment and packaging will be the only operations required if a whole mushroom pellet is desired. Drying, preferably in a convection drier, will produce a material to be utilized as a food additive. In general, the post-fermentation treatments are simple when compared to other fermentation productions and will contribute to the overall efficiency of the process.

The composition of the pellet form of *A. bisporus* mycelium grown in diluted and nutrient-supplemented peat hydrolysates is presented in Table 3, and compared with the composition of other *Agaricus* spp. mushroom biomass.

Table 3. Composition of *Agaricus* spp. mushroom biomass grown in submerged fermentation.

		Component (% Dry Basis)			
Species	Substrate	Protein	Fat	Ash	References
A. blazei	Glucose	32.5	*	*	(a)
A. campestris					
NRRL 2334	Glucose	13.9	8.4	*	(b)
	Molasses	40.9	1.5	*	(b)
	Sulfite waste	34.3	0.8	*	(b)
	Peat hydrolysate**	44.4±1.6	2.8±0.4	9±0.5	(c)
NRRL 2335	Glucose	23.2	9.9	*	(b)
	Molasses	39.9	3.3	*	(b)
	Sulfite waste	34.7	0.9	*	(b)
NRRL 2336	Glucose	13.4	6.6	*	(b)
	Molasses	39.8	2.6	*	(b)
	Sulfite waste	25.2	0.5	*	(b)
A. bisporus	Peat hydrolysate**	35.0±2.8	5.5±1.1	8±1.5	(d)

*Data not reported.
**Mean values ± standard deviation.

(a) Block *et al.* 1953.
(b) Reusser *et al.* 1958.
(c) Martin and Bailey 1985b.
(d) This work.

CONCLUSIONS

Although the acceptance of mushrooms as a flavorful and nutritious food product is widespread, the biochemical mechanisms involved in the production and development of flavor has not yet been determined. Advances in the fermentation industry and mushroom genetics could provide a mycelial product which would compete and/or substitute for the mushroom fruit body. In this work, the fundamental aspects for a process to produce mushroom mycelium has been presented. The particular substrate employed, peat hydrolysate, provides the advantages of being an inexpensive raw material and of providing savings by reducing the amount of sterilization needed. Mycelial pellets could be produced by carefully controlling the cultivation process. Pellets will result in easy product separation and development of flavor. Although the mycelial concentration obtained is comparable to other reported results, the growth parameters, as expressed by the dry mycelial concentration, yield and efficiencies, are low. Medium composition modifications and strain improvements are required to enhance these parameters, which are important in regard to the economic feasibility of the process. Finally, toxicity, nutritional and flavor studies together with a general economic analysis, need to be conducted to complement the process design.

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AUTOMATIC CONTROL OF MOISTURE IN FOOD EXTRUDERS[1]

LEON LEVINE,
STEVEN SYMES
and
JOHN WEIMER

The Pillsbury Company
311 2nd Street S.E.
Minneapolis, Minnesota 55414

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ABSTRACT

A computer simulation illustrating the possibility of extrudate moisture control via feedback of die pressure is described. The simulation of a controller, utilizing dead-time compensation, illustrates the advantages of such a control strategy. The sensitivity of the dead-time compensating controller performance to errors in parameter estimation is examined and discussed. This type of predictive control is found to provide a stable control strategy even with very large modeling errors. Such a control system is readily implemented with a microprocessor.

INTRODUCTION AND OBJECTIVES

The literature (Harper 1979; Harper 1981; Holay and Harper 1981; Van Zuilichem *et al.* 1975; Kauffman and Hatch 1977; Harmann and Harper 1973; Bruin *et al.* 1978) teaches that moisture control in extruders is a necessary requirement for product quality control. Variance in moisture causes changes in energy input, pressure, and strain applied to the extrudate. These can result in changing product quality. Therefore, it is clear that an automatic control system which would respond to, and correct moisture variations in the extrudate would be very desirable. These variations are likely to occur due to variations in raw material moisture or feeder outputs. It is within the experience of the authors that the moisture varia-

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tions of raw materials constitute a very real problem in actual plant operations.

The literature illustrates one method of controlling moisture in extruders (Harper 1981). This method used ratio controllers on the water and solids flow to the extruder to maintain moisture control. This does not incorporate continuous moisture measurement of the solids and its feedback for generating any subsequent error signal upon which a control action can be made. Consequently, this scheme will at best only eliminate moisture variations caused by upsets in the feeders and pumps. Since this scheme does not actually determine the moisture of the extrudate in any way, it cannot compensate for fluctuations in the moisture content of the raw materials, or any deviations from ideal performance of the ratio controller or plant.

Several sources (Jao *et al.* 1979; Levine 1982; Harper 1981) illustrate that the viscosity of the extrudate is a strong function of its moisture content. As a consequence, the die pressure of the extruder will vary strongly with moisture. The existence of a relationship between the moisture content of the extrudate and the die pressure offers the possibility of the use of feedback control of the moisture content of the extrudate. Using die pressure to generate an error signal would provide significant advantage, since the variable could be read continuously, while the time-consuming nature of current moisture analysis techniques would render any automatic control scheme including those employing feed forward impractical.

A difficulty arises in using classical control strategies for the feedback control described above: extruders exhibit a high percentage of their total residence time as a pure dead-time (transportation lag). This results in the introduction of instabilities in the control loop if reasonable controller gains are used. In order to render such systems stable, a small proportional gain is required. This results in a "sluggish" control response (De Keyser *et al.* 1981; Bahill 1983).

One way around the difficulty of controlling processes exhibiting dead-times was the use of predictive control techniques (Smith 1957). The dead-time compensator, sometimes called the Smith Predictor, was introduced to eliminate the effects of dead-time on the loop response, thus rendering the control system stable and improving response in systems exhibiting a significant time delay (Meyer *et al.* 1976). There are many variations on this basic theme described in the control literature (De Keyser 1981; Bahill 1983; Herget and Frazer 1980; Watanabe *et al.* 1983; Linkens *et al.* 1982; Herget and Frazer 1981; Palmor and Shinnar 1981; Furukawa and Shimemura 1983).

The dead-time compensator was proposed before the wide availability of low cost digital computing systems such as those based on the microprocessor. At the time of proposal, control systems, which were based

on pneumatics, mechanical linkages, passive electrical components and vacuum tube based active electronic components, could not realize the complex system functions required to implement practical predictive control schemes. Modern digital design has changed all this making predictive control a readily realizable control scheme. It should be mentioned that analog realizations based on the operational amplifier may also constitute a viable method for controller designs.

It is the objective of this paper to illustrate a dead-time compensating controller for use in the moisture control of food extruders, and elucidate its advantages and weaknesses. We will illustrate this through the use of computer simulation of the process and the controller.

PROCESS MODELS

In order to specify a process control system, there are two requirements: (1) a quantitative description of the time response of the system; and (2) a quantitative relationship between the controlled and controlling variables.

The time response curve for a typical food extruder may be found in the literature (Harper 1981; Bruin *et al.* 1978 Middleman 1977). A generalized time response, which has been calculated from literature data (Harper 1981) is presented in Fig. 1. Dimensionless time has been used

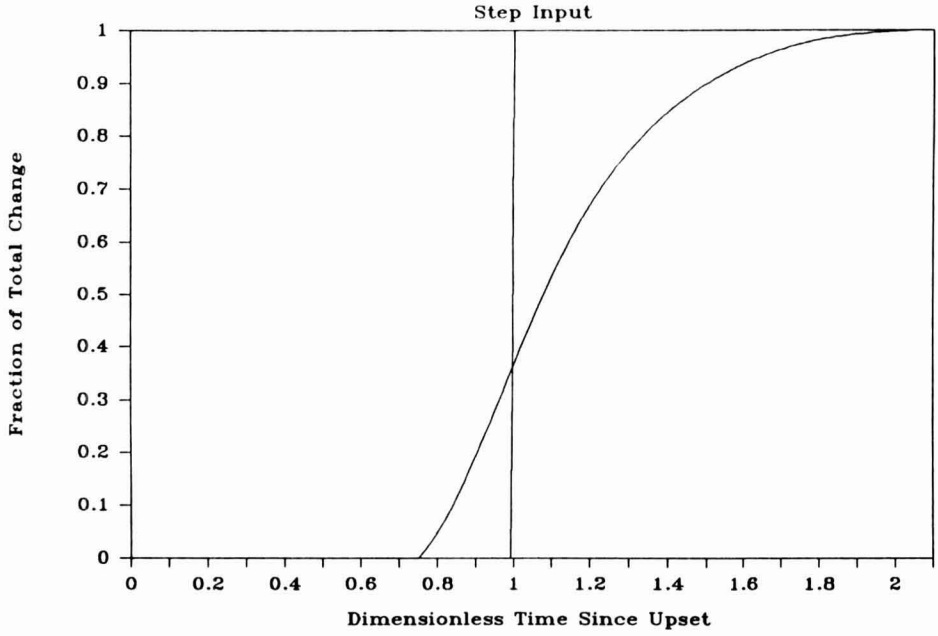

FIG. 1. PROCESS RESPONSE.

for all dynamic curves, and is equal to the ratio of actual elapsed time to the mean residence time of the extruder. In the figure we see that the pure delay in the extruder amounts to about 75% of the extruder's mean residence time. We will use this curve in our simulations.

Typical relationships of the extrudate viscosity to moisture may also be found in the literature. The dependence is usually expressed in the form (Eq. 1),

$$\mu \propto e^{AM} \tag{1}$$

where: μ is the apparent viscosity
A is an empirical constant
M is the extrudate moisture (dry basis).

Equation 1 may be linearized about the target moisture, and using knowledge of the die geometry and the flow rate of the system, a linearized relationship between pressure and moisture can be obtained. Since the actual moisture range that we are likely to encounter is small, the errors introduced by this linearization will not be significant.

For example an equation available from the literature (Levine 1982) may be linearized from (Eq. 2),

$$P \propto e^{-0.068M} \tag{2}$$

to (Eq. 3),

$$\frac{\Delta P}{\Delta M} \propto -0.068e^{-0.068M_o} \tag{3}$$

where: P is the die back pressure.

Over the range of moistures reported, 27.5% to 32.5% (wet basis), a linearization about the mean, 30%, introduces a maximum error of less than 10% of the mean value, or less than a 0.5% error in moisture prediction. This is illustrated in Fig. 2. The result of this error would be the appearance of small deviations in the actual moisture output of the extruder. Note that for more typical moisture variations of $\pm 1\%$, the error due to the linear approximation would be very small. Throughout this work we will use this linearized model of the process.

The linearization of the process model as discussed above is often times useful and a relatively standard practice, but not necessarily a requirement. The nonlinearized model can usually be realized with hardware, digital or analog, and/or programming. A detailed description of such an

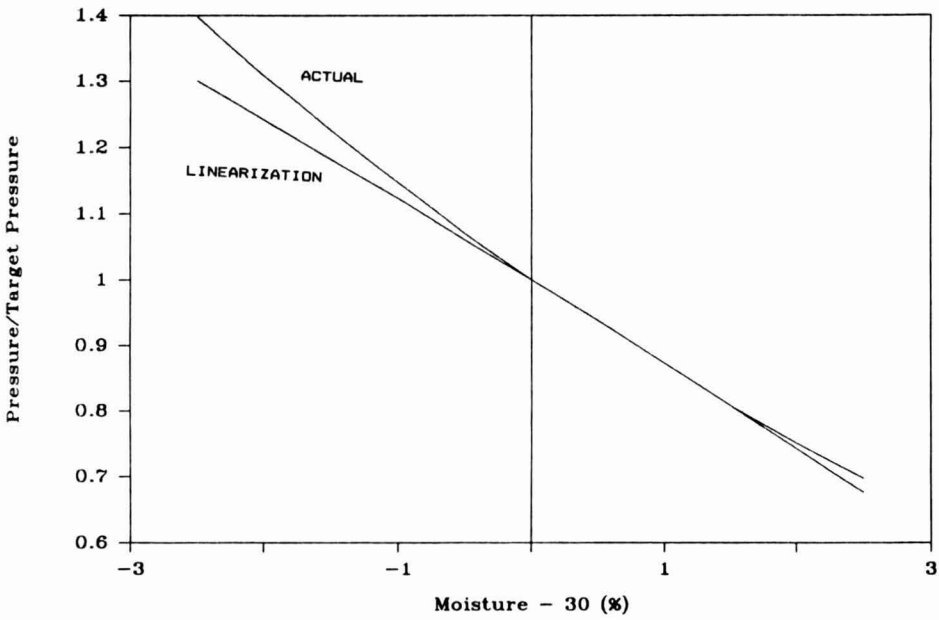

FIG. 2. LINEARIZATION OF PROCESS GAIN.

implementation is beyond the scope of this article, and is only mentioned for information's sake.

CONTROL MODELS

A typical conventional continuous contol loop is described in Fig. 3. The discrete dead-time compensation controller is described in Fig. 4. This model requires some explanation as to how it behaves.

The predictor is essentially a model of the process, but it incorporates memory. It "remembers" all changes in the controller output that have occurred in the past, and feeds back to the external loop that percentage of change due to controller action which has not yet occurred. This results in the controller reacting to only errors that are the result of external upsets, changes in set point, and model errors. This prevents the controller from taking multiple actions to an error that it has already reacted to, but has not yet been seen at the process output, because of the system dead-time. This is best illustrated in Fig. 5 for a very simple process which is composed of a pure delay and a process gain of unity.

The reader can see that even for a complicated process time response, the predictor calculations may be easily executed by a microprocessor, thus making the installation of such a control strategy a relatively easy and inexpensive task.

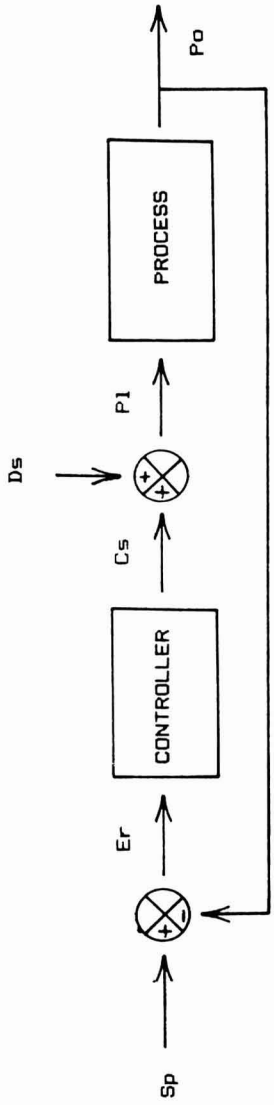

FIG. 3. CONVENTIONAL CONTROL LOOP.

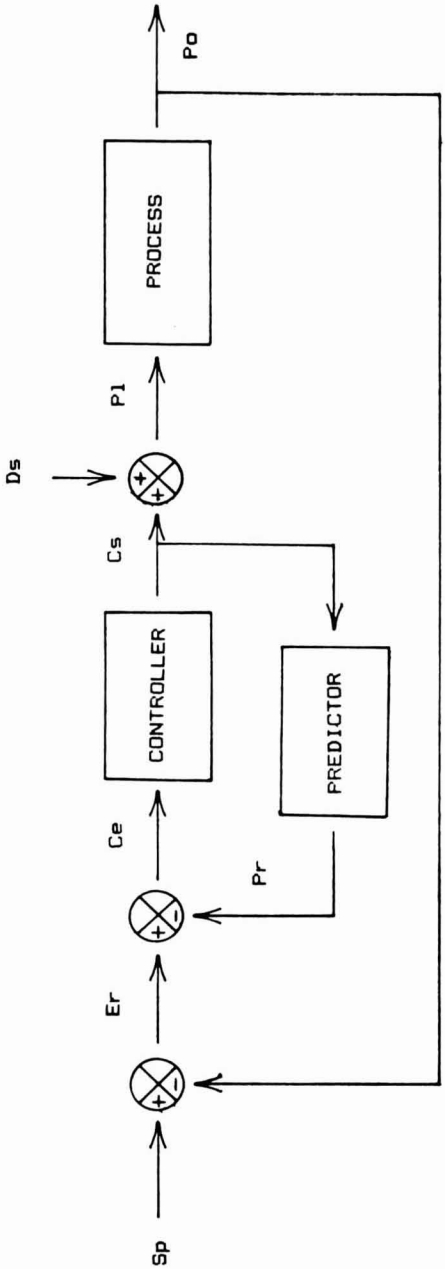

FIG. 4. CONTROL LOOP WITH DEAD TIME COMPENSATION.

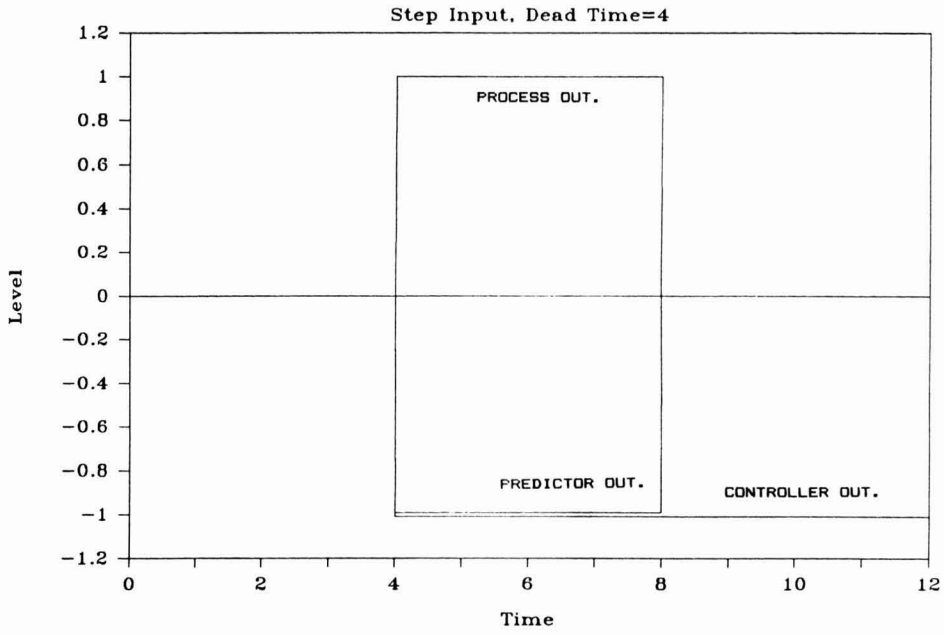

FIG. 5. COMPENSATOR FOR PURE DELAY.

THE CONTROL SCHEME

The proposed control scheme is illustrated in the block diagram of Fig. 6. This control scheme emphasizes the adaptability to an existing ratio control scheme. The ratio controller, R(s), controls the gross behavior of the water feed pump. An initial moisture measurement is made of the dry feed material, such as flour, and the correct proportion of water is determined and then with the flour flow rate is set for the ratio controller. The signal from a pressure transducer at the die of the extruder is fed back to the predictive controller shown as that portion of the block diagram consisting of C(s) and (1-exp-ts)G(s). Since we are measuring pressure, it is necessary in order to generate an error signal to convert the moisture setting, Spc, into an equivalent pressure signal. This converter is shown as the P(s) block. The predictive control signal is then used to fine-tune the water feed control signal from the ratio controller to the water feed pump.

If a ratio controller is not already in place, a modification of the above control scheme can be used. This is shown in Fig. 7. In this case the ratio control function and the moisture to pressure conversion are combined into a single realization. The overall scheme is still the same, the fine-tuning of the water feed control signal via die pressure feed back.

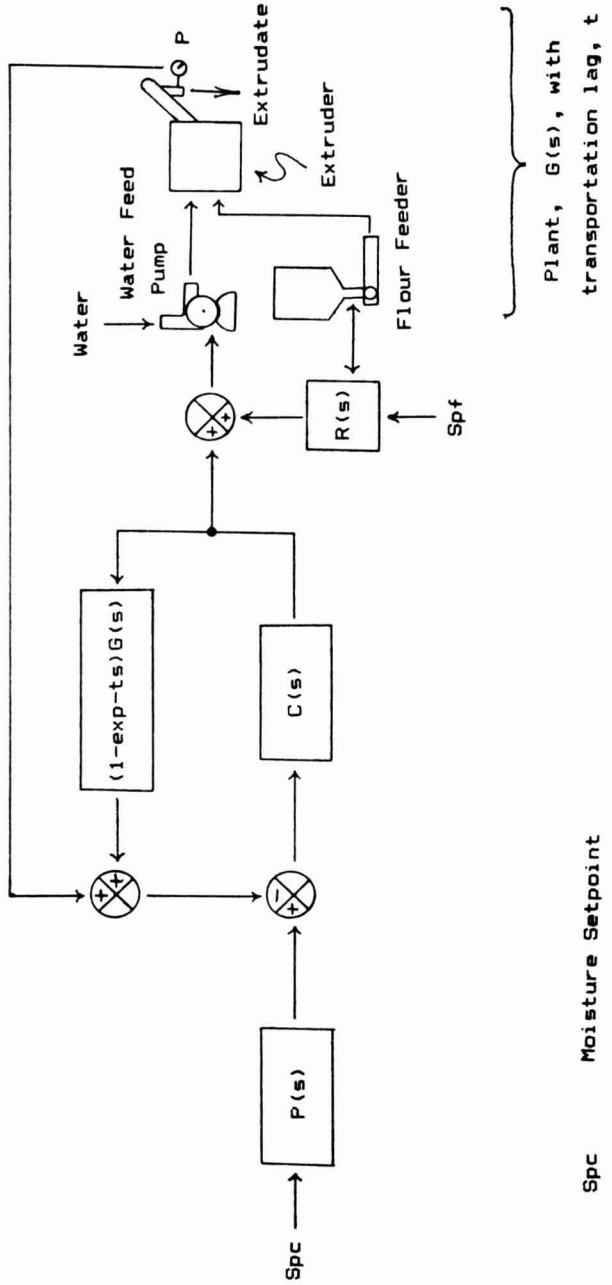

FIG. 6. PROPOSED CONTROL SCHEME.

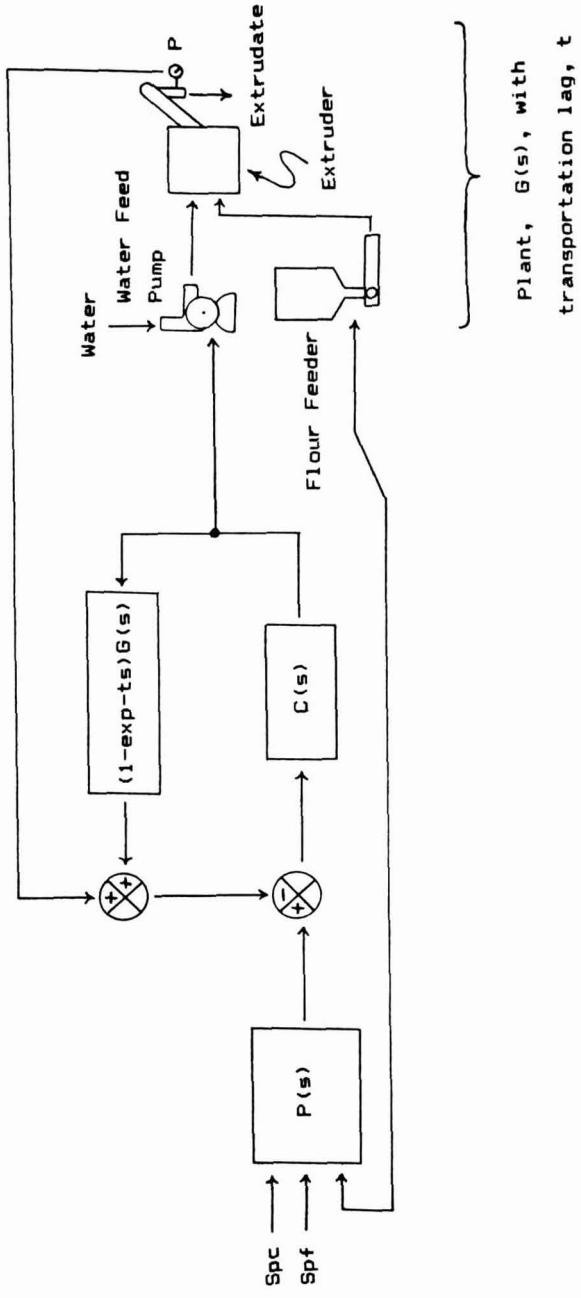

FIG. 7. MODIFIED CONTROL SCHEME.

The complete moisture controller consists of the functions in the P(s), C(s), (1-exp-ts)G(s), and R(s) blocks. It has three inputs, the moisture set point, the flour flow rate with estimated percent moisture, and the die pressure feed back signal. The controller has two outputs, the water and flour feed rate signals.

COMPARISON OF AN IDEAL DEAD TIME COMPENSATOR TO CONVENTION FEEDBACK CONTROL

To aid in simulation of the conventional controller and the dead time compensator, we have used a simulation package SLAMIIPC (Pritsker 1984) suitable for use on an IBM personal computer.

In executing this simulation, discrete convolutions of system inputs with data obtained from die studies were employed. In this way no system equations were necessary and we avoided errors due to approximating system equations from experimental data. The only real source of error that could be encountered by this approach would come from interpolations of numbers between actual data points, and these would be small with high enough data sampling rates.

For this part of the simulation, we introduced a unity step change in moisture as an upset to the process. This upset would be typical of that produced by a change in raw material moisture. For a proportional gain of unity the output moisture of the process is illustrated for the compensated system and the conventional control loop. In Fig. 8, we see that the conventional, albeit simple, control strategy exhibits offset, as would be expected with a proportional control system. More importantly, it exhibits oscillations of the process output which persist for a long time. The predictive controller exhibits a fast stable response.

If proportional gains greater than unity are used with the conventional controller, the process rapidly becomes more and more oscillatory, as is illustrated in Fig. 9.

The use of a high gain controller in the dead-time compensated system results in a very different form of response. For proportional controller gains other than unity it exhibits significant offset. This is illustrated in Fig. 10. Offsets can be eliminated using controller designs other than the proportional controller. A classical method for achieving reset is the addition of integral control action to the proportional controller.

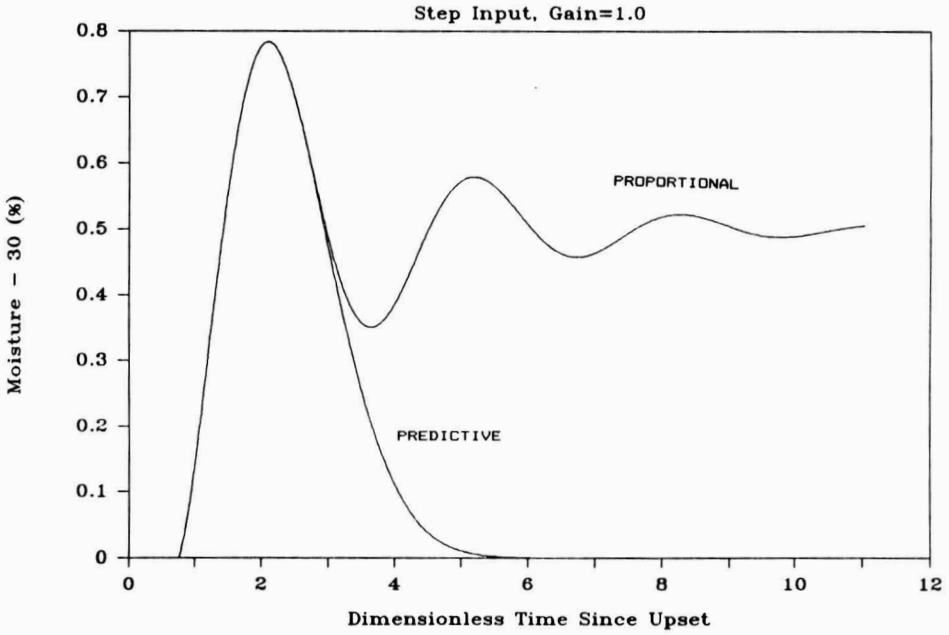

FIG. 8. PREDICTIVE VS. PROPORTIONAL CONTROL.

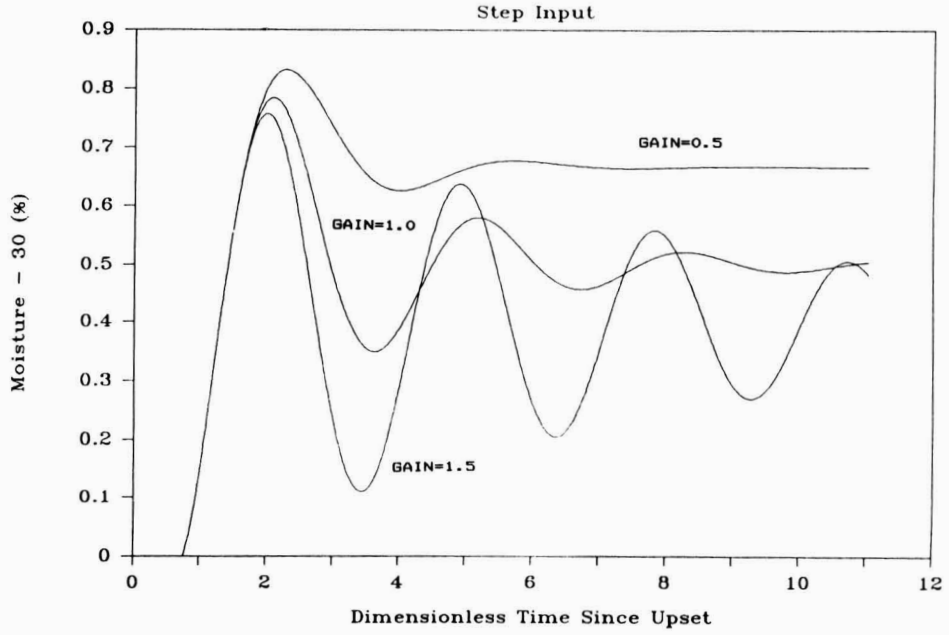

FIG. 9. GAIN IN PROPORTIONAL CONTROL.

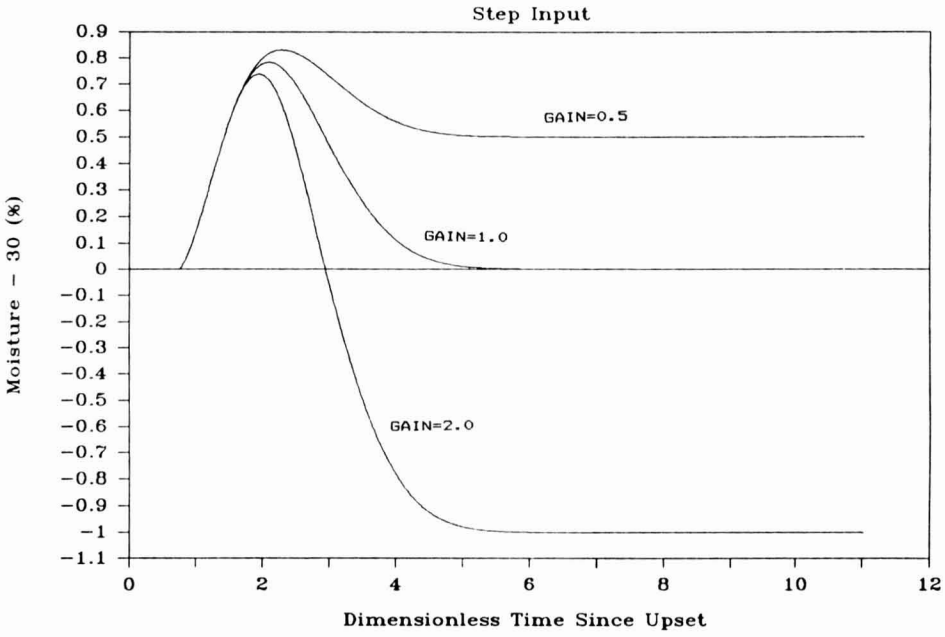

FIG. 10. PREDICTOR CONTROLLER GAIN.

PERFORMANCE OF THE REAL DEADTIME COMPENSATOR

It should be obvious that the performance of the compensator is highly dependent on the quality of model estimates. The literature (Herget *et al.* 1980; Palmor 1980; Bahil 1983) discusses this problem at length. Problems arise from errors in estimating the process model. We will now endeavor to illustrate the effect of modeling errors on the controller behavior. Such information is essential for us to estimate the true stability of the system, and to estimate the magnitude of effort required to experimentally establish the process model.

Another area, which must be considered, is the sampling rate of the control system. In the examples illustrated thus far we have used a high rate of sampling, a sampling period equal to 1.1% of the mean residence time of the extruder. It is important in the design of a digital controller, particularly those based on a microprocessor, to insure sufficient processing speed to keep the sampling rate at a high enough level. It is also important to insure sufficient memory for signal processing accuracy and for the realization of control algorithms. This would become especially important if one microprocessor was controlling several processing lines simultaneously.

To assess the effect of errors in estimating the process dead time, process lag, and gain, we have rerun the simulation with intentional errors in the models used by the predictor.

Figures 11 and 12 illustrate the results of simulations of model dead-time errors. These figures illustrate that the controller provides good control for relatively large errors in the estimate of dead-time. Underestimating the deadtime results in a small degree of overshoot in the process output. Overestimating the process dead-time reduces the rate of system response. Very satisfactory control is obtained even with model errors as large as 30%.

An error in the model can produce instability. Figure 13 illustrates that this occurs when the process gain is underestimated by 67%. Underestimation of the process gain by 33%, still results in excellent process control. Overestimating the process gain results in sluggish response, as is illustrated in Fig. 14.

Large errors in the prediction of the lag time of the process have relatively little effect on the controller performance. Figure 15 indicates that overestimates of process lag results in small offsets in process output. These offsets will disappear if the simulation is run for a long enough time. Figure 16 indicates that underestimates of the process lag results in small amounts of overshoot. In either case, errors in the model as large as 30%, still result in excellent control behavior.

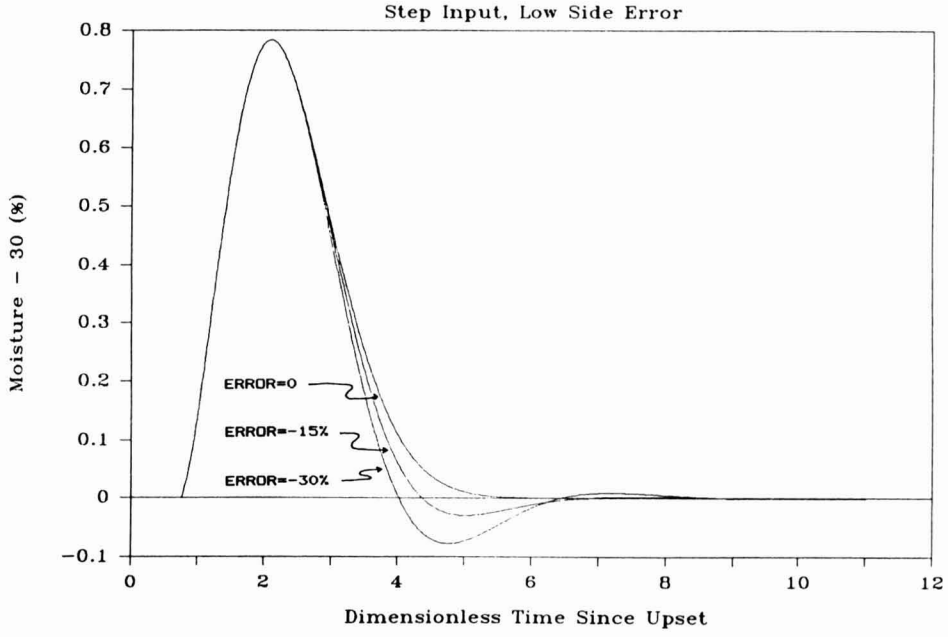

FIG. 11. MODEL DELAY ERROR.

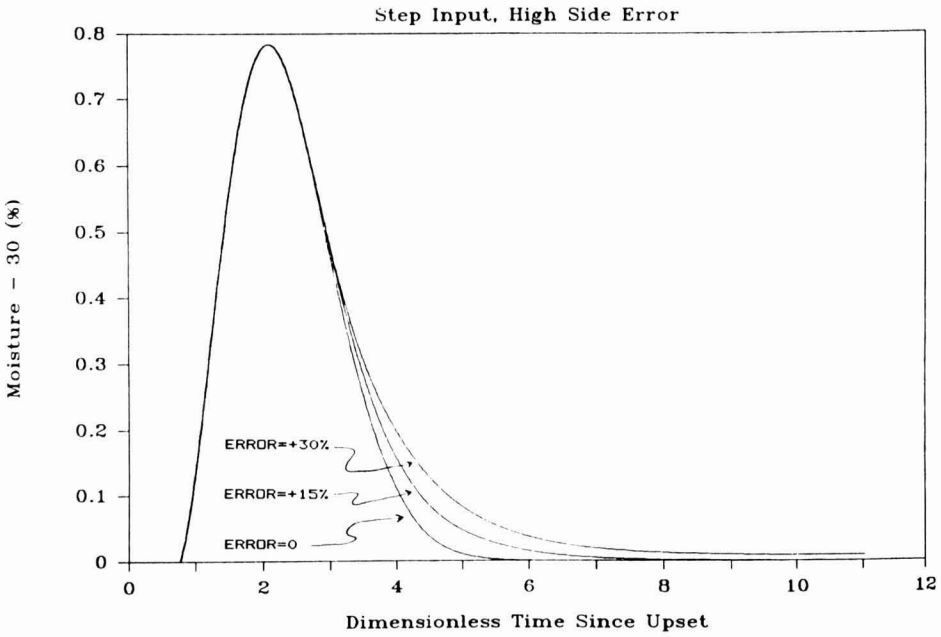

FIG. 12. MODEL DELAY ERROR.

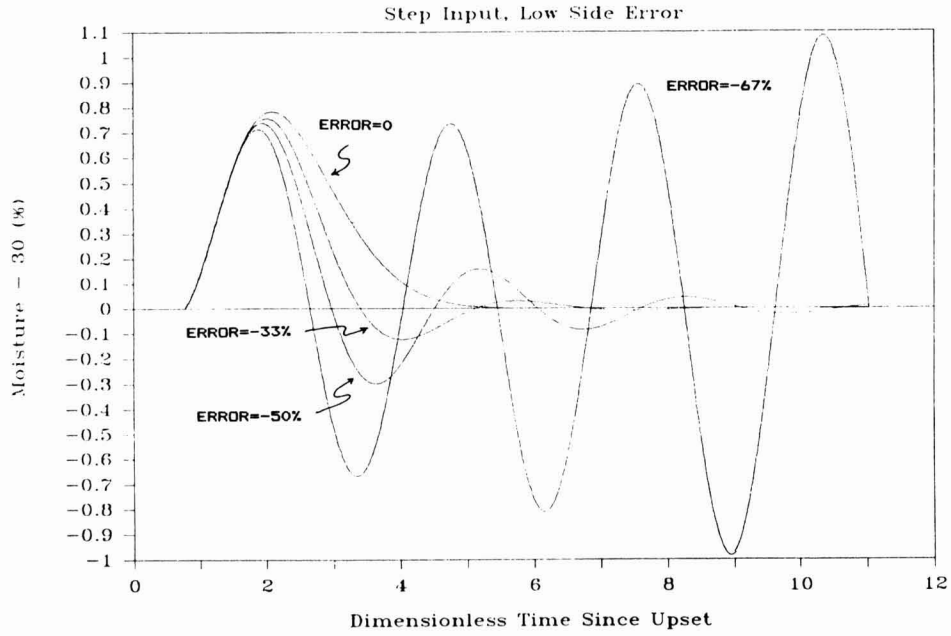

FIG. 13. MODEL GAIN ERROR.

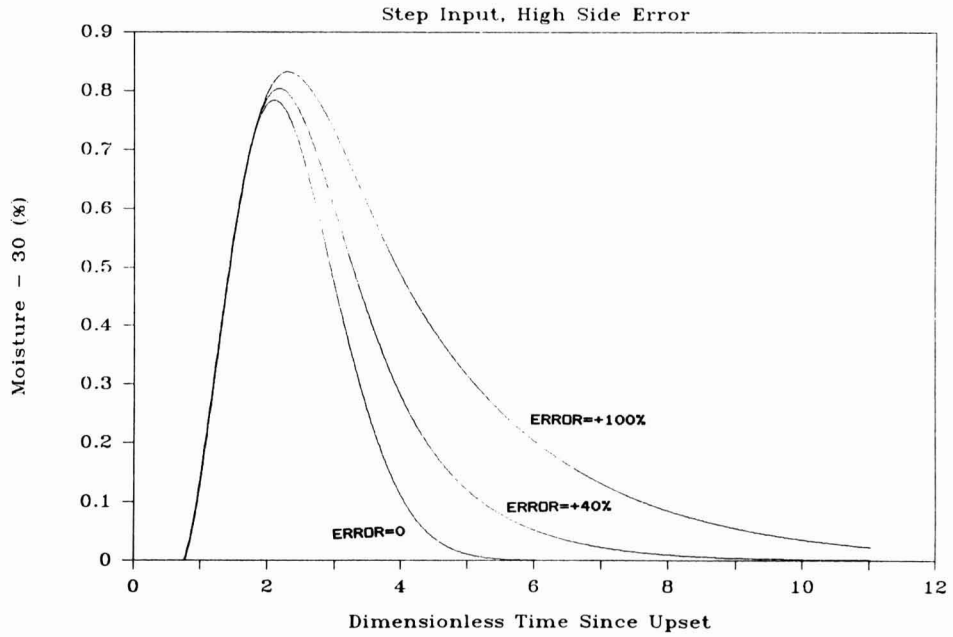

FIG. 14. MODEL GAIN ERROR.

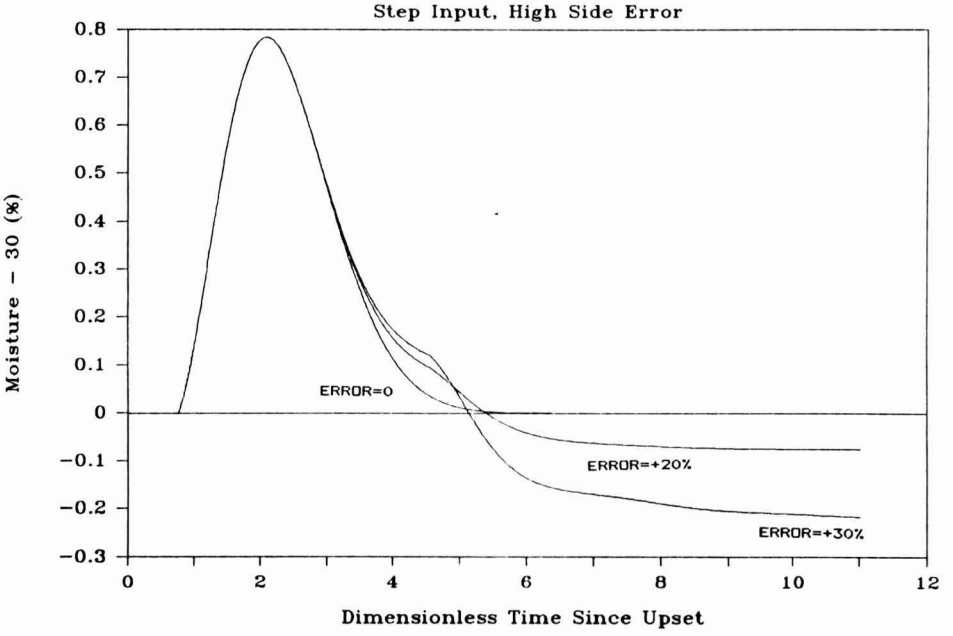

FIG. 15. PREDICTOR LAG ERROR.

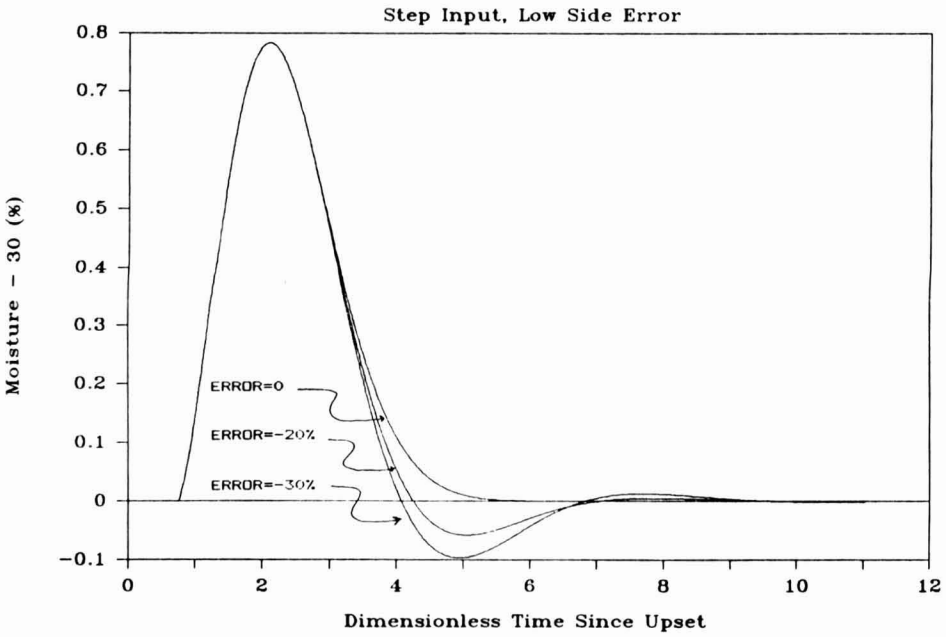

FIG. 16. PREDICTOR LAG ERROR.

It would seem that a great deal of effort is not required to obtain process models, since 30% errors in model estimation result in only a small deterioration in process control.

Decreased sampling frequency has relatively little effect on the controller performance. Figure 17 illustrates that increasing the sampling period to 22%, or even 44% of the mean residence time results in satisfactory control.

CONCLUSIONS

The use of die pressure feedback to control extrudate moisture is a realistic approach. Simulation of the use of a dead-time compensating controller to control moisture in food extruders illustrates that this control strategy offers significant advantages over conventional control strategies. In response to step change upsets, this control strategy results in less oscillation and faster response of process output than can be obtained with conventional control strategies.

The strategy is somewhat sensitive to modeling error, but not overly so, indicating that only small expenditures of time and effort are required to determine model parameters.

The magnitude of calculation required to implement the control strategy, indicates that relatively inexpensive microprocessors could be used for

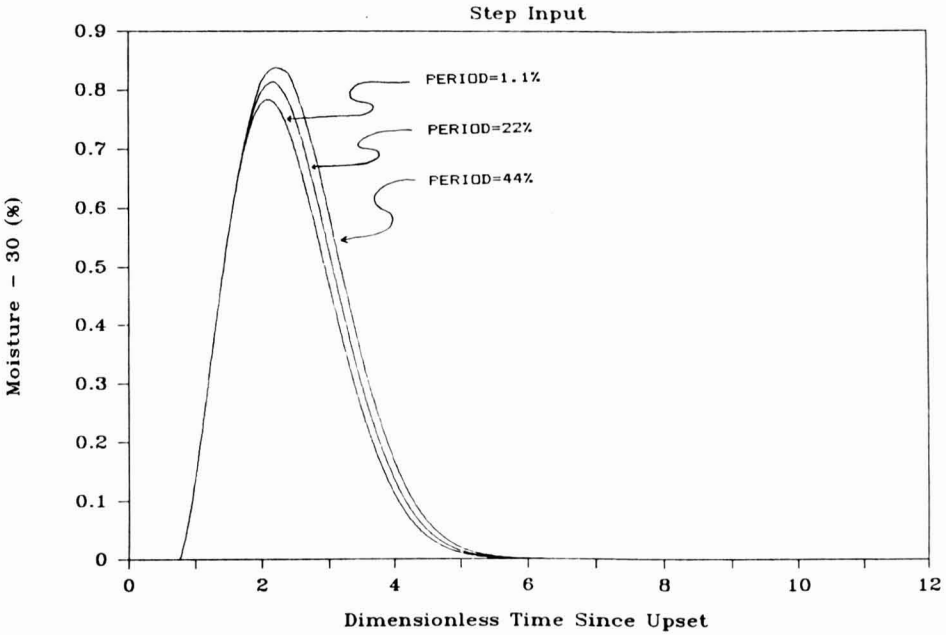

FIG. 17. PREDICTOR SAMPLING FREQUENCY.

this task. It is not inconceivable that one microprocessor could serve several process lines simultaneously.

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OPTIMUM ECONOMIC PIPE DIAMETER FOR PUMPING HERSCHEL-BULKLEY FLUIDS IN LAMINAR FLOW[1]

EDGARDO J. GARCIA
and
JAMES F. STEFFE[2]

Department of Food Science and Human Nutrition
Department of Agricultural Engineering
Michigan State University
East Lansing, MI 48824

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ABSTRACT

A method for determining optimum pipe diameter, for which total pumping system cost is minimum, has been derived for the transport of Herschel-Bulkley (H-B) fluids (power-law fluids with a yield stress) in laminar flow. The method accounts for pipe system cost as a function of diameter, and pump station and operating costs as a function of power requirements. The optimum diameter can be estimated given rheological properties, fluid density, mass flow rate and economic parameters. Optimum diameter does not depend on system elevation or pressure energy difference when a linear relationship is used for pump station cost. Friction loss in fittings and valves can be ignored when the pipe length is much greater than the pipe diameter. Pump station cost has less influence than operating cost in determining the optimum diameter.

INTRODUCTION

A problem associated with the design of fluid handling systems is the selection of tube or pipe size. The cost of process piping can be as much as 60% of the total cost of the plant processing equipment (Wright 1950; Skelland 1967; Perry and Chilton 1973). Therefore, it is important to design pipe sizes that require minimum total cost for pumping systems while

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[1]Michigan Agricultural Experiment Station Journal Article No.. 11782.
[2]Direct correspondence to Dr. Steffe, Department of Agricultural Engineering. Telephone: (517) 353-4544.

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meeting desired operating conditions and performance requirements. A criterion often used in this selection is the least annual cost. The least annual cost method is based on an economic balance of the pipe system cost, pump station cost and operating or electrical power cost. The pump station investment and operating costs vary directly with the pumping power requirement. For a given flow rate, power requirements decrease with increasing pipe diameter since the pressure drop due to friction varies inversely with pipe size. Consequently, the pump station cost and operating cost decrease with increasing pipe diameter. Conversely, the pipe system cost increases with increasing pipe diameter; therefore, there is a pipe diameter for which the sum of these costs is a minimum. This is known as the optimum economic pipe diameter (Downs and Tait 1953; Skelland 1967; Darby and Melson 1982).

Analytically, the optimum economic pipe diameter can be obtained by equating to zero the derivative of the total system cost with respect to the pipe diameter, and solving for the diameter (Skelland 1967; Jelen 1970). Various relationships have been developed to estimate optimum economic pipe diameter for systems handling Newtonian fluids under laminar and turbulent flow conditions. Genereaux (1937) was one of the first to present pipe diameter optimization methods based on the economic balance of the pipe and the operating costs. Further details of the work of Genereaux are given by Peters and Timmerhaus (1968). Downs and Tait (1953) based their analysis on the economic balance of the pipe and pump costs and provided correction factors to account for operating costs. Perry and Chilton (1973), and Peters and Timmerhaus (1968) presented optimum diameter relationships based on the concept of return on incremental investment. Other methods for determining economic pipe diameter for Newtonian fluids are discussed by Wright (1950), Sarchet and Colburn (1940), Nolte (1978), Dickson (1950), Braca and Happel (1953) and Nebeker (1979).

Optimum economic diameter relationships are more limited for non-Newtonian fluids. Duckham (1972) gave general guidelines to estimate the optimum diameter for non-Newtonian fluids. Skelland (1967) developed optimum diameter equations based on Metzner and Reed (1955) and Dodge and Metzner (1959) friction factor relationships for non-Newtonian fluids in laminar and turbulent flow, respectively. For laminar flow his relationship may be written in terms of the power law model. The analysis is based on the economic balance of pipe and operating costs, neglecting pump cost variability. Skelland (1967) also developed a relationship to estimate the optimum pumping temperature based on an economic balance of heating cost and operating cost. The latter decreases with increasing temperature due to the decrease of the consistency coefficient (or viscosity for Newtonian fluids) with increasing temperature. Application of the relationships

of Skelland to the food processing industry was presented by Boger and Tiu (1974b).

Recently, Darby and Melson (1982) used dimensional analysis to developed graphs from which the optimum pipe diameter could be obtained directly for Newtonian, Bingham plastic and power law fluids. In this analysis, the friction factor relationships of Churchill (1977) and Darby and Melson (1981) were used to estimate the frictional pressure drop for Newtonian and Bingham plastic fluids, respectively, covering all flow regimes. The equation of Dodge and Metzner (1959) was used for the turbulent flow of power law fluids. Unlike Skelland, their economic analysis includes the pump station cost for which they developed a linear relationship with power requirements. However, Darby and Melson (1981) assumed the friction factor to be constant in the differentiation of the total cost with respect to the pipe diameter. To date, a method to determine optimum pipe diameter for H-B fluids has not been available.

The objective of this work is to develop equations by which the optimum economic pipe diameter can be determined for a system transporting H-B fluids under laminar flow. The H-B model was selected due to its generality and wide application to fluid foods as well as other fluid materials (Holdsworth 1971; Higgs and Norrington 1971; Steffe *et al.* 1983; Boger and Tiu 1974a). Non-Newtonian flow behavior must be considered when designing pumping systems handling fluids of this type (Cheng 1975; Johnson 1982). Failure to do so may lead to a over or under sized system which is inefficient to operate or more costly to erect (Steffe 1983; Nolte 1978). In addition, it should be noted that the vast majority of pipe line design problems for non-Newtonian foods involve laminar flow conditions because very large (and uneconomical) pressure drops are required to achieve turbulence. Consult Garcia (1985) for additional development and the solution to pipeline design problems for H-B fluids in turbulent flow.

THEORETICAL DEVELOPMENT

Pumping System Cost

The total cost of a pumping system consists of three components: the pipe system cost, the pump station cost and the operating cost (Darby and Melson 1982). Pipe system cost primarily consists of the cost of pipe, fittings, valves and installation. In most cases, pipe system cost can be estimated as (Jelen 1970; Darby and Melson 1982)

$$C_{pc} = CD^s \tag{1}$$

The constants, C and s, can be obtained from a logarithmic plot of installed cost per unit pipe length versus the pipe inside diameter. Equation (1) should be evaluated using a range of diameters on either side of the optimum pipe diameter. However, when available cost data for the pipe system is not sufficient, C and s can be estimated from cost data of one size pipe (a one or two inch size is commonly used) and appropriate fittings (Genereaux 1937; Skelland 1967; Peters and Timmerhaus 1968; Nebeker 1979). This is done by letting $s = y$ and $C = (1 + R)\, X/(D_1)^y$ in Eq. (1) where X is pipe purchase cost (for diameter equal to D_1) per unit length ($/m). R is the ratio of total cost for fittings and installation of pipe and fittings to the purchase cost of new pipe, and y is an empirical constant. In this case, y only depends on pipe material. Typical values of y are given by Nolte (1978). R is estimated at the reference pipe size (D_1) and is assumed to be independent of pipe diameter. Hence, this method should be used only in preliminary pipe sizing when data and knowledge of the system are limited.

The total annual cost per unit lenght of installed pipe system is

$$C_{pi} = (a+b)CD^s \tag{2}$$

Notice that the annual fixed cost (a) and the annual maintenance cost (b) are assumed to be independent of pipe diameter. However, these costs, as well as other costs associated with the pipe system (which may depend on the pipe diameter), may be included in the estimation of the installed cost of the pipe system for the various pipe diameters. In this case, a and b will be included in the variables C and s. Then, (a + b) in Eq. (2), would be set equal to one.

Pump station cost mainly consists of the cost of pump, motor and installation. It can be expressed in terms of power requirements:

$$C_{ps} = AP^m + B \tag{3}$$

Equation (3) permits the use of a linear ($m = 1$) or logarithmic ($B = 0$) relationship between installed pump station cost and power requirements. Equation (3) can also be interpreted as the sixth-tenths factor rule ($m = 0.6$) or similar relation if one lets $m = q$, $B = C_I$ and $A = C_D/(P_1)^q$ where q is the power of the pump of known cost, and C_D and C_I are the direct and indirect cost respectively of a pump of size P_1 (Perry and Chilton 1973). This interpretation permits the estimation of the constant of Eq. (3) from the cost of a given pump size. Typical values of q for different pump types and power ranges are given by Jelen (1970), Peters and Timmerhaus (1968) and Perry and Chilton (1973).

The total annual cost of the pump station per unit length of pipe can then be expressed as

$$C_{pu} = (a'+b')(AP^m + B)/L \tag{4}$$

Again, the fixed cost (a') and the maintenance cost (b') may be included in the estimation of the installed cost of the pump station for different pump sizes accounting for these costs in the variables A, B and m. Then, the term (a' + b') in Eq. (4) would be set equal to one. Cost data for pipes, pumps and fittings are presented by Peters and Timmerhaus (1969), Jelen (1970), Marshall and Brandt (1974) and Barret (1981). Current data should be used whenever possible.

The cost of operation is primarily the annual electrical energy consumption:

$$C_{op} = \frac{C_e h P}{L} \tag{5}$$

The total cost of a pumping system per unit length of pipe (C_T) is obtained by adding Eq. (2), (4) and (5) giving, after rearrangement,

$$C_T = (a+b)CD^s + \frac{C_e h P}{L}\left[\frac{(a'+b')(AP^m + B)}{C_e h P} + 1\right] \tag{6}$$

Equation (6) allows calculation of the total cost as a function of inside pipe diameter and power requirements.

Power Requirements

For a given mass flow rate, power required to pump a fluid a given distance increases with decreasing pipe diameter. The work per unit mass required to pump an incompressible fluid through a pipe system is given by the mechanical energy balance equation:

$$W = F + \frac{\Delta p}{\rho} + g\Delta z + \Delta KE \tag{7}$$

The total energy loss due to friction can be written in terms of the Fanning equation and the summation of the energy loss in fittings and other devices in the line as (Steffe *et al.* 1984)

$$F = \frac{2f\bar{V}^2L}{D} + \Sigma K\frac{\bar{V}^2}{2} \tag{8}$$

The mass average velocity is

$$\bar{V} = \frac{4M}{\pi\rho D^2} \tag{9}$$

Substituting Eqs. (8) and (9) into Eq. (7) and assuming negligible kinetic energy change gives

$$W = \frac{32fM^2L}{\pi^2\rho^2D^5} + \frac{8M^2}{\pi^2\rho^2D^4}\Sigma K + \frac{\Delta p}{\rho} + g\,\Delta z \tag{10}$$

The power requirement can then be expressed as

$$P = \frac{MW}{E} \tag{11}$$

Laminar Flow of Herschel-Bulkley (H-B) Fluids

The flow behavior of many fluid foods and other industrially important fluids may be described by the H-B model which can be written as (Herschel and Bulkley 1926)

$$\tau = \tau_o + \kappa\dot{\gamma}^n \tag{12}$$

This model simplifies to other well known models: pseudoplastic fluids for $\tau_o = 0$ and $0 < n < 1$; dilatant fluids for $\tau_0 = 0$ and $n > 1$; Bingham plastic fluids for $n = 1$ and $\kappa = \eta$ (plastic viscosity); and Newtonian fluids for $\tau_o = 0$, $n = 1$ and $\kappa = \mu$ (viscosity).

Equation (12) can be integrated to obtain a relationship between the frictional pressure drop and flow rate (Cheng 1970; Charm 1978; Skelland 1967; Govier and Azis 1972). It can also be written in terms of the Fanning friction factor (Hanks 1978):

$$f = \frac{16}{\psi\,Re} \tag{13}$$

where Re is the generalized Reynolds number (by definition),

$$\mathrm{Re} = \frac{D^n \bar{V}^{2-n} \rho}{8^{n-1} \kappa} \left[\frac{4n}{1+3n}\right]^n \tag{14}$$

and

$$\psi = (1+3n)^n (1-\xi_o)^{1+n} \left[\frac{(1-\xi_o)^2}{(1+3n)} + \frac{2\xi_o(1-\xi_o)}{(1+2n)} + \frac{\xi_o^2}{(1+n)}\right] \tag{15}$$

where ξ_o, the dimensionless unsheared plug radius, is

$$\xi_o = \frac{\tau_o}{\tau_w} = \frac{\tau_o}{\left(\frac{D\Delta P}{4L}\right)} = \frac{\tau_o}{\frac{1}{2} f \rho \bar{V}^2} \tag{16}$$

ξ_o can also be written as an implicit function of Re and a generalized Hedstrom number (He) (Hanks 1978):

$$\mathrm{Re} = 2\mathrm{He}\left(\frac{n}{1+3n}\right)^2 \left(\frac{\psi}{\xi_o}\right)^{(2/n)-1} \tag{17}$$

where

$$\mathrm{He} = \frac{D^2 \rho}{\kappa} \left(\frac{\tau_o}{\kappa}\right)^{(2/n)-1} \tag{18}$$

To calculate the friction factor for Bingham plastic and H-B fluids, ξ_o is estimated through iteration of Eq. (17) using Eqs. (9), (14), (15) and (18). Then, f can be calculated using Eqs. (13), (14) and (15). For pseudoplastic, dilatant and Newtonian fluids, $\xi_o = 0$ and $\psi = 1$; hence, f can be computed directly using Eqs. (13) and (14).

The critical Reynolds number (Re_c), where the laminar flow ends, is given by (Hanks and Ricks 1974)

$$\mathrm{Re}_c = \frac{6464n\psi_c^{(2/n)-1}(2+n)^{\left(\frac{2+n}{1+n}\right)}}{(1+3n)^2(1-\xi_{oc})^{(2/n)+1}} \tag{19}$$

where ξ_{oc} is given as an implicit function of He and n:

$$He = \frac{3232(2+n)^{\left(\frac{2+n}{1+n}\right)} \xi_{oc}^{(2/n)-1}}{n(1-\xi_{oc})^{(2/n)+1}} \tag{20}$$

and ψ_c is calculated from Eq. (15) with $\xi_o = \xi_{oc}$

Optimum Pipe Diameter

The optimum pipe diameter (D_{opt}) can be obtained by differentiating C_T [Eq. (6)] with respect to diameter (D) and setting the resulting equation equal to zero ($dC_T/dD = 0$). Using Eqs. (6), (10) and (11), this operation yields,

$$\frac{dC_T}{dD} = (a+b)\, sCD_{opt}^{s-1} - \frac{32M^3 C_e h}{\pi^2 \rho^2 D^6 E} \left\{ \frac{(a'+b')mA(MW)^{m-1}}{C_e h E^{m-1}} + 1 \right\} \tag{21}$$

$$\left[5f - D_{opt} \frac{df}{dD} + \frac{D_{opt}}{L} \Sigma K - \frac{D_{opt}^2}{4L} \Sigma \frac{dK}{dD} \right]$$

The derivative of f with respect to D is found from Eq. (13) as

$$\frac{df}{dD} = -\frac{16}{\psi Re^2} \frac{df}{dD} - \frac{16}{\psi^2 Re} \frac{d\psi}{dD} \tag{22}$$

Replacing $\bar{V}$ with Eq. (9), in Eq. (14), and taking the derivative of Re with respect to D gives

$$\frac{dRe}{dD} = \frac{(3n-4)}{D} Re \tag{23}$$

Similarly, substituting ξ_o with Eq. (16), in Eq. (15), and taking the derivative of ψ with respect to D gives

$$\frac{d\psi}{dD} = \frac{\psi \phi \xi_o}{f} \frac{df}{dD} - \frac{4\psi\phi\xi_o}{D} \tag{24}$$

where (25)

$$\phi = \frac{(1+3n)(1+n)(1-\xi_o)^2 + 2\xi_o(1+2n)(1+3n)(1-\xi_o) + \xi_o^2(1+3n)(1+2n)(1+n)}{(1+2n)(1+n)(1-\xi_o)^3 + 2\xi_o(1+3n)(1+n)(1-\xi_o)^2 + \xi_o^2(1+3n)(1+2n)(1-\xi_o)}$$

Substituting Eqs. (23) and (24) into Eq. (22) and solving for df/dD, results a useful form of the derivative:

$$\frac{df}{dD} = \frac{64\xi_o\phi + 16(4-3n)}{Re\psi D(1+\phi\xi_o)} \tag{26}$$

Darby and Melson (1982) assumed df/dD = 0 in their development of optimum pipe diameter graphs. This assumption is valid if df/dD is much smaller than $5f/D_{opt}$ for the term $5f/D_{opt}$-df/dD which appears in Eq. (21) if that equation is divided by D_{opt}. However, for power law fluids in laminar flow, df/dD can vary from 20% of 5f/D for n = 1 to 74% of 5f/D for n = 0.1. For H-B fluids, df/dD can be as much as 68% of 5f/D depending on the fluid properties and flow conditions (Garcia 1985). Therefore, the assumption of df/dD << 5f/D may introduce significant errors under certain conditions. In searching for simplifications for the D_{opt} equation, friction loss coefficients must also be considered. For laminar flow, K depends on the fluids properties and increases significantly with decreasing Reynolds number, that is, increasing D for a given mass flow rate (Steffe *et al.* 1984). However, there is not a general equation for K values of different fittings. Therefore, K must be assumed to be independent of the pipe diameter, dK/dD = 0. This assumption will not seriously influence our results because when L >> D, L/D and $D^2/4L$ will be small numbers; hence, friction losses from values and fittings will have a small influence on D_{opt} as seen from Eq. (21). For this reason K values for Newtonian fluids (constants measured under turbulent flow conditions) are used as an approximation to evaluate D_{opt}. Then, substituting Eq. (26) into Eq. (21), using Eq. (11), eliminating the dK/dD term and solving for D_{opt} gives

$$D_{opt} = \left\{\left[\frac{32\dot{M}^3 C_e h}{(a+b)sC\pi^2\rho^2 E}\right]\left[\frac{(a'+b')\dot{m}AP^{m-1}}{C_e h} + 1\right]\right.$$
$$\left.\left[\frac{16(1+3n) + 16\xi_o\phi}{Re\psi(1+\xi_o\phi)} + \frac{D_{opt}}{L}\Sigma K\right]\right\}^{\frac{1}{s+5}} \tag{27}$$

The ΣK term may also be eliminated when L >> D and Eq. (27) can be written as

$$D_{opt} = \left\{ \left[\frac{4C_e \dot{M} h \kappa}{(a+b) s C \rho E} \right] \left[\frac{8\dot{M}(1+3n)}{\pi \rho n} \right]^n \left[\frac{1+3n+\xi_o \phi}{\psi (1+\xi_o \phi)} \right] \left[\frac{(a'+b')\dot{m} A P^{m-1}}{C_e h} + 1 \right] \right\}^{\frac{1}{s+3n+1}} \quad (28)$$

The procedure to estimate the D_{opt} is outlined in Fig. 1.

From Eqs. (27) or (28), it can be observed that, when a linear relationship (m = 1) is used for C_{ps} versus P, D_{opt} is independent of P; hence, it is also independent of Δz and Δp. In this case steps 8-10 in Fig. 1 are not required to estimate D_{opt}. Even if m is not equal to one, D_{opt} can be assumed to be independent of Δp and Δz since C_{ps} generally varies little

Input Variables		
$\dot{M}$, L, Δp, Δz, E, ΣK		Pumping system parameters
n, κ, τ_o, ρ		Fluid properties
C, s, a, b		Pipe system cost parameters
A, B, m, a', b'		Pump station cost parameters
C_e, h,		Operating cost parameters
1.	D_{est}	Guess D_{opt}
2.	$\bar{v}$	Calculate $\bar{v}$ from Equ. (9)
3.	Re	Calculate Re from Equ. (14)
4.	He	Calculate He from Equ. (18)
5.	ξ_o	Calculate ξ_o from Equ. (17) through iteration . $0 \leq \xi_o < 1$. $\xi_o = 0$ for He = 0 that is $\tau_o = 0$
6.	ψ	Calculate ψ from Equ. (15)
7.	ϕ	Calculate ϕ from Equ. (25)
8.	f	Calculate f from Equ. (13)
9.	W	Calculate W from Equ. (10)
10.	P	Calculate P from Equ. (11)
11.	D_{opt}	If Equ. (27) or(28) are true then $D_{opt} = D_{est}$, otherwise go to step 1

FIG. 1. CALCULATION SCHEME TO ESTIMATE THE OPTIMUM PIPE DIAMETER.

with D. Notice also that, if m = 1 and $L >> D_{opt}$, D_{opt} is also independent of L. In other words, D_{opt} can be estimated from the costs of a unit length of pipe. The independence of D_{opt} on Δp, Δz, ΣK and L is important because these variables may not be well known in preliminary sizing of pipe systems.

Equations (27) and (28) are only valid for laminar flow, hence, a criterion is needed for determining whether the optimum pipe diameter calculated results in laminar or turbulent flow. For a given flow rate, the critical pipe diameter (D_c) is the numerical value of D where laminar flow ends. The flow will be laminar for $D > D_c$ and turbulent for $D < D_c$. An expression for D_c can be developed, by equating the critical Reynolds number [Eq. (19)] to the definition of the Reynolds number [Eq. (14)] and replacing $\bar{V}$ with Eq. (9), as

$$D_c = \left\{ \frac{(50.5)8^n \kappa (2+n)^{\left(\frac{2+n}{1+n}\right)} \psi_{cd}^{(2/n)-1}}{(\rho n)^{n-1}(1-\xi'_{oc})^{1+(2/n)}} \left(\frac{M(1+3n)}{\pi}\right)^{n-2} \right\}^{\frac{1}{3n+4}} \quad (29)$$

where ξ'_{oc} is given implicity by Eq. (20) with $\xi_{oc} = \xi'_{oc}$ and ψ_{cd} is given by Eq. (15) with $\xi_o = \xi'_{oc}$. Notice that the pipe diameter is required to estimate He in Eq. (18): hence, ξ'_{oc} is estimated implicity by Eq. (20), requiring one to solve for D_c by iteration. Notice also that ξ_{oc} and ξ'_{oc} might have different values since these parameters are a function of the pipe diameter. For power law and Newtonian fluids, $\xi'_{oc} = 0$ and $\psi_{cd} = 1$ hence D_c can be estimated directly from Eq. (29).

Once D_c is known, Eq. (28) can be evaluated with $D_{opt} = D_c$, $\xi_o = \xi'_{oc}$, $\psi_c = \psi_{cd}$ and $\phi = \phi_{cd}$ where ϕ_{cd} is given by Eq. (25) with $\xi_o = \xi'_{oc}$. If the right hand side of Eq. (27) or (28) is greater than the left hand side, the flow will be laminar for D_{opt} and Eqs. (27) or (28) can be used. Otherwise, the optimum pipe diameter will occur in the turbulent flow regime and the methods outlined by Garcia (1985) must be utilized. It should be noted that laminar flow of non-Newtonian fluid foods is generally the rule, not the exception, in the food industry.

EXAMPLE PROBLEM FOR PUMPING TOMATO KETCHUP

The objective of this example problem is to demonstrate a method for determining the optimum tube diameter for moving tomato ketchup at a mass flow rate of 4.0 kg/s. The data for the problem are summarized in Table 1. Tomato ketchup can be considered to be a pseudohomogeneous non-Newtonian fluid described by the H-B model (Higgs and Norrington 1971) as shown in Table 1. The tube system to be designed consists of

100 m of 304 stainless steel tubing with both ends at the same elevation and pressure. The system is to have three tees (used as elbows), three 90° elbows, twenty-one union couplings and two plug valves giving an overall fittings resistance coefficient of ten. A positive displacement rotary pump is to be used assuming a pump and motor (combined) efficiency of 70%. In addition, the system is to be operated 75% of the year with an electrical energy cost of 0.06 $/kW h. All cost information has been obtained from typical manufacturers data.

Table 1. Data for example problem.

Properties of Tomato Ketchup at 25°C	n	0.27
	κ (Pa.s^n)	18.7
	τ_o (Pa)	32.0
	ρ kg/s	1110.0
Pumping system parameters	$\dot{M}$ (kg/s)	4.0
	L (m)	100.0
	ΣK	10.0
	E	0.7
	Δz	0
	Δp	0
Tube system cost parameters	C ($/m^{s+1})	2021.0
	s	1.156
	a (1/yr)	0.18
	b (1/yr)	0.10
Pump station cost parameters	B ($)	7722.0
	A ($/W^m)	0.027
	m	1.0
	a' (1/yr)	0.28
	b' (1/yr)	0.10
Operating cost parameters	C_e ($/W hr)	6.0×10^{-5}
	h (hrs/yr)	6570.0

The variation of the installed costs of the tube system, per meter length of tube, with tube diameter is shown in Table 2 and plotted in Fig. 2. The values of C and s obtained from this plot are also given in Table 1. In addition, the fixed cost (a) and maintenance (b) annual cost ratios are presented. The value of a was estimated from the uniform recovery factor equation assuming zero salvage value, an interest rate of 12% and a lifetime of 10 years. The value of b was taken as 10% of the installed cost of the tube system.

The variation of the installed cost of the pump with power is shown in Table 3 and plotted in Fig. 3. Fitting a straight line (m = 1) described by Eq. (3) gives the constants, B and A shown in Table 1. This table also

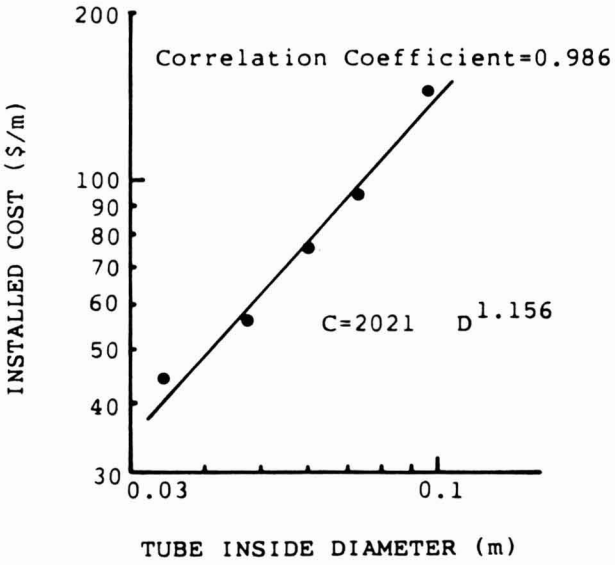

FIG. 2. VARIATION OF THE INSTALLED COST OF THE TUBE SYSTEM WITH TUBE INSIDE DIAMETER.

shows the fixed (a′) and maintenance (b′) annual cost ratios for the pump. An interest rate of 12%, zero salvage value and life time of 5 years were used to estimate a′. For the tube system, b′ was taken as 10% of the installed pump station cost.

To determine if the optimum condition occurs in laminar or turbulent flow, the critical diameter D_c was first evaluated with Eq. (29) and found to be 0.02527 m. The right hand side of Eq. (27) was then estimated with $D_{opt} = D_c$ and the variables given in Table 1. The result was 0.03923 m which is greater than D_c indicating that the flow will be laminar at D_{opt}

Table 2. Variation of the cost (including cost of fittings, valves, and approximate costs of installation) of a 304 stainless steel tube system, per meter length of tube, with tube diameter.

DIAMETER		INSTALLED COST
OD (in)	ID (m)	$/m
1 1/2	0.0348	44.24
2	0.0475	56.57
2 1/2	0.0602	76.35
3	0.0729	94.65
4	0.0974	144.71

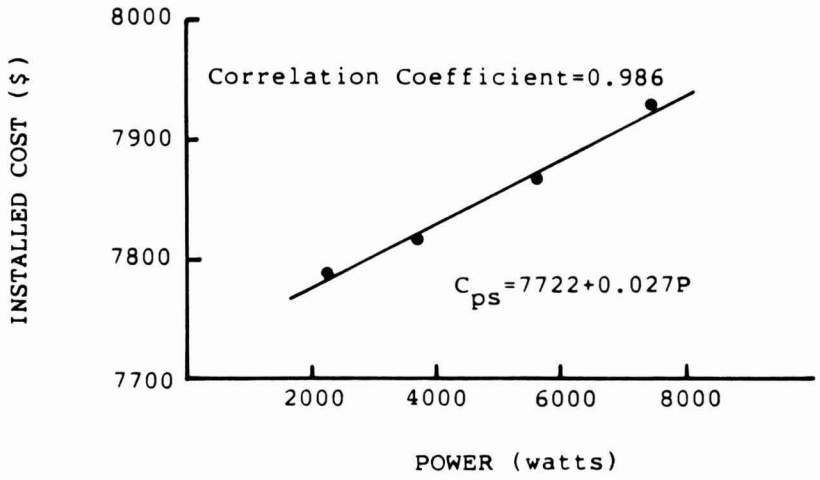

FIG. 3. VARIATION OF THE INSTALLED COST OF THE PUMP STATION WITH POWER REQUIREMENTS.

and Eqs. (27) and (28) are valid. The optimum tube diameter was then calculated (using a CDC computer) from Eq. (27) using the procedure outlined in Fig. 1. For this example, steps 8-10 are not required to estimate D_{opt} since m = 1. The pumping system costs (C_{pi}, C_{pu}, C_{op}, C_T) at D_{opt} were also estimated from Eqs. (2), (4), (5) and (6), respectively. In addition, W, Re, He, $\bar{V}$ and f at D_{opt} were calculated with results summarized in Table 4. The D_{opt} was also estimated graphically by plotting the variation of the costs with tube diameter as shown in Fig. 4. D_{opt} obtained graphically and analytically [Eq. (27)] was 0.0618 for a C_{Tmin} of \$68.5/yr m.

As seen from Fig. 4, C_{pu} varied insignificantly with tube diameter. This is to be expected since A is small and C_{ps} varies little with P. Notice also that the total cost deviates from the minimum more slowly as the diameter increases after passing an optimum. This would occur in many cases of

Table 3. Variation at the cost (including cost of motor, speed reducer, and approximate cost of installation) of a positive displacement rotary pump with power.

POWER		INSTALLED COST
HP	WATTS	$
3	2237.1	7790
5	3728.5	7815
7 1/2	5592.75	7870
10	7457	7930

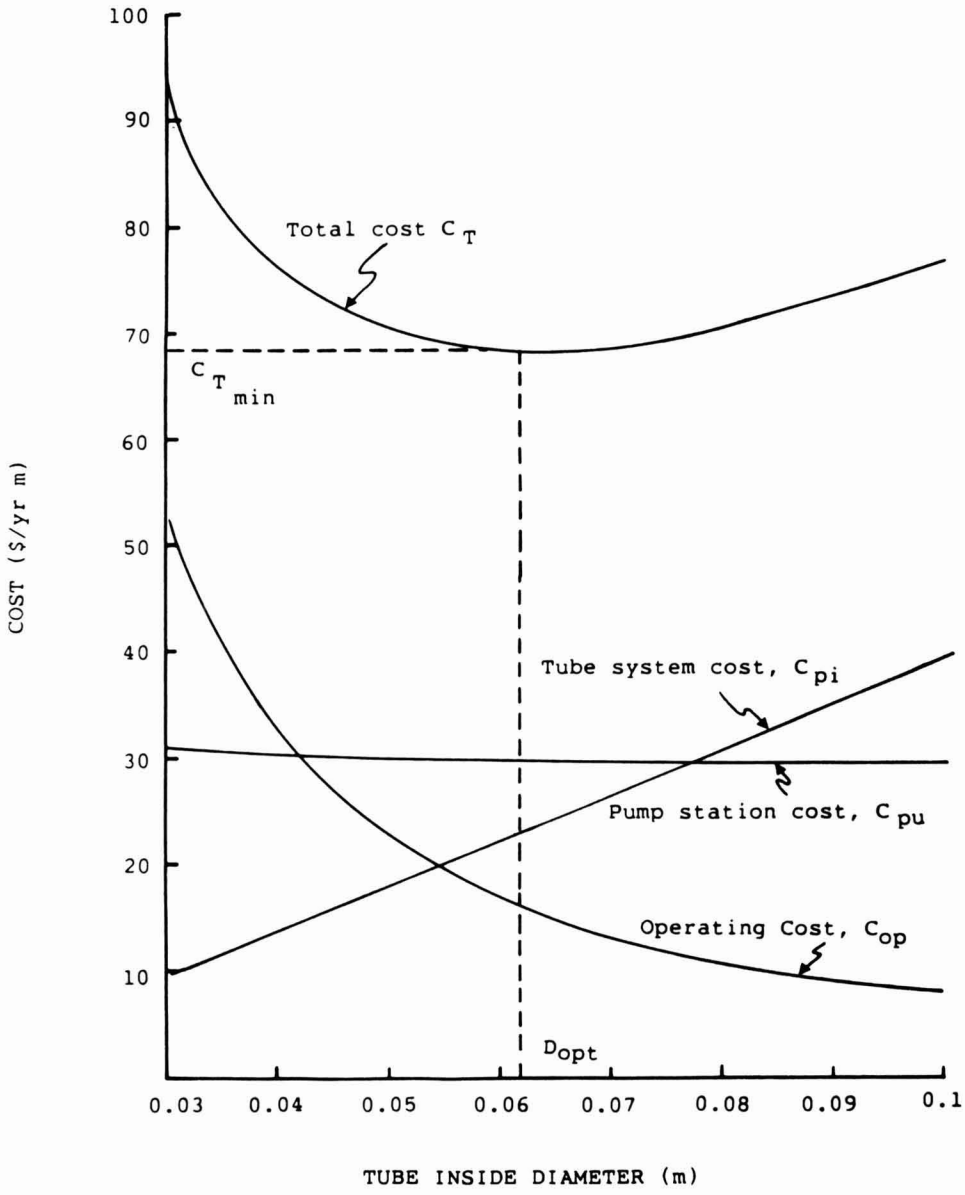

FIG. 4. VARIATION OF TUBE SYSTEM COST, PUMP STATION COST, OPERATING COST AND TOTAL COST WITH TUBE INSIDE DIAMETER.

pipe optimization; hence the next higher standard size could be selected once the D_{opt} (inside diameter) is obtained. For this example, however, a 2½ in. OD tube (0.0602 m inside diameter) would be selected. A sharper minimum may be found in other examples.

Table 4. Optimum economic tube diameter pumping system costs, theoretical work and flow conditions at optimum for the example problem.

D_{opt}	(m)	0.06181
$C_{T_{min}}$	($/yr m)	68.47
C_{pi}	($/yr m)	22.66
C_{pu}	($/yr m)	29.76
C_{op}	($/yr m)	16.05
Re		152.55
He		7.09
f		0.1512
W	(J/kg)	712.46

SUMMARY AND CONCLUSIONS

A method has been developed for determining the total annual cost of a pumping system as a function of tube diameter (based on the costs of the tube system, pump station and operation) for systems handling Herschel-Bulkley fluids under laminar flow conditions.

A method has been developed for determining the optimum economic tube diameter for pumping systems handling Herschel-Bulkley fluid under laminar flow conditions.

When the pump station cost varies little with power requirements, this cost has less influence than the operating cost in determining the optimum economic pump diameter.

The optimum economic tube diameter is independent of any elevation difference (Δz) and pressure energy difference (Δp) in the system if a linear relationship ($m = 1$) is used to correlate the pump station cost to power requirements. In addition, Δz and Δp do not have to be known accurately if the variation of the pump station cost with power is small.

The optimum economic tube diameter can be obtained from the pumping system costs of a unit length of tube if a linear relationship is used to correlate the pump station cost to the power requirements ($m = 1$) and the frictional loss due to fittings is insignificant compared to the friction loss in the pipe ($L >> D_{opt}$).

NOMENCLATURE

A = empirical constant for the pump station cost, $\$/W^m$

a = annual fixed cost of the pipe system expressed as a fraction of its initial installed cost, 1/yr

a' = annual fixed cost of the pump station expressed as a fraction of its initial installed cost, 1/yr

B = empirical constant for the pump station cost, \$

b = annual maintenance cost of the pipe system expressed as a fraction of its initial installed cost, 1/yr

b' = annual maintenance cost of the pump station expressed as a fraction of its initial installed cost, 1/yr

C = empirical constant, $\$/m^{1+s}$

C_D = total direct cost of equipment of size P_1

C_e = cost of electrical energy, \$/W h

C_I = total indirect cost of equipment of size P_1

C_{op} = total annual operating cost per meter length of pipe, \$/yr m, Eq. (5)

C_{pc} = total installed pipe system cost; including fittings, valves, installation, etc., \$/m, Eq. (1)

C_{pi} = total annual cost of installed pipe system per meter length of pipe, \$/yr m, Eq. (2)

C_{ps} = total cost of installed pump station, \$, Eq. (3)

C_{pu} = total annual cost of installed pump station per meter length of pipe, \$/yr m, Eq. (4)

C_T = total annual cost of a pumping system per meter length of pipe, \$/yr m, Eq. (6)

$C_{T_{min}}$ = minimum total cost of a pumping system per meter length of pipe, \$/yr m, value of C_T at D_{opt}

D = pipe inside diameter, m

D_{est} = best estimate of D_{opt} to start numerical search, m

D_{opt} = optimum economic pipe diameter, m

D_1 = diameter of pipe of known cost, m

E = combined fractional efficiency of pump and motor

F = energy loss due to friction, J/kg, Eq. (8)

f = Fanning friction factor, Eq. (16)

f_c = laminar-turbulent transition valve of f

g = acceleration due to gravity (9.8 m/s²)

h = hours of operation per year

He = generalized Hedstrom number Eq. (18)

K = friction loss coefficient for valve or fittings

L = pipe length, m

$\dot{M}$ = mass flow rate, kg/s

m = dimensionless exponent of pump station cost equation

n = flow behavior index, dimensionless

P = power requirements, W, Eq. (11)

P_1 = power at pump of known cost, W

q = cost capacity factor

R = ratio the total cost for fittings and installation of pipe and fittings to purchase cost of new pipe

Re = generalized Reynolds number, Eq. (14)

Re_c = laminar-turbulent transition value of Re, Eq. (19)

s = dimensionless exponent of pipe system cost equation

$\bar{V}$ = mass average velocity, m/s

W = work per unit mass, J/kg, Eq. (7) and (10)

X = purchase cost of a pipe diameter per unit meter of pipe length, \$/m

y = constant for purchase cost of pipe dependent on the pipe material, dimensionless

GREEK LETTERS

$\dot{\gamma}$ = rate of shear, s^{-1}
Δp = change in pressure, Pa
ΔP = pressure drop due to friction, Pa
Δz = change in elevation, m
ΔKE = change in kinetic energy, J/kg
η = plastic viscosity, Pa s
κ = consistency coefficient, Pa s^n
μ = Newtonian viscosity, Pa s
ξ_o = dimensionless unsheared plug radius, τ_o/τ_w
ξ_{oc} = laminar-turbulent transition value of ξ_o
ξ'_{oc} = laminar-turbulent transition value of ξ_o for D_c at a given M
π = pi (3.1415)
ρ = fluid density, kg/m^3
ϕ = parameter in the df/dD Eq. for laminar flow, Eq. (25)
ϕ_{cd} = laminar-turbulent value of ϕ for D_c at a given M
τ = shear stress, Pa
τ_o = yield stress, Pa
τ_w = value of τ at the wall, $D\Delta P_f/(4L)$
ψ = laminar flow function, Eq. (15)
ψ_c = laminar-turbulent transition value of ψ
ψ_{cd} = laminar-turbulent transition value of ψ for D_c at a given M

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