A PHENOMENOLOGICAL STUDY ON TWIN SCREW EXTRUDERS

L. P. B. M. JANSSEN

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LÉON PETER BERNARD MARIE JANSSEN

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SAMENVATTING

Hoewel het gebruik van dubbelschroefextruders in de polymeerindustrie steeds toeneemt, zijn de theoretische achtergronden relatief weinig ontwikkeld. De literatuur staat vol met tegenstrijdigheden en vaak wordt de lezer voorgehouden, dat al zijn extrusieproblemen opgelost zijn als hij een bepaald nieuw ontwerp gebruikt. Tot nog toe is de ontwikkeling van succesvolle machines vooral het resultaat van commercieel waardevolle empirie.

Bij dit onderzoek is kennis vergaard over de relevante fenomenen die de werking van tegendraaiende nauwsluitende dubbelschroefextruders bepalen. Omdat in deze machines twee schroefwormen nauw in elkaar sluiten ontstaan er min of meer C vormige kamers waar zich lekspleten tussen bevinden. De grondslag van het model, dat in dit proefschrift gepresenteerd wordt, is, dat de extruder beschouwd kan worden als een machine waarin, door de draaiing van de schroefwormen, twee series kamers verplaatst worden van de vultrechter naar de spuitopening. De lekstromen zorgen voor de interactie tussen deze kamers. De opbrengst van een dubbelschroefextruder kan, door dit model te gebruiken, berekend worden door het totaal aan lekstromen af te trekken van het totale volume aan C-vormige kamers dat per tijdseenheid aan de spuitopening vrijkomt. Dit is gecontroleerd met modelextruders die volledig gevuld waren met Newtonse vloeistoffen. De resultaten hiervan waren zeer bevredigend.

Bij de presentatie van de relatie tussen druk en debiet van volledig met Newtonse vloeistoffen gevulde extruders is gebruik gemaakt van twee nieuwe dimensieloze kentallen, $Q/2mNV$ en $\Delta P/N\eta$. Door het gebruik van deze getallen kan deze relatie voorgesteld worden door een rechte lijn, die onafhankelijk is van de viscositeit en het toerental.

Teneinde meer inzicht te krijgen in de mate waarin de verschillende geometrische variabelen van invloed zijn op de relatie tussen druk en debiet is een gevoeligheidsanalyse uitgevoerd. Hierbij is gebruik gemaakt van computerprogramma's, die gebaseerd zijn op de berekeningen van de verschillende lekken. Het bleek, dat als gebruik gemaakt werd van de hierboven genoemde dimensieloze kentallen, de druk-debietkarakteristieken niet veranderden, als de schaal van de extruder geometrisch werd vergroot.

In veel kwalitatieve artikelen wordt gesteld, dat in dubbelschroefextruders een veel betere homogenisatie op microschaal plaats vindt en dat er bovendien minder spreiding in de verblijftijd is dan in enkelschroefextruders.

De experimenten zoals deze in deze dissertatie worden beschreven wijzen echter uit, dat de dimensieloze verblijftijdsverdeling slechter kan zijn dan die van een enkelschroefextruder. Video-opnamen van tracer-injecties in een perspex modelextruder hebben uitgewezen, dat de menging binnen in de kamers eveneens vaak overschat wordt.

De laatste hoofdstukken behandelen de extrusie van echte polymeren. Het model voor ideale vloeistoffen, zoals in het eerste gedeelte van deze dissertatie is voorgesteld, is aangepast, zodat het ook gebruikt kan worden om polymeerextrusie te beschrijven.

Hiermee wordt een verklaring gegeven van de relatieve ongevoeligheid van het debiet voor de druk vóór de spuitopening, indien polymeergranules worden gebruikt. Door middel van experimenten, waarbij een echte extruder werd stop gezet en gedemonteerd, kon de voorspelling bevestigd worden dat de lengte waarover de extruder volledig met gesmolten polymeer gevuld was, varieert met de einddruk.

Ten slotte kon kwalitatief aangetoond worden dat het smeltproces in een nauwsluitende dubbelschroefextruder aanzienlijk verschilt van het smeltproces in een enkelschroefextruder.

SUMMARY

Although more and more twin screw extruders are being used in the polymer industry, the theoretical background is relatively undeveloped. The literature abounds in contradictions and often informs the reader that all extrusion problems can be solved if a certain new design is considered. The development of successful machines has mainly been possible through the application of commercially valuable empirical knowledge.

In this research some understanding has been gained of the relevant phenomena that control the working of a counter-rotating intermeshing twin screw extruder. In these machines the two screws form more or less C shaped chambers with leakage gaps between them. The basis of the model presented in this thesis is that the extruder can be regarded as two series of chambers moved from hopper to die by the rotation of screws, while the leakage flows provide interactions between these chambers.

The throughput of a twin screw extruder can be calculated with this model by subtracting the total amount of leakage from the total volume of the C shaped chambers that come free per unit time. This has been checked with model extruders fully filled with Newtonian liquids giving completely satisfactory results. For these conditions the throughput to pressure relationship, presented in terms of two new dimensionless groups, $Q/2NmV$ and $\Delta P/N\eta$, can be represented by a straight line independent of viscosity and rotational speed. In order to obtain insight into the importance of the geometrical variables for the throughput to pressure relationship a sensitivity analysis has been made using computer programs based on the calculations of the leakage gaps. When using the two dimensionless numbers the characteristics are retained when the extruder is scaled up geometrically.

Many quantitative articles report that twin screw extruders provide a better homogenisation on micro scale and that there is a narrower spread in residence time distributions than in single screw extruders. The experiments described in this thesis show that the dimensionless exit age distribution of a twin screw extruder can be worse than that of single screw extruders. Video recordings of tracer injections in a perspex model extruder have shown that the mixing within the chambers is also generally overestimated.

The last two chapters deal with the extrusion of real polymers. The model as proposed in the first part of the thesis is extrapolated to practical situations. An explanation of the relative insensitivity of the throughput to the die pressure in a twin screw extruder when working with polymer granules is given. Experiments in which a real twin screw polymer extruder was stopped and dismantled confirm the predictions of the model that the length over which the extruder is fully filled with melt changes with die pressure.

Finally it is shown quantitatively that the melting process in an intermeshing counter rotating twin screw extruder is markedly different from that in a single screw extruder.

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CHAPTER I
INTRODUCTION

I.1 HISTORICAL REVIEW.

The art of extrusion is already old. In England in 1797 Joseph Bramah constructed a hand operated piston press for the manufacture of seamless lead pipe, which machine is generally thought to have been the first extruder. It was not long after that that other materials like soap, macaroni and building materials were processed with similar machinery. The call for a screw extruder came from the cable industry. While the intermittency of a ram extruder could be accepted for the earlier applications, it was an unacceptable limitation to processes for cable covering. The earliest known concept of single screw extrusion is found in an 1873 drawing owned by Phoenix Gummiwerke A.G. The first known twin screw extruder was developed in 1869 by Follows and Bates in England for sausage manufacture. The first twin screw extruder for plastics processing was constructed just before the second world war by Roberto Colombo and Carlo Pasquetti in Italy. Since then the use of these machines has steadily increased. In particular since most of the major mechanical design problems, including the design of thrust bearings, have been solved within the last decade more and more twin screw extruders have come into use in the polymer processing industry.

I.2 SINGLE SCREW EXTRUDERS.

Over the years single screw extrusion has been thoroughly investigated, and its working is now well understood (I.1, I.2, I.3). The major difference between a single screw extruder and a twin screw extruder lies in the mechanism of transportation. A single screw extruder has a screw rotating in a closely fitting barrel. It is easy to understand that if the process material sticks to the screw and slips at the barrel surface, there will be no output from the extruder, because the material rotates with the screw without being pushed forward. In order to achieve maximal output the material has to slip as freely as possible on the screw surface and should adhere as much as possible to the wall. Under these circumstances the rotational speed of the material is less than that of the screw, so that material is forced along the extruder by the leading edge of the flight. In normal operation with a viscous fluid the velocity gradients (and associated shear stresses) are greater near the barrel than near the screw surface, and conveying takes place. In this case three flow components can be distinguished:

1. a drag flow, caused by contact effects between the material and the barrel and screw surfaces,
2. a pressure flow, due to the pressure that is built up at the die end of the extruder. The direction of this flow is opposite to that of the drag flow,
3. a leakage flow through the gap between the barrel and the flight of the screw. This flow is normally very small and is usually neglected.

I.3 TWIN SCREW EXTRUDERS.

In a twin screw extruder two parallel screws are placed in a figure-of-eight section barrel. The objective was to overcome the effects of slip at the wall. Generally speaking screws can be divided into the two major categories of intermeshing and non-intermeshing screws (see figure I.1).

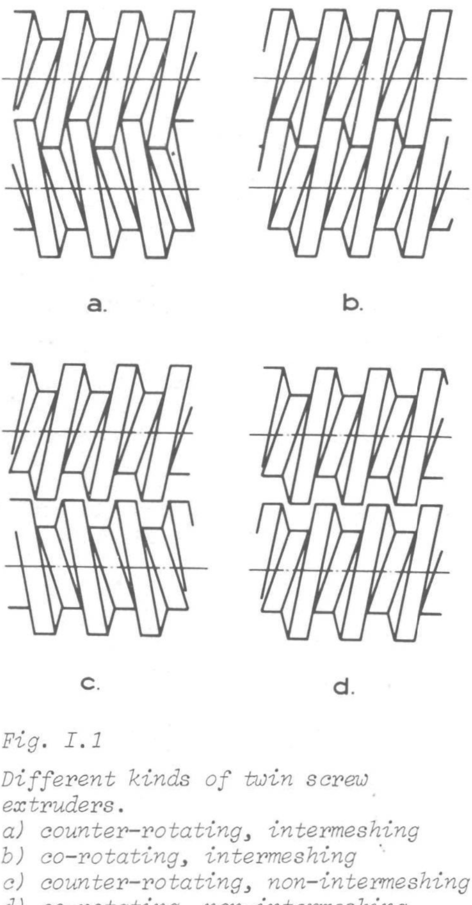

Fig. I.1

Different kinds of twin screw extruders.
a) counter-rotating, intermeshing
b) co-rotating, intermeshing
c) counter-rotating, non-intermeshing
d) co-rotating, non-intermeshing

In non-intermeshing extruders the separation between the screw axes is at least equal to the screw outer diameter. This can be regarded more or less as two single screw extruders which influence each other. In fact Kaplan and Tadmor (I.4) have developed a three parallel plate model for the working of this machine which is very similar to the two parallel plate model conventionally used for single screw extruders. When the screws are intermeshing the separation between the screw axes is somewhat less than the outer screw diameter; in the limit the screw surfaces can be in mutual contact. Intermeshing screws imply that more or less C-shaped chambers are present that positively convey the process material to the die end of the extruder. Thus slip at the wall becomes irrelevant, since the intermeshing part of one screw prevents the material in the other screw rotating freely. Schenkel (I.3) stated that the idealised theoretical output from a multi-screw extruder is the number of C-shaped chambers becoming free per unit time multiplied by the volume of one chamber.

This gives for a twin screw extruder with m thread starts per screw:

$$Q_{th} = 2\,m\,N\,V \qquad (I.1)$$

N here represents rotation rate and V chamber volume. In practice the output is less than this ideal value, because of the presence of leakage flows within the extruder.

In literature (e.g. I.5, I.6) a correction factor is used by which the theoretical output has to be multiplied to obtain the practical output, so:

$$Q_{pr} = d\, Q_{th} \qquad (I.2)$$

with $0 < d < 1$. The correction factor proposed in this way was an undefined empirical constant which was not related directly to the phenomena that occur in the extruder. Because of the need for mechanical clearances the C shaped chambers are not perfectly sealed, even with closely intermeshing screws, and leakage flows occur within the extruder. The analysis which is presented here is based on the examination of the various leakage flows which are generally present. As illustrated in figure I.2, four kinds of leakage can be distinguished:

1. The leak (Q_f), through the gap between the flight and the barrel wall. This leakage is somewhat similar to that in a single screw extruder. We call this the flight leak.
2. The leak (Q_c) between the bottom of the channel of one screw and the flight of the other screw. Because of some similarity to a calender we suggest the name calender leakage.
3. The leak (Q_t) through the gap that goes from one screw to the other between the flanks of the flights of the two screws. With closely intermeshing screws this passage-way is generally narrow and long near the screw axis and wide and short near the barrel wall. We name this the tetrahedron gap.
4. The leak (Q_s) through the gap between the flanks of the screws perpendicular to the plane through the screw axis. This gap, which confuses the already complicated flow situation considerably, is called the side gap. In behaviour this leak is rather similar to the calender leak.

The mass flow through a particular leakage gap is determined partly by drag from the locally moving surfaces and partly by the pressure difference between the relevant chambers. This pressure difference must involve not only the contribution of the build-up resulting from the throttling effect of the die on the positive displacement action of the screws, but also the locally generated pressure gradient produced by the same mechanism of principal drag pressure generation that drives a single screw extruder.

By subtracting the sum of the leakage flows from the volumetric theoretical throughput the practical throughput is obtained and the influence of several factors on the throughput can be studied.

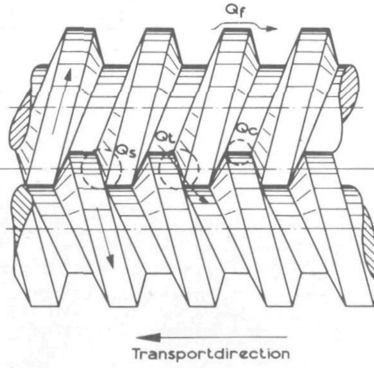

Fig. I.2

Various leakage flows in the extruder.

I.4 COMPARISON BETWEEN TWIN SCREW AND SINGLE SCREW EXTRUDERS.

Twin screw extruders have specific advantages and disadvantages. The major problem is the thrust bearing assembly. Although improved considerably during the last twenty years it is still limited in capacity compared with the thrust bearing in a single screw extruder. The reason for this is simple. In a twin screw extruder, the thrust bearing assembly has to be designed for two parallel shafts in a situation where there is a relatively small distance between the centres of the two shafts, whereas a single screw extruder only requires one bearing which can be made any desired size. This bearing problem is one of the major reasons why a twin screw extruder is much more expensive than a single screw extruder. On the other hand twin screw extruders can operate at lower speeds than single screw extruders generally do, while still having comparable output capacity and the ability to build up reasonably high die pressures. Because of these low screw speeds twin screw extruders need a far lower power input than comparable single screw extruders do. This leads to other advantages. An extruder is a thermodynamic unit. Most of the power required to drive the screws is converted into heat, but because of losses in the motor unit, the gear system and the bearings, heat produced by viscous dissipation is more expensive than heat acquired from simple thermal heaters. Heat supplied through the barrel can in some cases also be waste heat from other units. As most polymers are subject to thermal degradation, good temperature control is needed In a single screw extruder much heat is generated by viscous dissipation. This dissipation can be controlled by changing screw speed. The associated variation in output can be especially troublesome when the extruder is supplying other machines. When the viscous dissipation is relatively small and the major part of the heat is supplied externally, as it is in a twin screw extruder, the screw speed and therefore the output can remain constant while the heaters can be used for controlling the process over a wide range of conditions.

This ability to achieve good temperature control is essential when processing polyvinylchloride. Different from most polymers the apparent viscosity of p.v.c. increases with increasing temperature (I.7). When plasticised p.v.c. is processed, frictional heating increases the apparent viscosity, which in turn leads to more frictional heat generation until the material degrades and burns if the heat is not removed. On scaling up, since the heat transfer surface increases with the square of the diameter while the volume increases with the cube of the diameter, large single screw p.v.c. extruders cannot be controlled sufficiently by screw or barrel cooling. As frictional heat generation is relatively small in twin screw extruders the machines are particularly suitable for applications using p.v.c. in which addition of thermal stabilizers to the polymer has to be limited, for example because of toxicity.

I.5 MOTIVATION.

Although the industrial application of twin screw extruders and processors is steadily increasing the theoretical background is very little developed.

The most important reason for this is the complicated geometry, above all a result of the continued intersection of the screw channels by the threads of the other screw. The literature abounds in contradictions. The development of succesful machines has been a result of a combination of commercially valuable empirical knowledge coupled with the efficient application of mechanical engineering solutions to the severe constructional problems. The complexity of the problems and the limited phenomenological knowledge in this field still vitates a deep and thorough analysis. The purpose of the present work is to develop an understanding of the working mechanism and to indicate a possible mathematical treatment for twin screw extruders.

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CHAPTER II

THE STATE OF ART

II.1 INTRODUCTION

This chapter reviews papers on the theory and practice of twin screw extrusion published during the last fifteen years. From literature it is clear that twin screw extrusion technology is still more of an art than a science. Although the development of these machines started nearly fifty years ago little quantitative research into the phenomenology of the process has been published. The literature shows a clear distinction between qualitative and quantitative articles. The qualitative articles promise more than the quantitative articles can prove. Although some research has been carried out, a wide field remains unexplored.

II.2 INPUT, OUTPUT AND SCREW FILLING

The basic theory describing modern twin screw operation, to which most of the quantitative papers refer, was developed by Schenkel (II.1). According to this theory the positive conveying capacity with a completely filled cross section of a symmetrical system of s single-start screws at a rotational working speed n is the total volume that becomes free per unit time, thus:

$$Q_{th} = s\ N\ V \qquad \text{(II.1)}$$

where Q_{th} is the theoretical throughput and V is the volume of a single C shaped chamber. Schenkel states that for closely intermeshing screws of small radial clearance the real output will be between 90% and 95% of this depending on die pressure and the viscosity of the melt. In the same chapter of his book Schenkel also presents a basis for the calculation of the power consumption in twin screw extruders and a survey of commercial multiscrew extruders.

Among the quantitative articles, those of Klenk (II.2, II.3, II.4) give a good survey. These articles form part of a series about p.v.c. extrusion with both single and twin screw extruders. This excellent survey of the state of knowledge in 1970 was based on his thesis (II.5). The first paper referred to (II.2) gives both a general survey and a description of the extruders used. The second paper (II.3) is concerned with the throughput, and presents a modification of a melting model Dobóczky published in 1965 (II.6). Klenk reports that the throughput is proportional to the screw rotation speed. The efficiency of the extruder is defined as the ratio between the real throughput and the theoretical throughput and is only slightly dependent on the rotational rate. For several extruders with normal filling it lies between 34% and 41% i.e. less than half that the value given by Schenkel. With forced feeding this efficiency can be much higher. Twin screw extruders suffer from a disadvantage in that the output pressure fluctuates as successive chambers are discharged. Klenk states that this is due to the varying pressure within the chambers and suggests that this can be minimised by making the screws such that:

$$\frac{\partial p}{\partial z} = \frac{\partial p}{\partial x} \tan \Phi \qquad \text{(II.2)}$$

with x, z and ϕ as indicated in figure II.1. In the same article (II.3), it is stated that Konstantinov and Levin have measured these pressure fluctuations (II.7). They found big fluctuations in a starved extruder, which disappeared as the extruder was filled completely by forced feeding. When the extruder was positively filled with an excess of feed some fluctuations reappeared but were smaller than those when operating in the starved condition.

An interesting paper on the throughput of twin screw extruders was published by Dobóczky (II.8) as long ago as 1965. He collected and compared data on several twin screw extruders. After correcting the theoretical throughput for changing screw geometry and bulk density it was concluded that the ratio of practical to theoretical throughput is about 70%, though with a spread in measurements about 20% each side of this value. He also reported that the throughput from a twin screw extruder is about three times that from a single screw extruder of similar size and rotational speed.

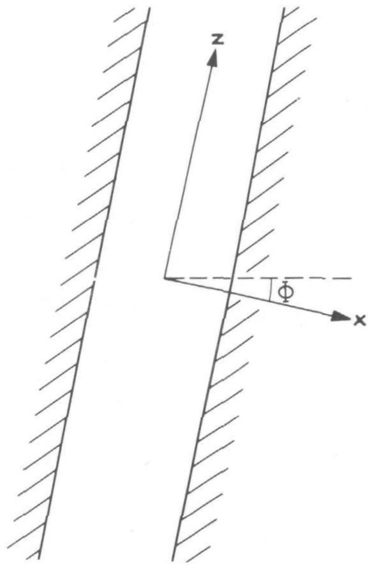

Fig. II.1

Unwound channel with coordinates.

The theory as described in chapter 6 of this thesis agrees fully with the experiments reported by Marhenkel in 1965 (II.9). He measured the axial pressure profile in a twin screw extruder with five transducers spaced along the barrel. In general pressure is generated at the die end of the extruder. It is only when non-uniform screws are used with one or more of the zones tending to be overfilled that a pressure generation upstream of that zone can be registered. There are also some parallels with the predictions of pressure losses in kneading sections of twin screw extruders by Armstoff and Zettler (II.10). In their article the pressure build up before a kneading zone in a co-rotating twin screw extruder is estimated from a modified single screw theory. The analysis and experiments concern the length of the extruder needed to build up a certain pressure within the melt, called a compression length. This should not be confused with the compression zone as normally understood in single-screw technology in which the channel depth is reduced (i.e. a geometrically defined region); in the present double screw case we are concerned with a uniform screw section in which a compression length is established purely from hydrodynamic considerations.

However, it should be remarked that in Armstoff and Zettler's analysis the geometry of the leakage gaps that form the basis of the analysis presented here was not taken into account. In a plot of measured compression lengths of the polymer before the kneading elements against those calculated, the spread in the measurements is fairly wide. To be honest the points appear to have a correlation coefficient of less than 0.75.

An interesting technique is the use of a single screw to feed the twin screw extruder with solids. Schiffers (II.11) reports on such an extruder where he varies the geometry and the rotational speed of the feed screw and the extruder. The material used was p.v.c. From his results it is clear that there are three working regions, one when the extruder is starving, one when compression of polymer takes place and one when the extruder tends to be overfilled. He also presents measurements of pressure build up. It is to be regretted that the size of the published figures concerning this part of the investigation is such that hardly any quantitative conclusion can be drawn.

II.3 FLOW, MIXING AND RESIDENCE TIME DISTRIBUTION

II.3.1 Visualisation techniques

Twin screw extruders are often claimed to have good flow, mixing and residence time distribution characteristics, but there have been few published investigations. These can be divided between theoretical calculations and practical measurements, the latter always concerning model experiments.

Jewmenow and Kim (II.12) established the flow within a single chamber by injecting aluminium particles in a model fluid of a polyisobutene solution in oil. Through an observation window in the extruder they took photographs of the streamlines.

Todd (II.13) has measured residence time distributions with polybutenes using methylene blue as a tracer. Since methylene blue is completely insoluble in polybutenes, it can be leached out with water, allowing an estimate of the integral of the "tail" of the residence time curve. He has, together with Irving (II.14) also presented experiments done with glucose solutions with KNO_3 and $NaNO_3$ added. The residence times were determined in that case from conductivity measurements.

II.3.2 Experimental work

In a single screw extruder the helical flow field is fully developed and is essentially the same at all sections along a uniform screw filled with melt. In contrast to this, the essentially closed nature of the chambers in the double screw geometry leads to a flow field of a fully three-dimensional character as is clearly established by experiments (II.12). Because of the great influence of the flight walls a well defined zero velocity layer exists in the chamber dividing fluid moving in opposite directions (see fig. II.2).

Jewmenow and Kim (II.12) found experimentally that the flow profile far away from the intermeshing zone is not affected by the absolute screw velocity, so that one can use a relative velocity to define the flow field.

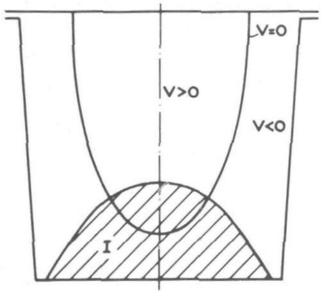

Fig. II.2

Directions of fluid velocity in the chamber.

Furthermore, this velocity profile is affected neither by the viscosity nor by variations of the die pressure over a wide range. Naturally the geometry of the gaps has an important influence on the profile since it affects the net throughput through the chamber and therefore the position of the zero velocity layer.

Todd and Irving (II.14) compared the twin screw geometry with a mixing device called the Poly-Con. This can best be described as a series of lengths of co-rotating paddles of elliptical section, effectively operating as a twin screw extruder with little positive displacement forward transport. They characterised the mixing process with a Peclet number and found that the axial mixing was greater for the continuous twin screw arrangement ($Pe < 6$). They also report that the Peclet number was independent of feed rate at a constant rotational speed. Increasing screw speed from 100 RPM to 200 RPM decreased the Peclet number from 6 to 3. The paper Todd published in 1975 (II.13) shows cumulative residence time distribution curves on log-probability paper. The major conclusions from this work are that residence time distribution data are particularly useful in diagnosing axial mixing phenomena, providing a basis for scale up and leading to improved equipment design. If very little axial mixing is required, Todd advises the use of screws with very low helix angles or alternatively straight segments which provide no axial forwarding. The same data have also been presented in a modified form as a Chemical Engineering Progress Capsule (II.15).

II.3.3 Theoretical work

The article published by Kim, Skatschkow and Jewmenow (II.16) gives a very thorough mathematical analysis of the mixing in a single C shaped chamber of a twin screw extruder. The basis of these calculations is the velocity profiles as calculated by Konstantinov in his thesis (II.17) by solving the equations:

$$\frac{\partial p}{\partial x} = \eta\left(\frac{\partial^2 v_x}{\partial x^2} + \frac{\partial^2 v_x}{\partial y^2}\right) \qquad (II.3)$$

$$\frac{\partial p}{\partial y} = \eta\left(\frac{\partial^2 v_y}{\partial x^2} + \frac{\partial^2 v_y}{\partial y^2}\right) \qquad (II.4)$$

$$\frac{\partial p}{\partial z} = \eta\left(\frac{\partial^2 v_z}{\partial x^2} + \frac{\partial^2 v_z}{\partial y^2}\right) \qquad (II.5)$$

Although the thesis itself is unfortunately not generally available, from the complex results quoted, it can be concluded that the flow is essentially three dimensional. Together with the velocity profile in the calender gap Kim et al. calculated the average integral shear rates. From this shear the reduction in striation thickness in the extruder can be estimated.

A further theoretical paper by Wyman which deals with velocity profiles restricts itself to shallow channels (II.18). In this case the flow was considered to essentially two dimensional and could be solved analytically. It is however the fact that normal twin screw extruders never have shallow channels and therefore the flow must be considered to be three dimensional. The author refers to the two Russian papers dealt with above (II.12), (II.16) and concludes that they are in general agreement with his results, though this claim appears to be too simplistic.

II.4 TEMPERATURE DISTRIBUTION

Klenk (II.19) describes the temperature distribution of the polymer in the die head of a twin screw extruder beyond the breaker plate. He used three thermocouples, one in the middle and one asymmetrically on each side (fig. II.3). When assuming a symmetrical temperature profile in horizontal direction this gives a five point profile. Temperature profiles like those shown schematically in figure II.4 were found. Curve A indicates the temperature profiles when working with a high rotational speed, curve B when working at low rotational speeds. Klenk explains these results in terms of developing temperature profiles. When the speed of the polymer is low a steady state temperature profile can build up, corresponding to that expected for flow through a pipe. The temperature in the middle is highest because of viscous dissipation and the poor thermal conductivity of the polymer. When the speed of the polymer is high the influence of the two screws is still discernible at the point of measurement.

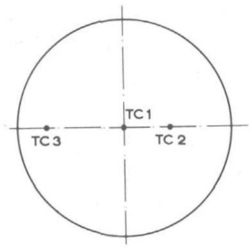

Fig. II.3

Arrangement of the thermocouples according to Klenk.

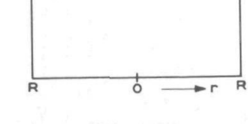

Fig. II.4

Temperature profiles according to Klenk.
A) high speed
B) low speed.

In a later article (II.4) Klenk reports results with a symmetrical configuration of three thermo-couples, so that only three points of the temperature profile were determined. He considered the influence of barrel temperature, die resistance, screw geometry and composition of the polymer. Although these measurements give a good general impression it would have been useful to have more detailed temperature profiles since measurements in single screw extruders (II.20), (II.21) have shown, that even for that relatively simple geometry more than three measurement points are required for a good description of the temperature field.

II.5 SOME OTHER QUANTITATIVE ARTICLES

a) Engineering aspects.

An analysis of the engineering and constructive properties of a twin screw extruder is given by Stasiek (II.22). He compares technical data from some commercial single screw and twin screw extruders. The installed motor power of single screw and twin screw extruders is plotted graphically against screw diameter. The ratio of motor power for single screw extruders to that for twin screw extruders decreases from about 3 to 1.7 as the screw diameter increases from 60 mm. to 160 mm., but single screws are of course longer than double screws of the same diameter. Another graph depicts installed barrel heating as a function of screw diameter. The spread in points is very large; almost the only clear conclusion is that for screw diameters over 120 mm. the installed barrel-heating in single screw extruders is slightly greater than that used in twin screw extruders. The ratio of investment cost to output for twin screw extruders is about 30% higher than that of single screw machines, although the ratio between extruder weight and output is the same for both machines.

b) Additives and compounding

Gale (II.23) discusses the effects of additives on twin screw extrusion. p.v.c.suspension polymer with a constant amount of damped tribasic lead sulphate was used on the tests. Two more additives, glycerol monostearate (GMS) and stearic acid (SA) were used. The total concentration of these two additives together was constant. At screw speeds varying from 9 rpm to 27 rpm the amount of molten polymer in the chambers was increasing with increasing GMS. Furthermore the effect of these additives on output rate, motor current, back pressure and impact strength of rods extruded at various screw speeds were investigated. Also the effect of MBS impact modifier on the melting rate was investigated at several screw speeds. In the second part of the paper (II.24) the effects of varying melting rate on the impact properties of a typical commercial product and how these properties are effected by the different melting mechanisms of the twin screw and single screw extruders are discussed.

Another article on additives was published by Heilmayr (II.25). He compared 26 commercial kinds of p.v.c. in their processing and product performance. Also the influence on transparancy, thermal stability and light stability of some components of the stabilisers were investigated.

To achieve good conditions for process control while compounding heat and shear sensitive polymers Adams (II.26) suggested the use of a fast running twin screw extruder together with a slowly running single screw extruder as a melt pump in series. He concluded that this configuration can produce crosslinkable compounds successfully at high volume rates. He stated that this two stage system provides:

a) short and uniform retention time,
b) control over stock temperature substantially below the activation point of the material in order to assure the full effect of cross-linking agents and
c) uniform and complete dispersion quality.

II.6 QUALITATIVE ARTICLES

During the last decade many qualitative articles have appeared. Some of them describe the process and inform the reader that twin screw extruders are wonderful machines that can solve all extrusion problems, usually if a certain new design is considered. We tend to agree with Prause who states (II.27) "If there is one key point we hope to hammer home, it is simply this: If your single screw extruder is doing an efficient, high quality job for you, then continue to use it. Twin screw extruders, while they have a definite important place in the U.S. market (particularly in the compounding and direct extrusion of rigid p.v.c.), are not necessarily everyone's panacea".

Some articles give information about new applications for twin screw extruders. Gras (II.28) discusses the addition of rovings or chopped glass through an open degassing adaptor. The continuous glass strands that form the rovings are chopped and homogenized within the extruder. The main advantage of rovings over chopped glass is that rovings do not require any special metering unit while chopped glass does. Mack (II.29) discusses the use of a twin screw extruder as a reactor and reports on successful applications for bulk polymerization. For instance, because of the good internal mixing, post condensation time can be reduced by a factor 6 compared with what he describes as heavy-layer-type equipment. Another advantage claimed is the accurate temperature control that is possible in this type of machine. Gras (II.30) discusses new applications for twin screw extruders. He considers the advantages of these machines when dealing with devolatilization and concentration of low viscosity polymer systems, flash operation polymerisation of two component monomer systems and the compounding of incompatible materials. The use of a twin screw extruder instead of a tandem system of two single screw extruders for extruding polystyrene foam is discussed by Collins and Kraus (II.31). The special features needed for foam extrusion: plasticising the polymer dispersing and mixing the blowing agent with the melted polymer and cooling the admixture, can be performed in one twin screw extruder of 150 mm. (5.9 inches) instead of a tandem of 110 mm. and 150 mm. single screw extruders. The special advantages of using a twin screw machine are that the floor space can be reduced to half, the drive has about 1/3 of the total horse-power requirements of a single screw tandem system and there is no need to balance the screw speeds since only one machine is involved.

When compiling a bibliography of literature of this type it is necessary to beware of possible confusion between articles. There are at least three papers from different authors with the following very similar titles: Single-screw versus twin-screw extruders" (II.32), "Twin screw extruders versus single screw extruders" (II.33) and "Twin-screw versus single screw extrusion (II.34). Two further authors report on "Twin screw extruders, when to use and why" (II.27) and "Twin screw extruders, where to use and why" (II.35).

In the field of more or less qualitative articles there are some useful papers that give a good introduction to the geometry e.g. (II.36).

Articles have been published which compare screw extruders of various geometries from several manufacturers (II.37; II.38; II.39; II.40), but we know of no recent papers of this type. Schultz (II.37) focusses on the mechanical point of view especially and his article is principally concerned with the bearings: this is natural since the article was published in 1962 at a time when bearing and gearbox problems were very severe and limiting the development of twin screw extruders. A discussion of bearing and gearbox design can also be found in 1968 (II.38) when Prause reported that the problems were more or less solved. In that article he divides the screws into four basic categories:

- "Low work" screws with little or no geometric compression
- "Standard" screws with a geometric compression ratio between 1.5 and 2,
- "High shear" screws with a geometric compression ratio in excess of 2 and
- "Vented" screws, which are multistage screws.

The problem of bearing design was also treated in another article in 1963 (II.39). Here the use of screws with multiple thread starts is suggested in order to avoid large pressure fluctuations at the die, with the objective of achieving a longer life from the bearings. Müller (II.40) also compared several screw systems, suggesting the use of two twin screw extruders in series, one for the melting process and one to build up pressure. The screws of the first have to be deep cut with a small pitch, and operated with high rotational speed, while the screws of the second extruder have to be shallow with a coarse pitch and operating with a low rotational speed.

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CHAPTER III

BASIC CALCULATIONS

III.1 SCREW GEOMETRY

Before the leakages in a twin screw extruder are considered, it seems sensible to discuss the requirements set for the screw geometry in order to make the screws fit into each other. These requirements differ for counter-rotating and co-rotating screws. In the case of counter-rotating screws one screw has a left hand thread and the other a right hand thread. The rotation of the screws is such that they diverge at the top side and converge at the bottom of the intersecting area. Looking in a direction from hopper to die the right screw has a left hand thread and a clockwise rotation while the left screw has a right hand thread and rotates anti-clockwise. This choice of rotation gives the extruder good filling properties because the material coming from the hopper is distributed through the whole chamber by the rotation of the screws.

In the case of a co-rotating extruder the screws rotate either clockwise both with left hand thread or anti-clockwise both with right hand thread.

The basis of the calculations of the fitting of the screws is to determine the distance in the axial direction between some point on the flank of one screw in the intermeshing zone to the adjacent point on the flank of the other screw (for instance see figure III.1). This clearance must alway be positive or the screws will not fit into each other. The axial position of a point on the screw flank is determined by two factors, namely the pitch of the screw and the flight wall angle.

The component of the distance determined by the pitch of the screws can be expressed in the radial and tangential coordinates of the point concerned with origins at the screw axes of the two screws (see fig. III.2) and is:

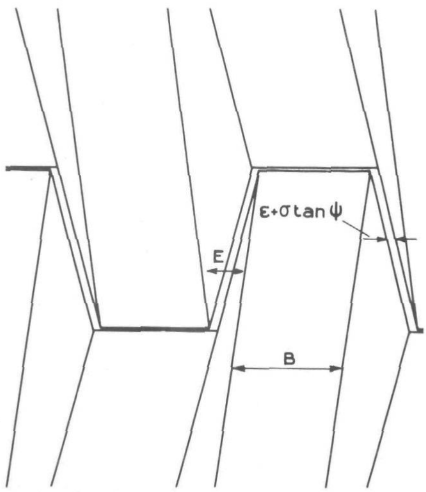

Fig. III.1

Axial distance between the flights in the intermeshing region.

$$E_1 = \frac{1}{2\pi m}(\beta_r S_r + \beta_l S_l) \qquad \text{(III.1)}$$

When dealing with a screw of trapezoidal form, that is a screw with a flat root and uniformly angled flank flights (fig. III.3), the component of the distance determined by the flight wall angle can be expressed as:

$$E_2 = (r_r + r_1 - 2R + H) \tan \Psi \qquad \text{(III.2)}$$

The axial distance can moreover be influenced by the side gap and the calender gap. This component becomes:

$$E_3 = \varepsilon + \sigma \tan \Psi \qquad \text{(III.3)}$$

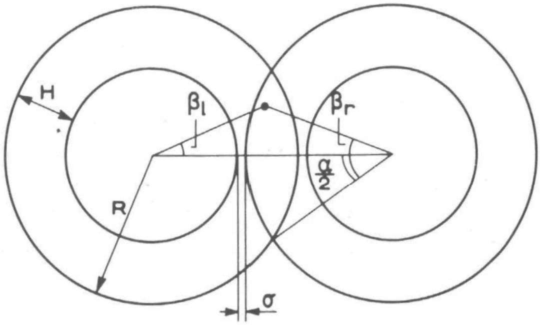

Fig. III.2

Screw geometry and coordinates

In these formulae the subscripts r and l concern the right screw and the left screw respectively. β is the coordinate angle, r is the distance of the point observed to the screw axis, S is the pitch, R is the outher radius, H is the chamber height, Ψ is the flight wall angle, σ is the calender gap and ε is the side clearance (figures III.2 and III.3).

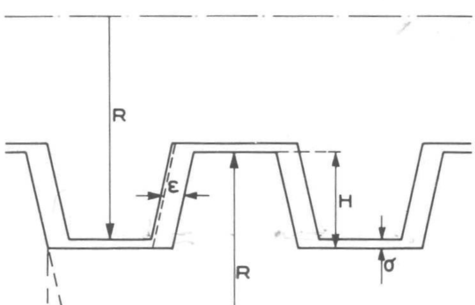

Fig. III.3

Screw geometry.

For counter-rotating screws the axial distance between the flanks in the intermeshing zone is, since:

$$S_1 = -S_r \quad \text{(III.4)}$$

$$E = \frac{S}{2\pi m}(\beta_1 - \beta_r) + (r_r + r_1 - 2R + H + \sigma)\tan\Psi + \varepsilon \quad \text{(III.5)}$$

If the flight angle Ψ and the side gap ε become too small the clearance distance E will no longer be positive. This danger appears to be greatest at the outer radius of the screw. When working with 7 cm. diameter screws with a chamber height of 1 cm. and a pitch of 2.5 cm. a side gap at the root of at least 0.2 mm. is necessary for a flight flank angle of 2^0 and 0.1 mm. for a 7^0 flank. However these gaps are so small that they can be ignored in practical cases where a side gap is always present.

In case of co-rotatingscrews the situation is quite different. Using formula III.1 with:

$$S_r = S_1 \quad \text{(III.6)}$$

the distance E becomes

$$E = \frac{S}{2\pi m}(\beta_r + \beta_1) + (r_r + r_1 - 2R + H + \sigma)\tan\Psi + \varepsilon \quad \text{(III.7)}$$

The position of the point observed can be chosen so that both β angles are negative and therefore the whole term concerning the pitch is negative. This negative value is biggest when the angles are greatest and the most critical point appears to be at the top (or bottom) of the intermeshing zone where:

$$r_r = r_1 = R \quad \text{(III.8)}$$

and

$$\beta_r = \beta_1 = \frac{\alpha}{2} \quad \text{(III.9)}$$

Here α is the angle of the overlapping area (fig. III.2)

$$\alpha = 2\ \text{atan}\left[\frac{\sqrt{(RH - H^2/4)}}{(R - H/2)}\right] \quad \text{(III.10)}$$

Inserting III.8 and III.9 in III.7 and setting E to zero the minimum values of the side gap for counterrotating screws can be expressed as

$$\varepsilon_{min} = -(H + \sigma)\tan\Psi + \frac{S}{2\pi m} \quad \text{(III.11)}$$

It can be concluded that the sidegap has to be much bigger in co-rotating screws than in counter-rotating screws. This can also be seen qualitatively in figures III.4 and III.5. Here cross sections in the middle between and parallel to the screw axes and perpendicular to a plane through the screw axes are drawn for counter-rotating and co-rotating screws respectively.

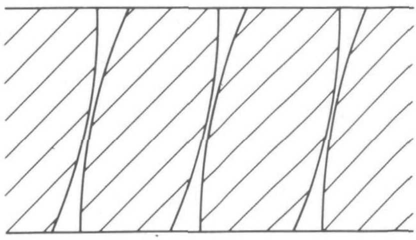

Fig. III.4

Cross section in the intermeshing zone (counter-rotating screws)

III.2 THE CHAMBER VOLUME

The most simple expression for the throughput is:

$$Q_{th} = 2\, m\, N\, V \qquad \text{(III.12)}$$

As this formula is the basis for more sophisticated models, the chamber volume V is an important parameter. This chamber volume can be found by subtracting the volume of a certain length of screw from the volume of the same length of the empty barrel. Carrying out these calculations for one pitch length and one screw only, this will give the volume of m chambers where m is the number of thread starts.

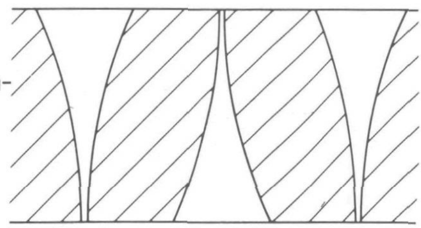

Fig. III.5

Cross section in the intermeshing zone (co-rotating screws).

The volume of one barrel half over one pitch length can be found by elementary calculation:

$$V_1 = \left\{ (\pi - \frac{\alpha}{2}) R^2 + (R - \frac{H}{2}) \sqrt{(RH - \frac{H^2}{4})} \right\} S \qquad \text{(III.13)}$$

where α is the overlap angle in radians as given by formula III.10, R is the outer radius of the screw (and taken equal to the inner radius of the barrel), H is the depth of a C shaped chamber and S is the pitch of the screw.

The volume of the screw root over one pitch length is:

$$V_2 = \pi\, (R - H)^2 S \qquad \text{(III.14)}$$

The volume of one screw flight is:

$$V_3 = \int_{R-H}^{R} b(r)\, 2\pi r\, dr \qquad \text{(III.15)}$$

where b(r) is the width of the flight in axial direction at a radius r. For straight sided screws b(r) can be approximated by

$$b(r) = B + 2(R - r)\tan\Psi \qquad \text{(III.16)}$$

B is the width of the flight in axial direction at the outer radius and Ψ is the flight wall angle. Combination of formulae III.15 and III.16 and integration gives:

$$V_3 = 2\pi\{(RH - \frac{H^2}{2})\,B + (RH^2 - \frac{2}{3}H^3)\tan(\Psi)\} \qquad \text{(III.17)}$$

Defining b(r) as the flight width in the axial direction, (and not as the flight width perpendicular to the flight), the effect of the pitch can be neglected. The error induced doing so is very small and caused by the fact that flanks which are straight in a direction perpendicular to the flight have a little curvature in an axial direction. The error caused by this effect is in general much less than one percent of the chamber volume and therefore neglected. The total volume of one chamber

$$V = \frac{V_1 - V_2 - mV_3}{m} \qquad \text{(III.18)}$$

was checked by weighing the water needed to fill an extruder model. The calculated values were within the spread of the measured values which was less than two percent so the calculations can be regarded as good.

III.3 AXIAL PRESSURE GRADIENT IN THE EXTRUDER

Pressure changes over the screw of an extruder originate from two sources. One source is the pressure which is developed at the die. The other source is the moving wall of the extruder and the flow which occurs within the chamber itself. The contribution of the pressure changes due to the latter source can be estimated from an approximate calculation. In the conventional way, as is usual in single screw extruders, the coordinate system can be reduced to a cartesian geometry in terms of x and y. The x coordinate is taken as the axial direction, the y coordinate perpendicular to x in a radial direction (see figure III.6). In the middle of the chamber it follows from a force balance

$$\frac{\partial p}{\partial x} = -\frac{\partial \tau}{\partial y} \qquad \text{(III.19)}$$

in which p is the pressure and τ is the shear stress. For Newtonian fluids where

$$\tau = -\eta\frac{\partial v}{\partial y} \qquad \text{(III.20)}$$

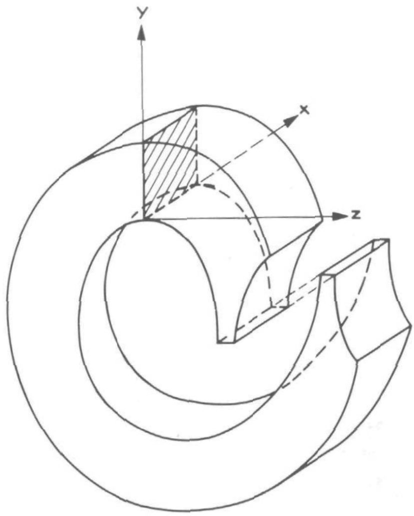

Fig. III.6

C-shaped chamber with coordinates

the equation becomes

$$\frac{\partial p}{\partial x} = \eta \frac{\partial^2 v}{\partial y^2} \qquad \text{(III.21)}$$

The boundary conditions are:

$$\begin{aligned} y &= 0 : v = 0 \\ y &= H : v = w \end{aligned} \qquad \text{(III.22)}$$

v is the velocity and w is the wall velocity given by

$$w = N\,S \qquad \text{(III.23)}$$

N is the number of screw revolutions per unit time and S is the pitch. Assuming the pressure gradient not to be a function of y, the solution of equation III.21 is

$$v_x = \frac{1}{2\eta} \frac{dp}{dx} (y^2 - Hy) + w \frac{y}{H} \qquad \text{(III.24)}$$

For continuity reasons the total flow through a plane at constant x must be equal to the leakage through the flight gap, so:

$$Q_f = \int_0^H v_x \, dy = \frac{wH}{2} - \frac{H^3}{12\eta} \frac{dp}{dx} \qquad \text{(III.25)}$$

In practical cases:

$$Q_f << \frac{wH}{2} \qquad \text{(III.26)}$$

so that equation III.25 can be rewritten as:

$$\frac{dp}{dx} = 6\eta \frac{w}{H^2} \qquad \text{(III.27)}$$

In literature several computer programs have been described to calculate the two dimensional stream lines in square ducts with moving boundaries (e.g. III.1, III.2). From these programs it can be concluded that near the barrel wall the streamlines are approximately straight and parallel. This is also true for deeper cut screws. This justifies the approximation that near to the barrel wall the pressure gradient in the axial direction is constant. Therefore the pressure change due to the moving cylinder wall can be written as:

$$\Delta p = 6\eta \frac{w}{H^2} \left(\frac{S}{m} - B\right) \qquad \text{(III.28)}$$

Using screws with constant profile the total leakage remains constant over the entire completely filled length of the extruder. Therefore when working with screws with constant profile and using isoviscous fluids the pressure gradient due to pressure build up at the discharge openeing is presumed to be constant over the filled length of the screw. Superposition of the drag pressure in the chambers on the linear pressure gradient in the extruder gives a pressure profile as indicated in figure III.7.

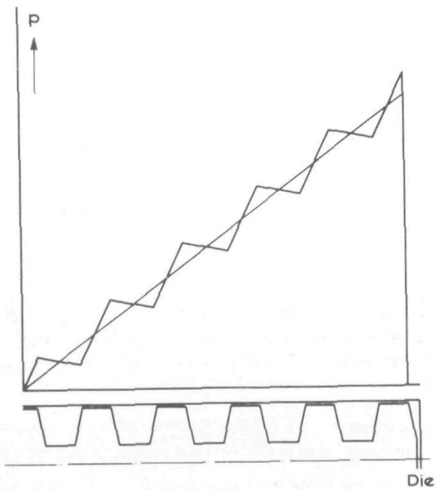

Fig. III.7

Axial pressure profile in the extruder.

III.4 TANGENTIAL PRESSURE GRADIENT IN THE EXTRUDER

A pressure is also built up in tangential direction within the chambers because of drag forces. To predict the leakage flows it is important to know pressure gradient. With counter-rotating screws, at the converging side of the screws there tends to be a build up of pressure while on the diverging side a low pressure region occurs due to the tangential pressure gradient (fig. III.8).

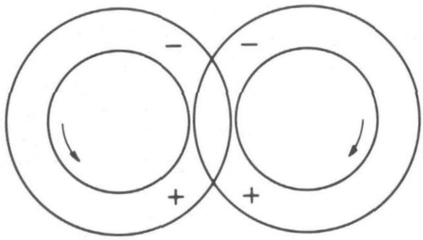

Fig. III.8

Pressure build up in a counter-rotating extruder.

This pressure difference in the screw chambers mainly influences the leaks through the calender gap and through the side gap. Coming to this point it can be seen that with a low die pressure the material in the chambers of one screw of a counter-rotating extruder mixes very poorly with the material in the chambers of the other screw.
On the other hand with co-rotating screws it can be concluded that the tangential pressure build up influences the tetrahedron gap most (fig. III.9).
In this case the process material moves into the adjacent channel of the other screw. The transfer of material from one screw to the other which is due to both the extra tangential pressure build up and the drag flow between the moving boundaries in the tetrahedron gap, creates a movement around both screws in a figure 8 pattern so a good mixing between the material in both screws is achieved.

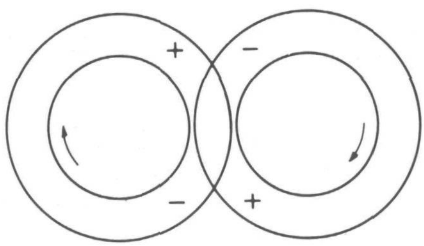

Fig. III.9

Pressure build up in a co-rotating extruder.

To calculate the tangential pressure some assumptions have to be made. The flow in the channel is assumed to be steady and isothermal, the fluid is Newtonian and incompressible and the pitch of the channel can be neglected. In case of creeping flow when inertia forces are negligibly small, as is common in extruders, the Navier-Stokes equations for a region far from the intermeshing zone reduce to:

$$\eta\left\{ \frac{\partial}{\partial r}\left[\frac{1}{r} \frac{\partial}{\partial r} (r\, v_\gamma)\right] + \frac{\partial^2 v_\gamma}{\partial x^2} \right\} = \frac{1}{r} \frac{\partial p}{\partial \gamma} \qquad \text{(III.29)}$$

Assuming the screws stationary while the barrel is moving, the boundary conditions are for a straight sided screw

$$r = R - H \qquad 0 < x < \frac{S}{m} - B \qquad v_\gamma = 0 \qquad \text{(III.30)}$$

$$R - H < r < R \qquad x = \frac{S}{m} - B \;;\; x = 0 \qquad v_\gamma = 0 \qquad \text{(III.31)}$$

$$r = R \qquad 0 < x < \frac{S}{m} - B \qquad v_\gamma = 2\pi N R \qquad \text{(III.32)}$$

Since only one velocity component is involved there is no need to use the continuity equation. Rieger and Sesták (III.3) proposed a solution method based on Fourier transform in the z direction after which the partial differential equation reduces to an ordinary inhomogenious second order differential equation. After solving this equation and applying the inverse transformation formula, the throughput can be obtained by integrating the velocity over a cross section of the channel.

$$Q = \frac{4w}{\pi^2}\left(\frac{S}{m} - B\right)^2 \sum_{n=1,3,5\ldots}^{\infty} \left\{\frac{1}{n^2}\left(A(n)\left[I_0\left(\frac{Rm}{S-Bm}\pi n\right) - I_0\left(\frac{(R-H)m}{S-Bm}\pi n\right)\right]\right.\right.$$

$$- B(n)\left[K_0\left(\frac{Rm}{S-Bm}\pi n\right) - K_0\left(\frac{(R-H)m}{S-Bm}n\pi\right)\right] +$$

$$\left.\left.\frac{2(S-Bm)}{n^2\pi^2 m\eta w}\left(-\frac{dp}{d\gamma}\right)\ln\left(\frac{R}{R-H}\right)\right)\right\} \qquad \text{(III.33)}$$

$$A(n) = \frac{C(n)}{F(n)}\left[\left(\frac{R-H}{R} - \frac{2}{n\pi C(n)}\right)K_1\left(\frac{(R-H)m}{S-Bm}\pi n\right) - K_1\left(\frac{Rm}{S-Bm}\pi n\right)\right]$$

(III.34)

$$B(n) = \frac{C(n)}{F(n)}\left[I_1\left(\frac{Rm}{S-Bm}\pi n\right) - \left(\frac{R-H}{R} - \frac{2}{\pi n C(n)}\right)I_1\left(\frac{(R-H)m}{S-Bm}\pi n\right)\right]$$

(III.35)

$$C(n) = \frac{(S-Bm)^2}{wm^2(R-H)n^3\pi^3}\left(-\frac{dp}{d\gamma}\right) \qquad \text{(III.36)}$$

$$F(n) = I_1\left(\frac{(R-H)m}{S-Bm}\pi n\right)K_1\left(\frac{Rm}{S-Bm}\pi n\right) - I_1\left(\frac{Rm}{S-Bm}\pi n\right)K_1\left(\frac{(R-H)m}{S-Bm}\pi n\right)$$

(III.37)

Here I_0 and I_1 are modified Bessel functions of the first kind and zero and first order and K_0 and K_1 are modified Bessel functions of the second kind and the zero and first order (III.4). The function obtained gives an unique relation between throughput and pressure gradient for the case of a stationary screw and a rotating barrel.

To apply this formula to a chamber in a twin screw extruder the throughput has to be related to a stationary barrel and a rotating screw, since the intermeshing zone is stationary relative to the barrel and not to the screw. Therefore a correction factor is required, which has to be subtracted from the throughput. This factor is the channel volume displaced in one complete rotation times the rotational speed, so:

$$J = 2\pi N \int_{R-H}^{H} \frac{S}{m} - B + 2(r-R) \tan \Psi \; r \, dr$$

$$= 2\pi N \left\{ \left(\frac{S}{m} - B\right) \left(RH - \frac{H^2}{2}\right) + \left(\frac{2}{3} H^3 - H^2 R\right) \tan \Psi \right\} \qquad \text{(III.38)}$$

To examine the influence of the flight wall angle ($\Psi \neq 0$) a computer program is made to calculate the velocities in the z direction and from this the throughput-pressure characteristic from a channel. In the general case where $\Psi < 12$ degrees this influence is so small that it can be neglected.

III.5 THE TETRAHEDRON GAP

Since in most twin screw extruders the flight walls are not perpendicular to the channel bottom of the screws, but have an angle Ψ, a gap exists between these flight walls. This gap is particularly important for the mixing of material in the extruder between one screw and the other, because it is the only gap through which the material is transported directly from one screw to another, as can be seen in figure I.2. In close fitting screws the gap is approximately tetrahedral, being very narrow and long near the plane through the two screw axes and wide, but very short at the end of the intermeshing zone (fig. III.10). In this gap no net drag flow occurs in counterrotating extruders since the walls move in opposite directions. Because of the form of this particular gap it can be concluded that the flow resistance is large in the small part and small in the wide part of the gap.

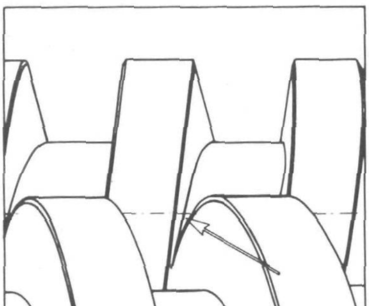

Fig. III.10

The tetrahedron gap.

Therefore the flow in the widest part is the most important. It can be concluded that entrance effects control most of the flow through the tetrahedron gap. This phenomenon together with the complicated geometry makes it extremely difficult to calculate this leak. Therefore an empirical formula has been developed for the Newtonian case. Since the moving boundaries are of no importance for the tetrahedron leak in counter-rotating screws simple models with stationary boundaries and various geometries can be used. A dimensional analysis and regression analysis of the measurements made with those simple models leads to:

$$\frac{Q_t \eta}{\Delta p R^3} = 0.0054 \left(\frac{H}{R}\right)^{1.8} \left(\Psi + 2 \left(\frac{\varepsilon}{H}\right)\right)^{2} \qquad \text{(III.39)}$$

where ε is the distance between the screw flanks in the plane through both screw axes (see fig III.3)

$$\varepsilon = \frac{1}{2} \left(\frac{S}{m} - B\right) \qquad \text{(III.40)}$$

III.6 THE FLIGHT GAP

The leak over the flight of the screws is produced by a pressure term and a drag term. Assuming the xy plane as indicated in figure III.11 and using a model with stationary screw and moving wall a force balance in the flight gap shows:

$$\frac{\partial p}{\partial x} = - \frac{\partial \tau}{\partial y} \qquad \text{(III.41)}$$

Providing the usual assumptions of incompressibility, steady flow, and constant viscosity are made and the acceleration terms can be neglected, for the Newtonian case this equation can be rewritten:

$$\frac{\partial p}{\partial x} = \eta \frac{\partial^2 v}{\partial y^2} \qquad \text{(III.42)}$$

The boundary conditions are:

$$\begin{aligned} y &= 0 \quad : \quad v = 0 \\ y &= \delta \quad : \quad v = w \end{aligned} \qquad \text{(III.43)}$$

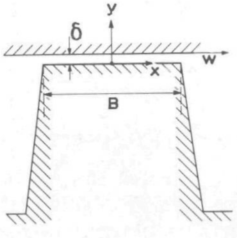

Fig. III.11
The flight gap.

where δ is the flight gap width and w the axial wall velocity as indicated in formula III.23.

If the processing fluid is Newtonian and the pressure gradient is assumed not to be a function of y the velocity in the gap is given by:

$$v = \frac{1}{2\eta} \frac{dp}{dx} (y^2 - \delta y) + \frac{wy}{\delta} \qquad \text{(III.44)}$$

Integration of this velocity over the height of the gap and multiplying this with the total flight gap length gives the flight leak:

$$Q_f = (2\pi - \alpha) R \left(\frac{w\delta}{2} + \frac{\delta^3}{12\eta} \frac{dp}{dx}\right) \qquad \text{(III.45)}$$

The contribution to the pressure drop over one flight arising from internal pressure generation in the chamber is given by equation III.28. Therefore the total pressure drop over one flight can be written

$$\Delta p = 6\eta \frac{w}{H^2} \left(\frac{S}{m} - B\right) + 2\Delta P \qquad \text{(III.46)}$$

where ΔP is the pressure drop between two consecutive opposed chambers due to die pressure. Since in a shallow channel the pressure drop can be approximated by:

$$\frac{dp}{dx} = \frac{\Delta p}{\Delta x} \qquad \text{(III.47)}$$

and in the flight gap

$$\Delta x = B \qquad \text{(III.48)}$$

equation III.45 can be rewritten as

$$Q_f = (2\pi - \alpha) R \left\{ \frac{w\delta}{2} + \frac{\delta^3}{6\eta B} \left[3\eta \frac{w}{H^2} \left(\frac{S}{m} - B\right) + \Delta P \right] \right\} \qquad \text{(III.49)}$$

III.7 THE CALENDER GAP

The theories that have been published for calendering (e.g. III.5, III.6) are not easily applicable to the flow between the intermeshing screws in a twin screw extruder or processor. Unlike that in a normal calendering process, the liquid layer passing through this gap is not discharged as a free film from the gap. As well as this, in a twin screw machine there is a net pressure drop across the calender gap. Nevertheless the calculations can follow those for the normal calendering process to a certain extent.

The velocity distribution again follows from the equation

$$\frac{\partial p}{\partial x} = -\frac{\partial \tau}{\partial y} \tag{III.50}$$

which again leads with a Newtonian fluid to

$$\frac{\partial p}{\partial x} = \eta \frac{\partial^2 v}{\partial y^2} \tag{III.51}$$

For the boundary conditions some approximations are made. Since the calender gap surfaces have different radii the velocities are different too. The analysis is sufficiently accurate if we assume the surfaces both to have the same radii, but rotating at different speeds, so

$$y = h \quad : \quad v = 2\pi N (R-H) \tag{III.52}$$
$$y = -h \quad : \quad v = 2\pi N R \tag{III.53}$$

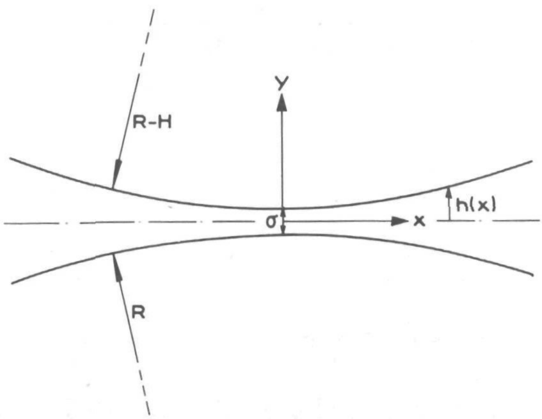

Fig. III.12

The calender gap

Here h is the distance from the centre line of the gap to the surfaces of the screws (see fig. III.12). Assuming the pressure gradient not to be a function of y, the velocity in the x-direction can be written, after straightforward integration as:

$$v_x = \frac{1}{2\eta}\frac{dp}{dx}(y^2-h^2) - \frac{\pi NH}{h}y + \pi N(2R - H) \tag{III.54}$$

Integration of this velocity from -h to +h and multiplication by the length of the gap (i.e. S/m-B) gives the throughput through the gap:

$$Q_c = (\frac{S}{m} - B)\left\{-\frac{2}{3\eta}\frac{dp}{dx}h^3 + 2\pi N\,(2R-H)h\right\} \qquad (III.55)$$

Assuming the circles of the boundary to be parabolics, this can be written:

$$x^2 = (2R-H)\,(h - \frac{\sigma}{2}) \qquad (III.56)$$

where σ is the calender width at the narrowest point. The error in this assumption is limited to the small differences that would result in the flow development. The velocity field at the narrowest point is not expected to be significantly at fault. Introducing the dimensionless variable

$$\xi = \frac{x}{\sqrt{(2R-H)\,\sigma/2}} \qquad (III.57)$$

equation III.55 becomes after some straightforward calculation:

$$\frac{dp}{d\xi} = 12\eta\,\frac{\sqrt{(2R-H)\,\sigma/2}}{\sigma^3}\left\{-\frac{m\,Q_c}{(S-Bm)(1+\xi^2)^3} + \frac{\pi N(2R-H)\sigma}{(1+\xi^2)^2}\right\}$$

(III.58)

Now pressure is a unique function of the dimensionless variable ξ and geometrical and process constants. Integration of pressure from minus infinity to plus infinity provides the pressure over the gap as a function of the throughput.

$$\Delta p = 6\pi\eta\,\frac{\sqrt{(2R-H)\sigma/2}}{\sigma^3}\left\{-\frac{3\,m\,Q_c}{4(S-Bm)} + N\pi(2R-H)\sigma\right\} \qquad (III.59)$$

The flow through the calender gap proves to be particulary sensitive to the spacing and the direction of the velocities of the approaching surfaces (i.e. whether the screws are co-rotating or counter-rotating). The pressure drop over the calender gap must indeed be the same as the sum of the pressure drop over a chamber in a tangential direction plus the contribution of the outlet pressure. Since the net flow through the chamber in tangential direction must indeed be the same as the flow through the calender and side gaps it is possible to calculate the throughput through the calender and side gaps as a function of die pressure.

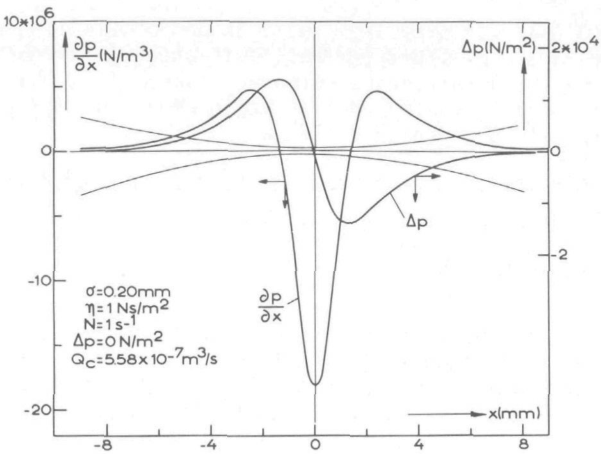

Fig. III.13

Pressure and pressure gradient in the calendergap (pressure difference = 0 N/m²)

A final remark has to be made about the error induced by integrating the pressure gradient from minus infinity to plus infinity while the calendergap is definite finite. To investigate this error a plot of pressure gradient and pressure as a function of the location in the calendergap is made. Figures III.13 and III.14 give the situation as obtained both with and without a pressure difference over the calender gap.

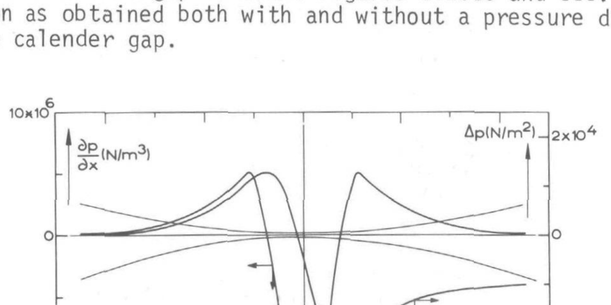

Fig. III.14

Pressure and pressure gradient in the calendergap (pressure difference = -1000 N/m²)

It can be seen that the pressure gradient approaches zero very quickly and therefore the pressure remains constant. Even when the calender gap is wide the error in the calender leak is less than three percent if the integration boundaries are changed to only plus and minus 1/3 of the total overlap distance.

III.8 THE SIDE GAP

Apart from the three basic gaps already dealt with, a fourth gap is commonly present which confuses the already complicated flow situation considerably.

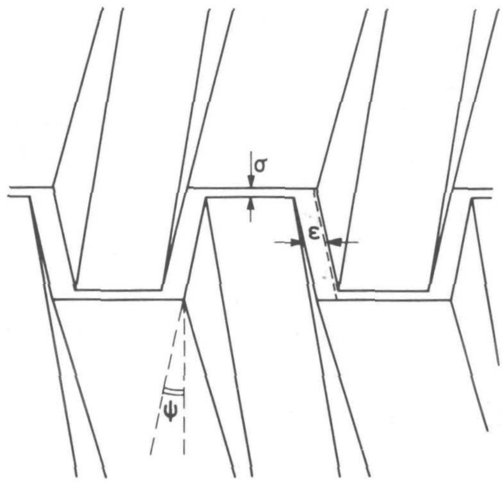

Fig. III.15

The side gap.

This is a gap between the flanks of the flights of the two screws through which there can be a flow from the upper to the lower side of the intermeshing zone. We will call this a side gap leakage. This leakage can be divided into a drag term and a pressure term. The velocities of the screw flanks vary linearly from $2\pi N(R-H)$ to $2\pi NR$. Neglecting the end effects (i.e. the contribution of the screw roots) the average drag velocity in the gap is

$$\bar{v} = \pi N(2R-H) \qquad \text{(III.60)}$$

As can be seen in figure III.15 the length of the gap is

$$\frac{H-\sigma}{\cos\psi} \qquad \text{(III.61)}$$

and the width is

$$\varepsilon \cos \Psi + \sigma \sin \Psi \qquad \text{(III.62)}$$

The total drag contribution of the side leakage is

$$Q_{s1} = \pi N(2R-H)(H-\sigma)(\varepsilon + \sigma \tan \Psi) \qquad \text{(III.63)}$$

where ε is the width of the side gap in the axial direction and σ is the width of the calender gap. Since there is no way to calculate the pressure dependent part because of geometrical complications and entrance effects an approximation formula for the flow through a rectangular duct is used.

Although the exact solution for this throughput is known (III.7)

$$Q_{s2} = \frac{p(H-\sigma)(\varepsilon+\sigma\tan \Psi)^3 \cos^2\Psi}{12\eta l}\left[1 - 192\,\frac{\varepsilon + \sigma \tan\Psi}{\pi^5\,(H-\sigma)} \cos^2\Psi + \sum_{i=1,3,5}^{\infty} i^{-5} \tanh\left(\tfrac{1}{2} i\pi\,\frac{H-\sigma}{\varepsilon + \sigma \tan \Psi} \cos^2\Psi\right)\right] \qquad \text{(III.64)}$$

a very good approximation (within 0.5%) is given by Weber (III.8):

$$Q_{s2} = \frac{\Delta p(H-\sigma)(\varepsilon+\sigma\tan \Psi)^3 \cos^2\Psi}{12\eta l}\left[1 - 0{,}630\,\frac{\varepsilon+\sigma\tan \Psi}{H-\sigma} \cos^2\Psi + 0{,}052\left(\frac{\varepsilon + \sigma \tan \Psi}{H-\sigma} \cos^2\Psi\right)^5\right] \qquad \text{(III.65)}$$

is used here. For l the length of the overlapping area is chosen.

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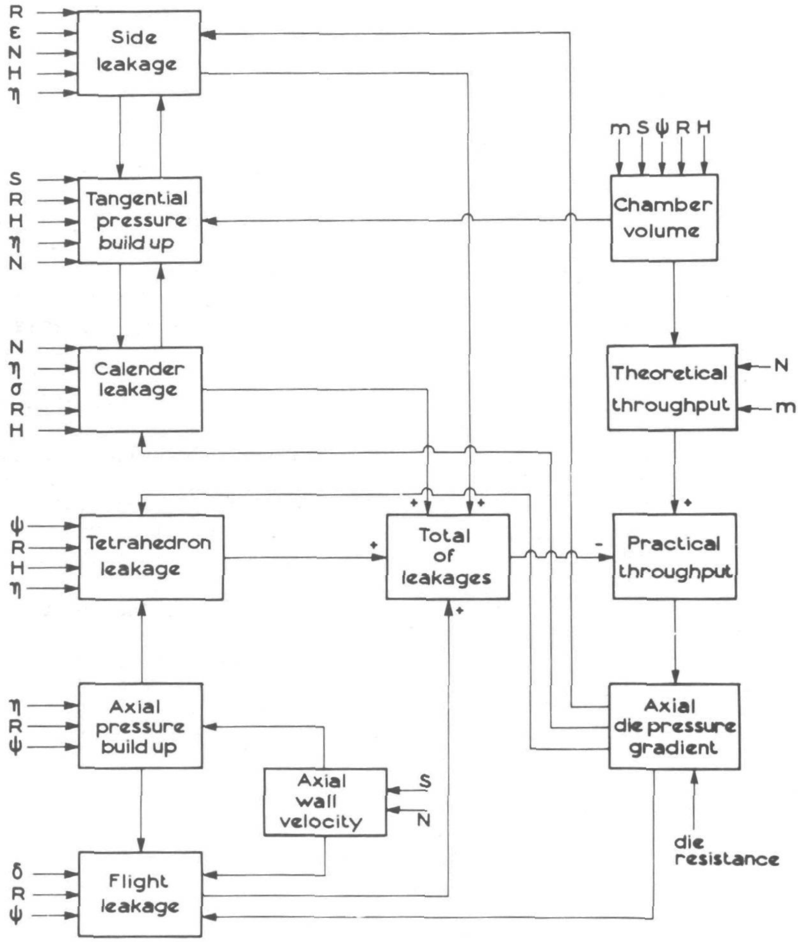

Fig. IV.1

Interactions between the several factors that control the throughput in a fully filled extruder.

CHAPTER IV

THROUGHPUT-PRESSURE RELATIONSHIP FOR THE MODEL EXTRUDER

IV.1 WORKING MECHANISM

In the preceding chapter the various factors that influence the output of a twin screw extruder have been given. In this chapter these are put together to obtain a simple model of a twin screw extruder, considering only the part of an extruder where the chambers are assumed to be completely filled with an isoviscous Newtonian fluid. The screws are assumed to have uniform profiles, which means that all chambers have the same shape. Although, as will be shown later, the length of real extruder screws that are fully filled changes for different die pressures while the pressure gradient and throughput remain the same, it is easier to understand a model in which the number of filled chambers is kept constant while the pressure gradient changes with the die pressure. In the model extruder this die pressure could be set by means of a valve at any value required and therefore the pressure gradient can be regarded as an independent variable, and the throughput becomes a dependent variable.

The interactions between the several factors are indicated in figure IV.1. This diagram is also an outline of a computer program using different subroutines for the various factors. As can be seen, the chamber volume, together with the rotation rate, gives the theoretical throughput as stated in formula I.1. By subtracting the total of leakages from this throughput the actual output is obtained. This actual throughput together with the die resistance gives the actual pressure gradient. The total amount of leakage is the simple addition of the four leakages that exist in the extruder, i.e. the calender leakage, the tetrahedron leakage, the flight leakage and the side leakage, all four of which are controlled by the axial die pressure gradient. Both the side leakage and the calender leakage are dependent on the tangential pressure build up. This particular drag pressure is further influenced by the side leakage, the calender leakage and the chamber volume. The tetrahedron leakage and the flight leakage are both influenced by the axial pressure build up. This drag pressure is influenced by the axial wall velocity which influences also the flight gap.

IV.2 INTERACTION BETWEEN CALENDER LEAK, SIDE LEAK AND TANGENTIAL PRESSURE GENERATION.

As stated before, the tangential pressure generation, as it occurs in the chambers of twin screw extruders, affects the leakages through the calender gap and the side gap. In turn these leakages control the tangential pressure generation. To put these phenomena together the expression for leakages and the pressure generation will be rewritten. The calender leakage and the side leakage can be combined as:

$$Q = Q_1 \Delta p_Q + Q_2 \qquad \text{(IV.1)}$$

The pressure generation can be expressed as:

$$q = q_1 \Delta p_q + q_2 \qquad \text{(IV.2)}$$

Following a closed path through the calender gap, the chamber and two tetrahedron gaps it follows that:

$$\Delta p_q + 2\Delta P - \Delta p_Q = 0 \qquad \text{(IV.3)}$$

where ΔP is the pressure drop over one tetrahedron leak due to the die pressure. Since the net flow through the chamber in tangential direction must be the same as the sum of the calender leakage and the side leakage it follows that:

$$q = Q \qquad \text{(IV.4)}$$

Using this last equation and inserting eq. IV.1 and eq. IV.2 in eq. IV.3 an over-all relationship can be obtained:

$$Q = \frac{2q_1Q_1}{q_1-Q_1} \Delta P + \frac{q_2Q_1-q_1Q_2}{q_1-Q_1} \qquad \text{(IV.5)}$$

which is a linear formula for the calender and side leakage as a function of the pressure drop per chamber transmitted back from the die.

IV.3 THROUGHPUT-PRESSURE RELATIONSHIP

A twin screw extruder with single thread start screws can be represented by a very simple model (figure IV.2).

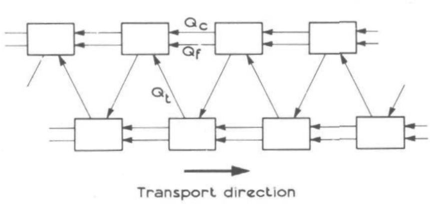

Fig. IV.2

Model for a twin screw extruder with single thread start screws.

In this model the whole extruder is shown as two series of chambers, one series moving from left to right for each screw. Between these chambers there is back leakage. The throughput from the extruder follows from a mass balance over a cross section:

$$Q = 2NV - Q_t - 2Q_f - 2(Q_c + Q_s) \qquad \text{(IV.6)}$$

According to figure IV.3 this throughput formula becomes, for screws with two thread starts:

$$Q = 4NV - Q_t - 2Q_f - 4(Q_c + Q_s) \qquad \text{(IV.7)}$$

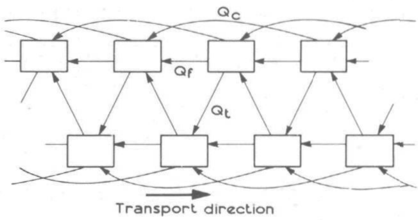

Fig. IV.3

Model for a twin screw extruder with double thread start screws.

In general, the throughput for screws with m thread starts is:

$$Q = 2mNV - Q_t - 2Q_f - 2m(Q_c + Q_s) \qquad \text{(IV.8)}$$

Using this equation, together with the formulae from the previous chapter, a throughput-pressure relationship for the filled length of an extruder can be obtained.

To check this model, experiments have been done with a model extruder. This extruder consists of a perspex body made in two halves which could be separated by narrow shims in order that the calender gap could be varied easily. Screw pairs of different geometry could be used in this extruder. The measurements were carried out with silicone oils of different viscosities. The output from the extruder could be closed by a needle valve so that the final pressure could be set at any value required. Both transducers and a bourdon-tube were used for the pressure measurements. It was possible to check the symmetry of the pressure between the two screws at both input and output points. In addition the linearity of the pressure gradient within the extruder could be checked by pressure transducers at 1/3 and 2/3 of the length along the extruder. These measurements of linearity and symmetry in the pressure were completely satisfactory.

Figures IV.4 and IV.5 show the measurements compared with theory for models completely filled with Newtonian liquid. The screw geometries are given in tables IV.i and IV.ii. At the horizontal axis in figures IV.4 and IV.5 the dimensionless die pressure per chamber is shown. ΔP is the die pressure divided by the total number of chambers in the model, η is the dynamic viscosity of the process fluid and N is the rotational speed in revolutions per unit time.

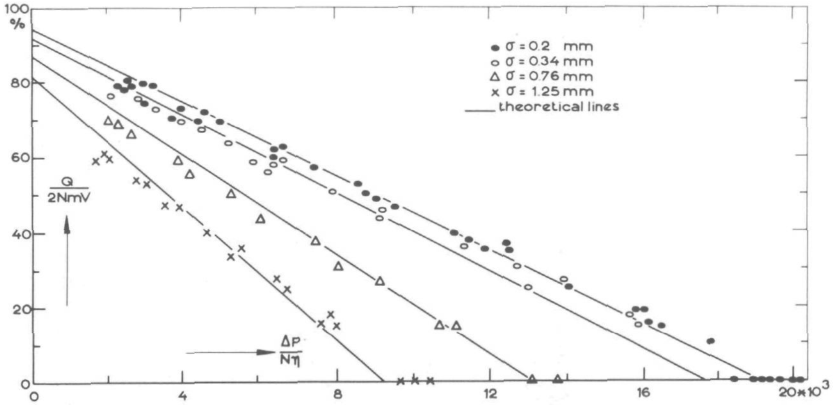

Fig. IV.4

Measurements compared with theory for the extruder described in table IV.i.

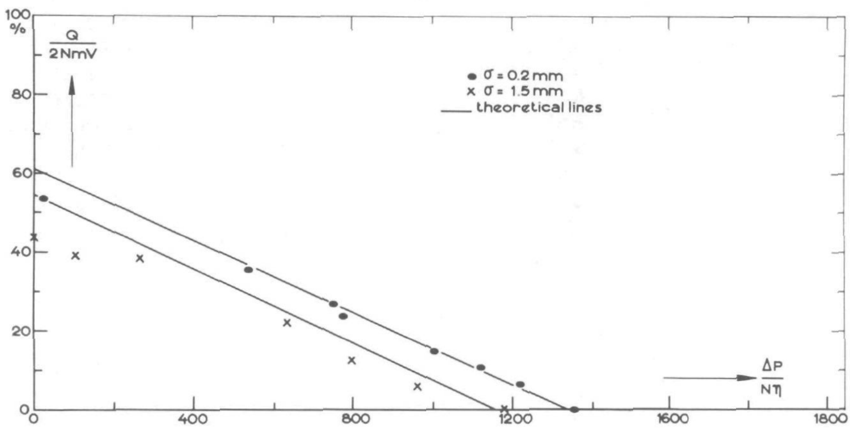

Fig. IV.5

Measurements compared with theory for the extruder described in table IV.ii.

17,214 cm^3

R	*screw radius*	*35 mm.*
S	*pitch*	*20 mm.*
Ψ	*flight wall angle*	*14^o*
H	*chamber depth*	*10 mm.*
δ	*flight gap*	*0.25 mm.*
ε	*side gap*	*0.05 mm.*

Table IV.i

R	*screw radius*	*35 mm.*
S	*pitch*	*25 mm.*
Ψ	*flight wall angle*	*5.14^o*
H	*chamber depth*	*10 mm.*
δ	*flight gap*	*0.1 mm.*
ε	*side gap*	*4 mm.*

Table IV.ii

The vertical axis shows the actual throughput which is made dimensionless by dividing it by the maximal theoretical throughput given by equation I.1. The abscissa intercept indicates the maximal dimensionless pressure that the extruder can build up. This is the pressure developed with a completely closed discharge channel. In this case the total amount of leakage is equal to the maximum theoretical throughput. The ordinate intercept indicates the maximum practical output as a fraction of the maximal theoretical output. In this case the die pressure is zero and the only leakage flows are produced from drag flow and pressure flow due to internal pressure generation.

The two dimensionless numbers used for throughput and pressure are rather powerful. When using those two numbers the throughput-pressure relationships for twin screw extruders or processors completely filled with Newtonian liquids are straight lines, independent of viscosity and rotational speed. Because these lines are only dependent on screw geometry and not on operating conditions the throughput-pressure relationship of a certain model can be characterised by two points (the abscissa and the ordinate intercepts for instance). Moreover when using these two dimensionless numbers for throughput and pressure, the characteristics are retained when the extruder is scaled up geometrically. When for instance radius, pitch, chamber height, calender gap, flight gap and side gap are all enlarged by the same factor, the dimensionless throughput-pressure graph remains valid.

IV.4 SENSITIVITY ANALYSIS

In order to obtain insight in the importance of the geometrical variables for the throughput-pressure relationship a sensitivity analysis has been made using computer programs based on the calculations in the previous chapter. A standard geometry given in table IV.iii has been assumed. Each time one geometrical variable is changed.

Fig. IV.6 shows the influence of the tetrahedron gap. If no back pressure is applied, the throughput is negligibly affected by the variations of this gap. The pressure dependency of this gap i.e. the variations in maximal pressure built up is, on the other hand, considerable.

The size of the calender gap affects the zero back pressure throughput considerably, as can be seen from fig. IV.7.

R	*screw radius*	35 mm.
S	*pitch*	20 mm.
Ψ	*flight wall angle*	6^0
H	*chamber depth*	10 mm.
δ	*flight gap*	0.1 mm.
ε	*side gap*	0 mm.
σ	*calender gap*	0.2 mm.

Table IV.iii

Standard geometry.

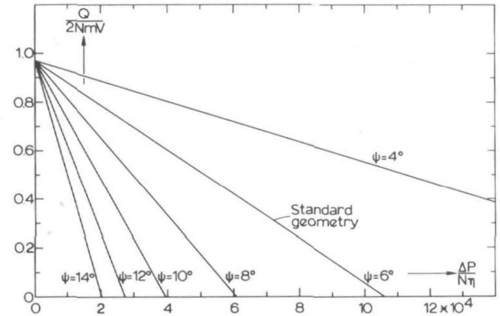

Fig. IV.6

Sensitivity to variations of the tetrahedron geometry

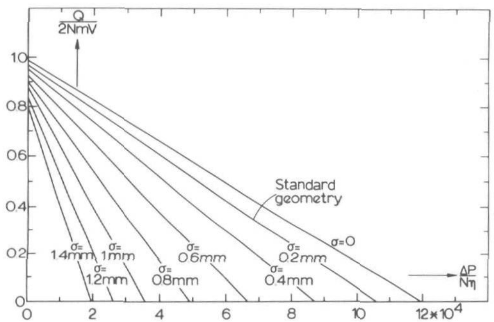

Fig. IV.7

Sensitivity to variations of the calender gap width

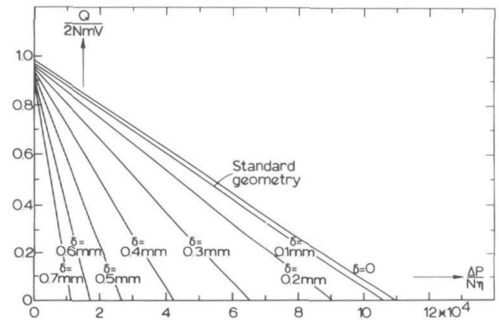

Fig. IV.8

Sensitivity to variations of the flight gap width.

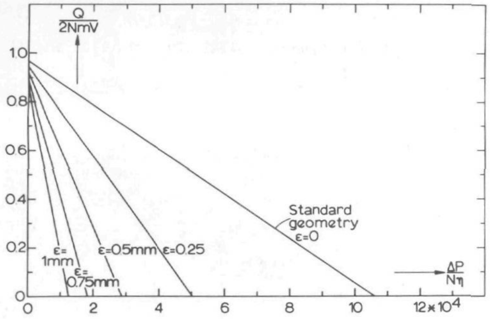

Fig. IV.9

Sensitivity to variations of the side gap width.

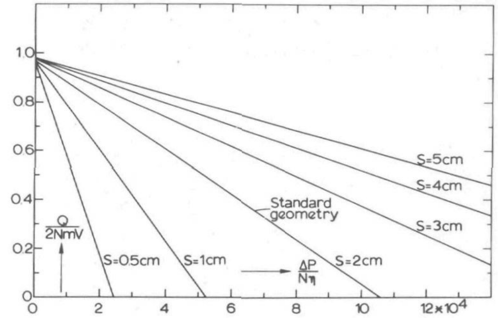

Fig. IV.10

Sensitivity to variations of the pitch

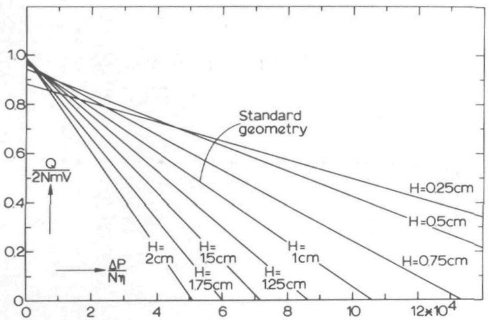

Fig. IV.11

Sensitivity to variations of the chamber height.

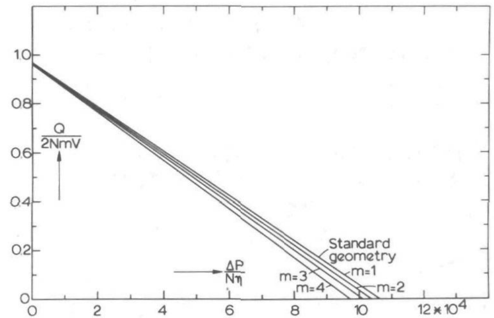

Fig. IV.12

Sensitivity to variations of the number of thread starts

This is to be expected since internal pressure generation in the chamber as well as in the calender gap itself is very high, and moreover there is a considerable drag flow in this gap.

From figure IV.8 it can be seen that the flight clearance in a twin screw processor is rather important. The throughput at zero die pressure is about as sensitive to the width of the calender gap as to the width of the flight gap. The pressure generation under zero net flow conditions on the other hand decreases very rapidly with increasing flight gap.

Although to a lesser extent than the flight gap, the side gap also has a great influence on the pressure generation, as can be seen from figure IV.9.

Other geometrical variables affect more than one gap. The effect of the variation of the pitch is shown in figure IV.10. Since a large pitch leads to a large chamber volume, the leakage flows must also be large when the die is closed. Only the length of the calender gap changes with the pitch, while the geometries of all other gaps remain similar. For this reason the pressure that is built up when the die is closed is large for such screws.

The chamber height (fig. IV.11) affects both the chamber volume and the tetrahedron gap. Since the tetrahedron leak is of minor importance when there is no die pressure, deep chambers give relatively high output rates at very low back pressures. Since the angle of the tetrahedron gap remains constant the surface of this gap increases quadratically with the chamber height, while the chamber volume increases linearily. This explains why the pressure developed with a closed die increases when the chamber height decreases.

The influence of the number of thread starts is very small when all other variables remain constant, as can be seen from figure IV.12. Nevertheless it has to be realised that the abscissa is the dimensionless pressure per chamber. Increasing the number of thread starts will therefore affect the pressure drop per unit screw length almost linearily. When working with more thread starts and the same pitch the number of chambers per unit screw length will increase, so this extruder can build up a greater pressure per unit length.

The last dependency to be discussed is the screw diameter. When all other variables remain constant, variations in screw diameter do not have much affect on the process. Even if the diameter of the screw is varied by a factor of five (3 to 15 cm.), the dimensionless output and closing pressure only change by about one and six percent respectively, an amount that cannot be satisfactorily represented in a figure.

IV.5 INFLUENCE OF NON NEWTONIAN LIQUID PROPERTIES.

All the models described previously are based on the assumption that the processing fluids have a Newtonian rheology i.e. the viscosity is independent of the imposed shear rates. The experiments described in the previous sections were limited to the use of Newtonian liquids. Although this simplification is very useful in obtaining better understanding of the mechanisms that control the process it is a fact that most molten polymers are extremely non-Newtonian.

In a complex geometry like that of a twin screw extruder this is a particularly severe complication. Within the leakage gaps the shear rates can vary considerably due to moving walls and pressure flows. To obtain a qualitative insight into the deviations of the theory when using non-Newtonian instead of Newtonian liquids, some experiments have been done with a sodium carboxyl methyl cellulose (C.M.C.) solution in water. The rheology of this solution can be described as a power law fluid,

$$\tau = -k \left| \frac{\partial v_x}{\partial y} \right|^{n-1} \frac{\partial v_x}{\partial y} \qquad \text{(IV.9)}$$

and therefore the apparent viscosity is

$$\eta_a = k \left| \frac{\partial v_x}{\partial y} \right|^{n-1} \qquad \text{(IV.10)}$$

k is the consistency and n is called the power law index. From formula IV.9 it can be seen that for a Newtonian fluid the power law index is 1 and that consistency is idential with viscosity. Since for CMC solutions the power law index is less than one, the liquid is shear thinning with an apparent viscosity that decreases with increasing shear rates.

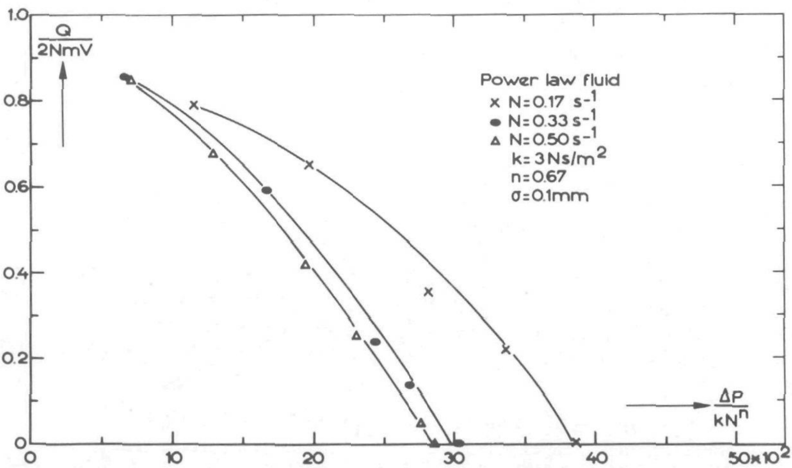

Fig. IV.13

Dimensionless pressure to throughput relationship for CMC solutions.

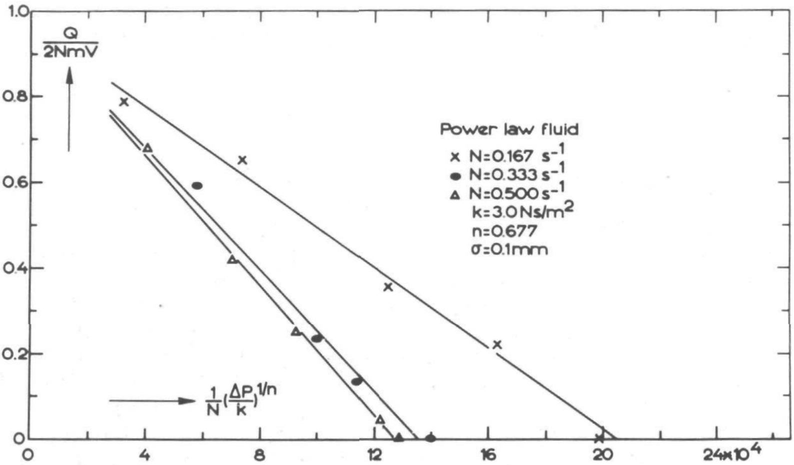

Fig. IV.14

Linearised dimensionless pressure to throughput relationship for CMC solutions; calender-gap width = 0.1 mm.

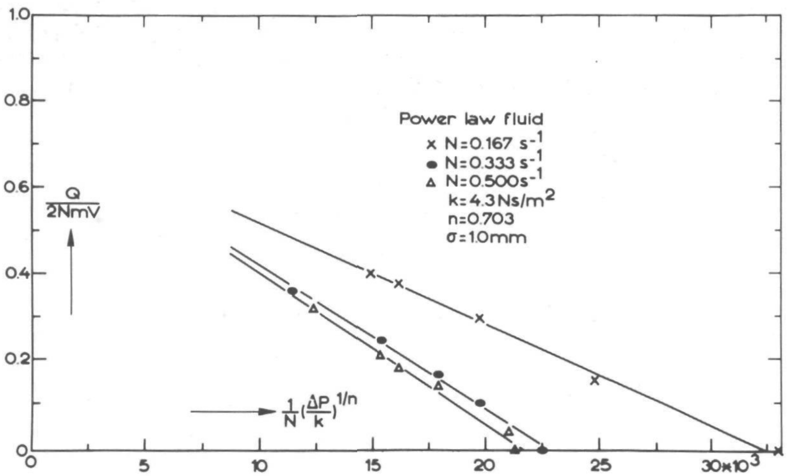

Fig. IV.15

Linearised dimensionless pressure to throughput relationship for CMC solutions. Calendergap width = 1.0 mm.

Figure IV.13 shows the dimensionless pressure to throughput relationship for a CMC solution. In order to obtain a dimensionless group the pressure term has to be expressed in the more general form

$$\frac{\Delta P}{kN^n} \qquad \text{(IV.11)}$$

This form is convenient in that it enables direct comparison of generated pressure characteristics for various fluids. It can be seen from figure IV.13 for example that the dimensionless closing pressure is lower than in the Newtonian case. This is due to the shear thinning character of the CMC solution. In the leakage gaps the shear rates are high because of the moving walls and the high velocities of the fluid when the die is closed. Therefore the apparent viscosity is low in these gaps and the extruder cannot build up as high a pressure when there is considerable die resistance. This effect is further exaggerated since the fluid in the chambers is less affected by the non-Newtonian properties of the liquid, because the shear rates in the chambers themselves are generally lower than those in the leakage gaps.

However the lines are no longer straight. Linearity can be achieved if the dimensionless pressure is expressed as

$$\frac{1}{N}\left(\frac{\Delta P}{k}\right)^{1/n} \qquad \text{(IV.12)}$$

This number is also consistent with the Newtonian case. These graphs are given in figures IV.14 and IV.15. It is clear that in both cases dimensionless pressure to throughput relationship is no longer independent of rotation rate, but that the dimensionless pressure that can be generated with the extruder decreases with increasing screw speed.

CHAPTER V

FLOW, MIXING AND RESIDENCE TIME

V.1 INTRODUCTION

A new area, so far hardly developed in which the twin screw geometry might have great advantages is in the use of a twin screw machine as a chemical reactor. The only information reported for this kind of application is for bulk polymerization (V.1, V.2). Knowledge of the mixing is especially important when dealing with slow reactions since the degree and scale of mixing influence the reaction time considerably (V.1). On the other hand the residence time distribution of the process is important in order to obtain a homogeneous end product. For instance in the twin screw polymerization reactor already mentioned the molecular weight distribution is largely affected by the spread in residence times of the reacting materials in the equipment. When using the twin screw geometry for "plain" extrusion these factors are also important. Good mixing improves the homogeneity of the polymer leaving the extruder and, if this can be achieved with a narrow residence time distribution, all the material will have the same history which will also improve homogeneity.

V.2 FLOW AND MIXING WITHIN ONE CHAMBER

To obtain more insight into the phenomena dealt with in the previous paragraph, some qualitative experiments have been made on the flow and mixing in the extruder. A model extruder with a perspex barrel has been used with aqueous polyvinylpyrolidone solutions. To establish the flow profiles aluminium particles and coloured dyes were injected. The stream lines were observed by television and video recorded. Afterwards these streamlines could be traced during frame by frame video replay of the tape.

Figure V.1 represents a single C shaped chamber as it exists within the extruder. In this chamber the flow can clearly be divided into two parts, the flow far from the region where the screws are intermeshing and the flow near that particular region.

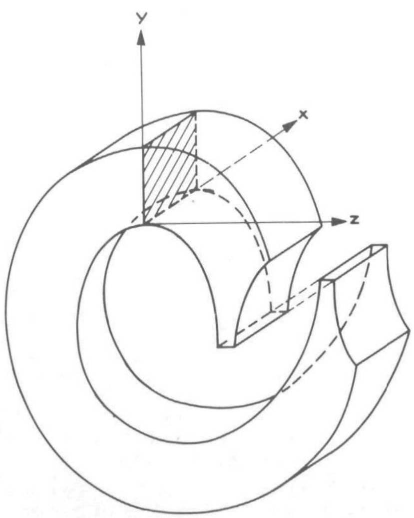

Fig. V.1

C shaped chamber with coordinates.

It is useful to divide the flow far from the intermeshing zone into two components, a component in the x-y plane and a component in the z direction. The flow in the x-y plane is similar to that in single screw extruders. Although twin screw extruders have in general more deeply cut channels than single screw extruders, the more sophisticated single screw extruder calculation techniques (e.g. V.3) are also applicable to twin screw extruders.

Rieger and Sesták (V.4) described an analytical method for the solution of the flow in the z direction in a channel with walls perpendicular to the screw root and a stationary screw and a moving barrel wall. In order to establish the influence of the flight wall angle ($\Psi \neq 0$) numerical calculations have been made.

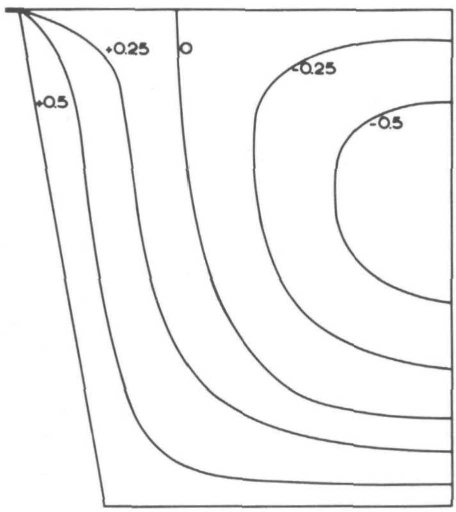

Fig. V.2

Flow in the tangential direction with isotach numbers.

A computer program using a grid of 20 x 20 points with a standard 5 point module (e.g. V.5) was used. From these it was learned that the flow field is rather insensitive to the flank angle, at least for normal twin screw extruder screws with flank angles Ψ of less than 14^{o}. From both the analytical and numerical solutions it follows that there is a counter-current flow within the chambers and a well defined zero velocity contour as indicated in figure V.2. The isotach numbers in this figure indicate the local velocity made dimensionless by dividing by the circumferential velocity at the outer radius.

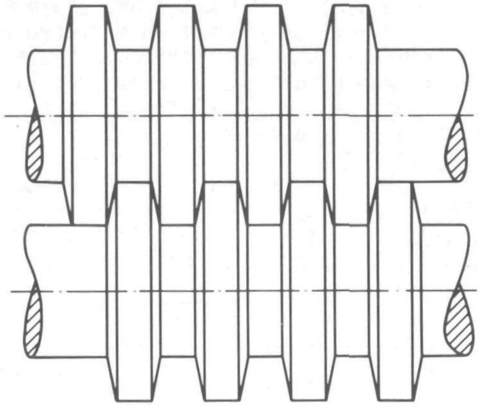

Fig. V.3

Disk model used to investigate the flow near the intersecting area.

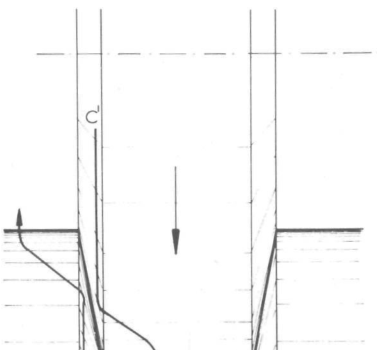

Fig. V.4

Streamlines near the intersecting area.

Since the flow near the intersecting area cannot be analysed either analytically or numerically, it was investigated by flow visualization with colour injection and video recording. To simplify the complex situation a disk model was used. This model, shown in figure V.3, can be regarded as a screw without pitch. Figure V.4 indicates the channel near the intersecting area at the converging side of the screws. Comparing three stream lines A, B and C approaching the intersecting area at the same height in the chamber the influence of the flight of the other screw can be determined. Streamline A which is quite near to the zero velocity layer returns at the same height in the chamber. Streamline B returns at a place much nearer to the intersecting area. It comes into the region of influence of the flight of the other screw and will therefore be moved towards the bottom of the channel.

Streamline C which is close to the channel wall passes through the tetrahedron gap to the opposite chamber. Its counterpart C' comes from the other chamber through the tetrahedron gap and is strongly affected by the flight of the other screw. In fact, it will be moved to the bottom of the channel.

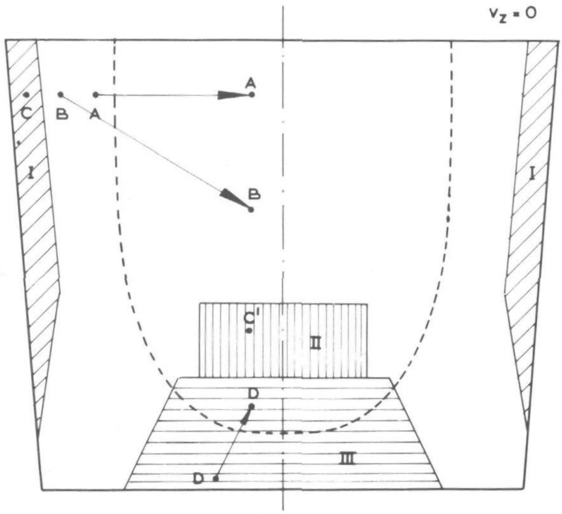

Fig. V.5

"where from / where to" balance over a cross section of the chamber.

Putting together a "where from/where to" balance over a cross section of the chamber related to the converging side of the screws, figure V.5 emerges. The fluid in the regions near the flight wall, (I), transfers to the opposite chamber while the corresponding fluid from the opposite chamber will be found in a region near the middle of the channel, (II). A horizontal plane that goes to the converging side of the screws (AB) returns as a vertical plane. Near the bottom of the channel (III), a remarkable closed and stable flow exists that mixes very poorly with the rest of the chamber, unless the calender gap is relatively large. In a model with intermeshing disks and no significant calender gap this volume remains in place for a very long time (figure V.6). In a processor with real screws it was seen from decoloration experiments that the bottom volume mixes very slowly with the bulk of the chamber. Such mixing as does occur is produced mainly by a rotation of the processing fluid in the x-y plane. Only if the calender gap was relatively wide, is fluid from this region drawn into this gap where it is sheared and then mixed with that in the rest of the chamber.

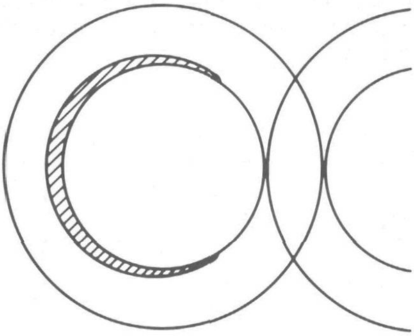

Fig. V.6

Unmixed zone near the bottom of the chamber.

This phenomenon can explain the measurements of Kim, et al (V.6), who used a twin screw extruder with soft p.v.c. and aluminium powder. When the process was at steady state the extruder was stopped, cooled and dismantled. The dielectric constant of samples taken out of the C shaped chambers were measured and the mixing calculated using a variationalcoefficient.

$$F = \frac{100}{\overline{c}}\sqrt{\frac{1}{M-1}\sum_{i=1}^{M}(c_i-\overline{c})^2} \qquad (V.1)$$

where M is the numer of samples, c_i the aluminium concentration in a particular sample and $\overline{c}$ the average aluminium concentration over a series of samples. They found that in the middle of the chambers the variational coefficient was much higher than near the intermeshing zone. This is shown in figure V.7, where the change of the variational coefficient with the C shaped chamber (from intermeshing zone to intermeshing zone) is given for different lengths of the fully filled zone in the extruder.

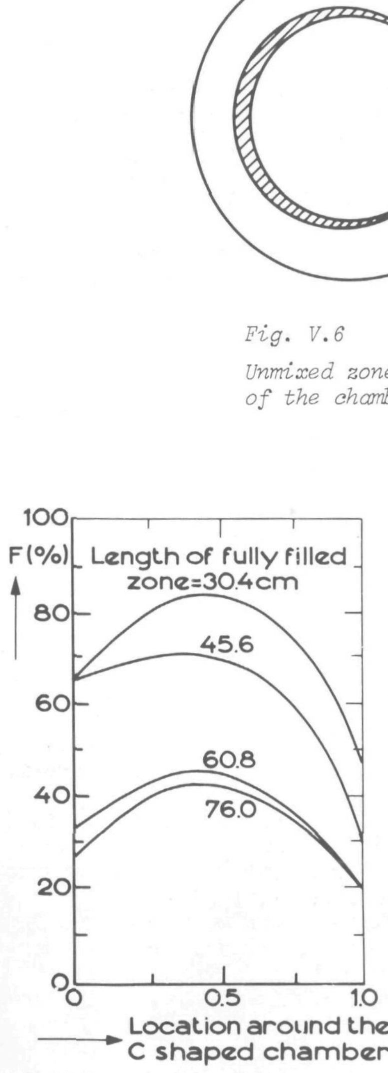

Fig. V.7

Mixing along one C shaped chamber expressed in terms of a variational coefficient.

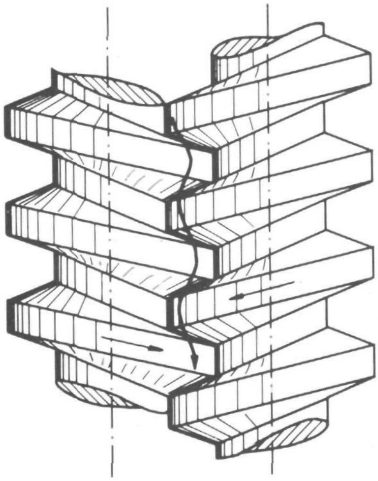

Fig. V.8

Leakage flows, large back pressure, no significant calender gap.

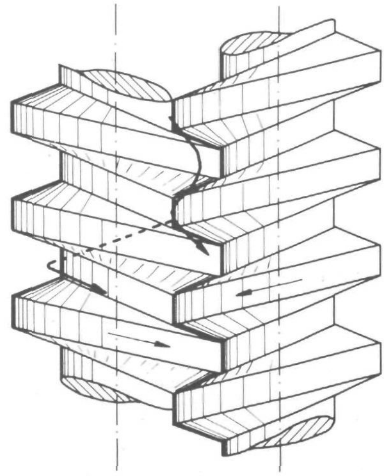

Fig. V.9

Leakage flows, large back pressure, wide calender gap

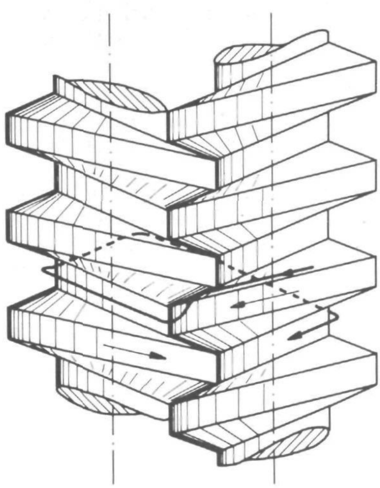

Fig. V.10

Leakage flows, small back pressure, no significant calender gap.

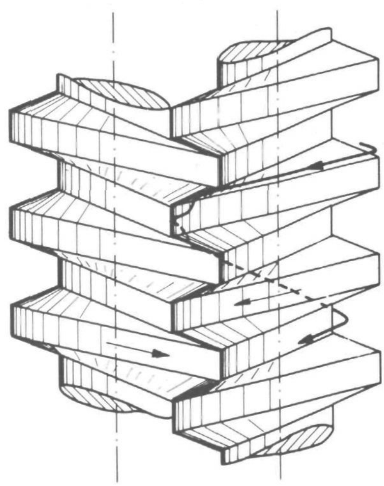

Fig. V.11

Leakage flows, small back pressure, wide calender gap

V.3 INTERACTION BETWEEN THE CHAMBERS

In order to understand the mixing process in a twin screw reactor it is not sufficient only to know the magnitude of the leaks, which was sufficient for the throughput model.

It is also necessary to know the interactions of the leakage flows with the fluid in the bulk of the chambers and with each other. Four different extreme situations can be distinguished. The processor can have a large calender gap or no significant calender gap coupled with either a large or small back pressure during operation. In these experiments with fully filled screws, the low back pressure condition was obtained with a nearly open discharge so that the output rate was about 80% of the positive displacement volume rate, corresponding to a total leakage flow at any cross section of about 20%. The throttled discharge used to produce the high back pressure approximately reversed these proportions. A large back pressure when there is no significant calender gap gives poor mixing (fig. V.8). It has been seen that the leakage flow goes straight through the tetrahedron gaps and hardly mixes with the fluid in the chambers.

With a similar large back pressure but when there is a wide calender gap (fig. V.9) some mixing occurs between the leakage fluid and the fluid in the bulk of the chamber.

With small back pressure and no calender gap the flow is as shown in figure V.10. The leakage passes through the tetrahedron gap at the converging side of the screws, through the centre of the next chamber without being mixed up well and then through the tetrahedron gap at the diverging side of the screws. The same mechanism is repeated. If the colour injection is made at the right place in the chamber, it can stay compact for a long time without really being mixed with the fluid in the rest of the chambers.

The last case, low back pressure and a fairly big calender gap gives a situation shown in figure V.11. The tracer goes through the tetrahedron gap at the converging side of the screws. Then it passes immediatly into the calender gap and at the diverging side of the extruder it goes through the next tetrahedron gap. The back mixing thus misses one chamber entirely. This is clearly visible if a big coloured pulse is injected at one point in the extruder. After some time the colour can very clearly be seen well mixed in alternate chambers along one screw only. The other chambers of that same screw and all the chambers of the other screw remain uncoloured. The micro mixing i.e. the mixing within one chamber, is generally good in this last case. It can clearly be seen from the figures that there is an interaction between the calender leakage and the tetrahedron leakage. The leakage through the flight gap stands completely on its own.

V.4 RESIDENCE TIME DISTRIBUTION

As already stated in chapter II, a narrow distribution of residence time is repeatedly claimed in literature as one of the major advantages of a twin extruder. To obtain more quantitative knowledge than this vague statement, measurements were done with an isoviscous Newtonian fluid in a model extruder with close clearance 70. mm. screws.

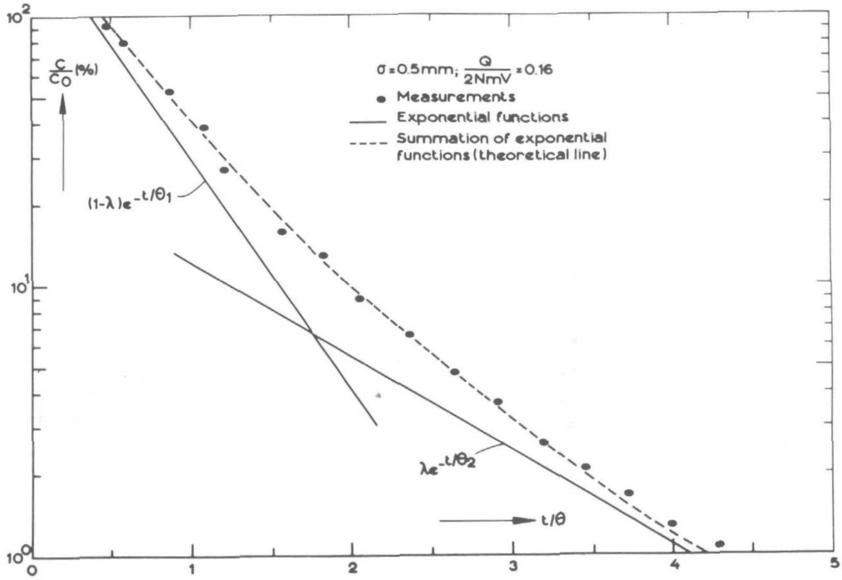

Fig. V.12

Exit age distribution with the two exponential functions and their summation.

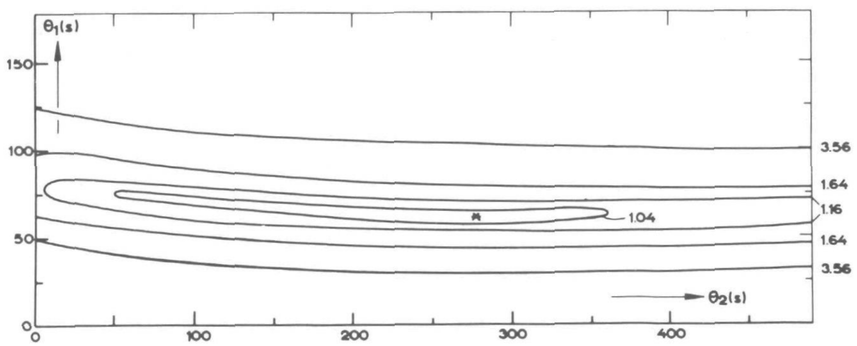

Fig. V.13

Contour plot of the sum of the squares of the deviations from the experiments and the curve fit as a function of the time constants.

At time = 0 a step change to transparent fluid was imposed on the extruder which was operating completely filled with blue coloured fluid. The change in composition of the fluid at the outlet gives a direct measure of the cumulative exit age distribution since the colour measured at time t corresponds to material that has been held in the extruder for time t or more. The change of colour was made by two valves mounted directly above both screws. The time needed to change the feed from blue to transparent fluid was less than 0.1 second. Since the mean residence time in the extruder was of the order of minutes, no serious error was developed at the input. The end of the extruder was closed by two orifices very close to each screw instead of valves. The output from these orifices fell freely into sample tubes. In this way the superimposition of an exit residence time distribution on the machine residence time was avoided. The fluid used was an aqueous polyvinylpyrolidone solution which was virtually Newtonian. This fluid was coloured with methylene blue. The methylene blue concentration in the sampled fluid was measured with a photo-spectrometer. Concentrations as low as one percent of the initial value could be measured accurately. In these measurements the distance between the screws (controlling the calender gap) and the die pressure were varied.

A semi logarithmic representation of an exit age distribution is shown in figure V.12. It appeared that the distribution could be represented by the summation of two exponential functions

$$\frac{C}{C_o} = (1 - \lambda)\, e^{-t/\Theta_1} + \lambda\, e^{-t/\Theta_2} \qquad \text{(V.11)}$$

where Θ_1 and Θ_2 are time constants and λ is the distribution ratio In order to calculate the various factors a least squares criterion was used. To obtain an insight into the accuracy of the method used a representative plot of the sum of the squares of the deviations between the experimental points and the fitted line at different Θ_1 and Θ_2 is shown at a fixed distribution rate λ (fig. V.13). The best values of the constants to fit a given set of experimental data can be determined by establishing the minimum of this contour plot. If the sum of the squares of the deviations is the same at different locations in the plot then each such set of constants results in an equally good fit to the experimental data. The major problem in the fit concerned is that the sum of the squares of the deviations forms a long narrow valley within which many combinations of constants give an almost equally good fit. For the most accurate determination of the constants best representing a set of experimental data, the contour plot should show a deep valley steep sided in all directions in the vicinity of the optimum point. This can be obtained by giving the measured values at low concentrations a higher weighting factor, which is only justified if the measurements at lower concentrations are more accurate. Therefore these measurements were done with extreme care.

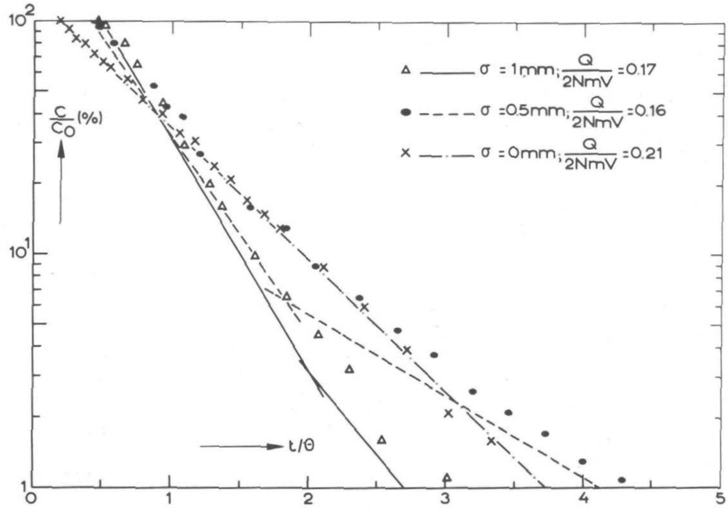

Fig. V.14

Change with time of output concentration following a step change of input concentration at time = 0. Large back pressure.

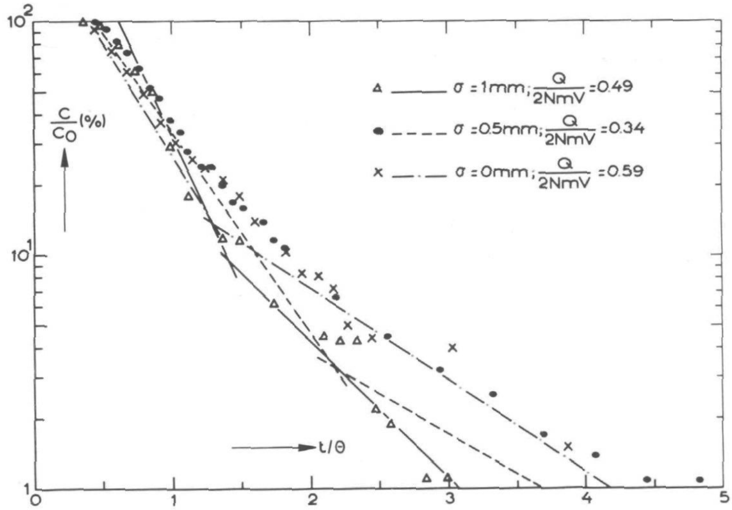

Fig. V.15

Change with time of output concentration following a step change of input concentration at time = 0. Medium back pressure.

For all measurements at low concentrations the same cuvette was used in the photospectrometer and this was carefully cleaned and dried after each measurement.

In the figures V.14, V.15 and V.16 apart from the experimental points the lines representing the two exponential functions, which added together give the "best fit" to a curve through the experimental points are drawn.

Figure V.14 shows the exit age distribution at low output rates, i.e. with a large back pressure. Three lines are shown, one measured with no significant calender gap, one with ½ mm. and one with 1 mm. calender gap. The vertical axis is the output concentration expressed as a percentage of the initial concentration at time = 0 given logarithmically. On the horizontal axis the time t is divided by the time constant of the system, Θ. This time constant can be obtained by dividing the total fluid volume in the extruder by the actual throughput. For a small output rate (figure V.14) the exponent lines have no or relatively small kinks, i.e. the second term of eq. (V.2) is small or substantially zero.

For medium output rates (figure V.15) non-linearities appear in the lines. When working with a big calender gap the "tail" of the exit age distribution remains small, which is represented by the relatively steep line at low concentrations.

Figure V.16 shows the situation at high output rates, i.e. for small back pressure. Here large non-linearities appear except when working with big calender gaps, corresponding to significant values of Θ_2 and λ.

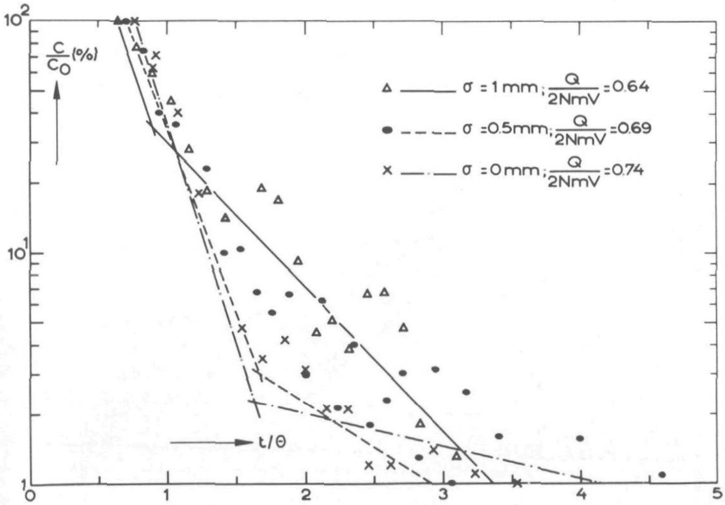

Fig. V.16

Change with time of output concentration following a step change of input concentration at time = 0. Small back pressure.

Values of the time constants and distribution coefficient defined in eq. V.II are given in table V.i for the experiments represented in figures V.14, 15 and 16. which are typical of all the results of these residence time distribution experiments. It has not so far been possible to extract the quantitative information that is needed before a good model description of the interaction between the design and operating parameters and the mixing performance of the extruder can be formulated, but clearly this approach will justify a further experimental programme.

$Q/2NmV$	σ	λ	Θ_1	Θ_2
0.74	0	0.03	13	142
0.59	0	0.28	50	142
0.21	0	–	185	–
0.69	0.5	0.072	21	103
0.34	0.5	0.13	111	357
0.16	0.5	0.20	141	333
0.64	1	0.50	16	50′
0.49	1	0.27	38	85
0.17	1	0.35	149	187

Some of the measurements of Todd (V.7, V.8) are also non-linear when transformed into the coordinate system used here. The non linearities reflect at least a partial segregation of the fluid passing through the extruder, and by implication a limited amount of mixing within the chambers. Only when using a wide calender gap or, as follows from Todd's measurements, a small screw angle,is the non linearity not significant suggesting that this geometry would be the most suitable when good homogenization is required, as may be the case for chemical reactions. Together with the interactions between the chambers discussed in the previous paragraph there is indirect evidence that the calender gap controls the homogenization in the extruder. When there is no calender gap at all the leakage flows mix poorly with the rest of the fluid in the chambers. This allows some of the material to remain quite a long time in the extruder. Therefore the residence time distribution has an extra long tail. When using a reasonable calender gap, all or most of the tetrahedron leakage will be drawn into the calender gap where it will be sheared and mixed, and no kink in the logarithmic output concentration curve is observed.

V.5 COMPARISON OF SOME EXIT AGE DISTRIBUTION CURVES

It is interesting to compare several exit age distributions. The measurements described before were done in a fully filled extruder under well defined conditions.

Todd (V.7, V.8) has performed measurements in a starving twin screw extruder with different screw geometries. Pinto and Tadmor (V.9) developed a model calculation for fully filled single screw extruders, which has been subsequently verified by other authors (V.10, V.11). A comparison can also be made with Poiseulle flow in a straight tube as calculated under various conditions by Taylor (V.12).

In figure V.17 it can be seen that the dimensionless internal age distribution characteristics for twin screw extruders vary for different geometries and operating conditions. In contrast to this, the characteristics for single screw extruders are all represented by the same curve. A more interesting conclusion is that the internal age distributions are not necessarily significantly better than the theoretical model and measurements for single screw extruders.

The fact that the twin screw machine has been found to have clear practical advantages as a processor might lie in the fact that the number of fully filled chambers is limited, as will be seen, though this has yet to be conclusively proved.

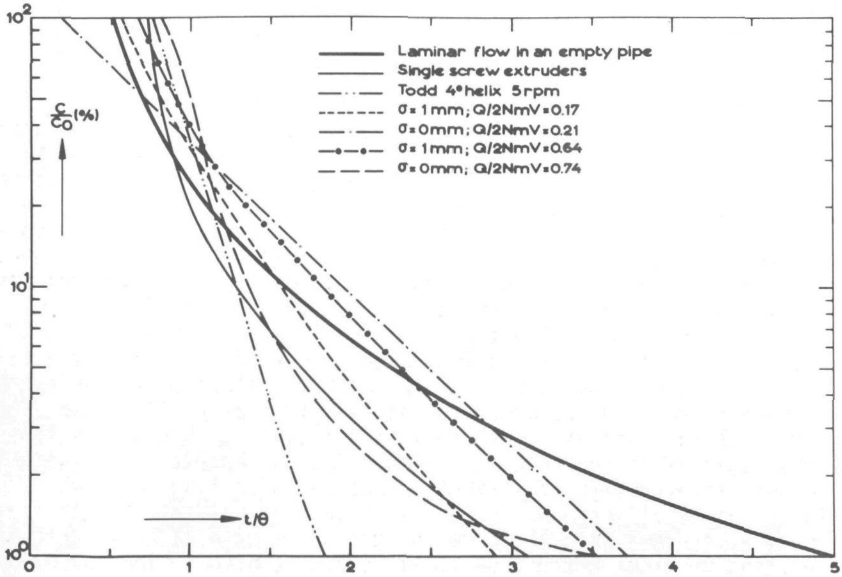

Fig. V.17

Comparison of some dimensionless exit age distribution curves.

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CHAPTER VI

OUTPUT AND PRESSURE BUILD UP IN A REAL TWIN SCREW EXTRUDER

VI.1 AN ISOVISCOUS MODEL

The main differences between the model experiments and real twin screw extrusion lie on the one hand in the melting mechanism and on the other in that the chambers are not all fully filled. In a real extruder polymer (particles or powder) enters the screws in the filling region. Because of the porosity of the particle bed and the limited ability of the hopper to fill the screws, the resulting melt does not completely fill the chambers. This can be expressed in terms of a filling degree which will be indicated by u. To obtain better filling, twin screw extruders often have a separate feeder. Alternatively their screws have a different geometry just beneath the hopper with a greater pitch and more thread starts than the main working length. The positive displacement capacity of such a filling zone is greater than that of the rest of the extruder. This can be expressed by a compression factor

$$\mu = \frac{m_\phi V_\phi}{mV} \qquad \text{(VI.1)}$$

Here m is the number of thread starts and V is the volume of one C shaped chamber, where the subscript ϕ indicates the filling zone. The volumetric throughput in the filling zone is

$$Q = 2\,N\,m_\phi\,V_\phi\,u = 2\,N\,m\,V\,\mu\,u \qquad \text{(VI.2)}$$

In single screw extruders a continuous, though perhaps particulate, solid bed is generally present, through which pressure can possibly be transmitted. In twin screw extruders on the other hand the solid bed is repeatedly divided by the intermeshing flights of the other screw. Therefore in a twin screw extruder in general there can be no pressure transmission back through the solid bed. In particular when working with polymer granules not much smaller than the size of the gaps, the only pressure feedback can be provided by the leakage flows. It will be shown later that in a well designed process, these leakage flows are very small in the entrance region, especially if the entrance region is cooled. There should be no influence from the die pressure on the filling process therefore.

Since the mass balance must be satisfied in a stationary process, the throughput must be constant and independent of die pressure. The output of the extruder is as indicated in formula VI.2 and dependent only on the geometry of the entrance zone, the rotational speed and the filling characteristics of the hopper. This means also that when temperature effects are neglected, the die pressure that is developed depends only on the rheological properties of the polymer, the geometry of the entrance region, the screw speed and the die geometry. This die inlet pressure is clearly independent of the screw geometry between the entrance region and the die.

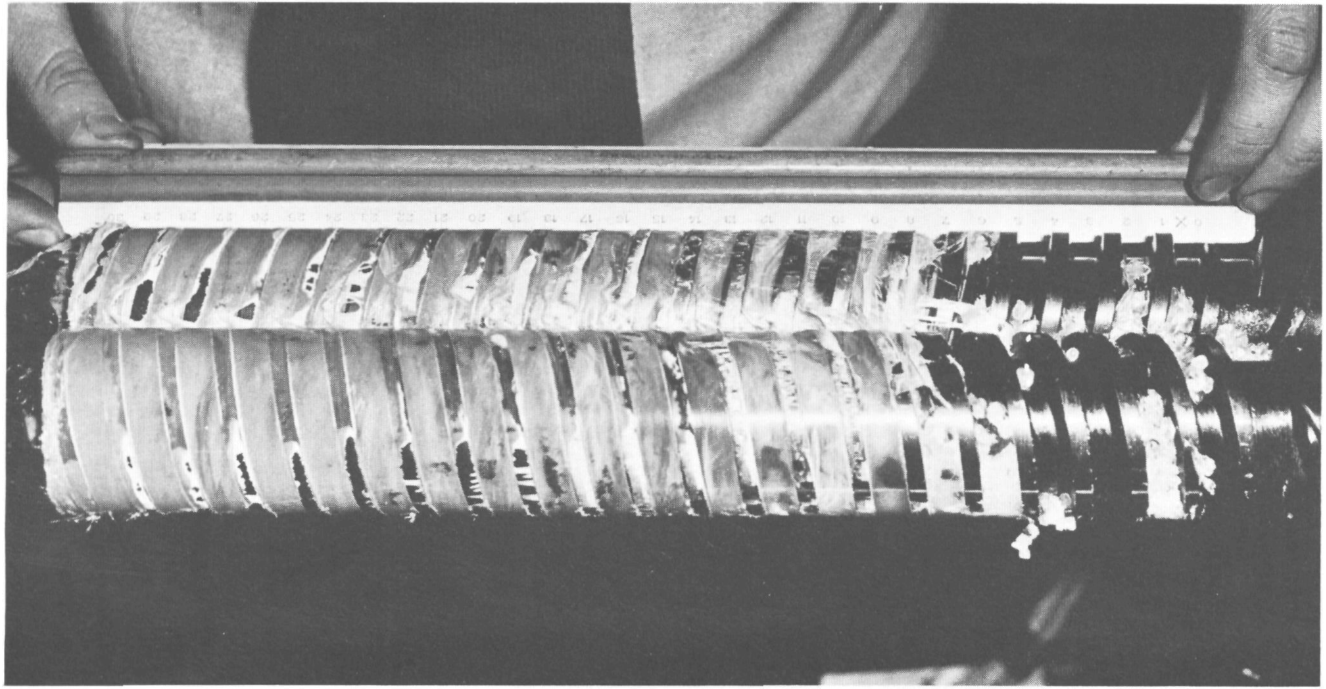

Fig. VI.1

Melt distribution in the screws after cooling the extruder and removing the barrel wall.

If a twin screw extruder is cooled and the barrel then removed, it is seen that only the chambers at the die end of the extruder are fully filled while the rest are partially empty (fig. VI.1). The positive conveying action of the fully filled chambers is as stated before

$$Q_{th} = 2 \text{ N m V} \qquad \text{(VI.3)}$$

This means that while the output of the extruder is as given by formula VI.2, the total amount of leakage over a cross section of the extruder can be given by

$$Q_{\ell} = 2 \text{ N m V } (1 - \mu u) \qquad \text{(VI.4)}$$

Surprisingly this back leakage is independent of die pressure and must be constant over the entire fully filled zone of the extruder if the screws in this zone have a constant profile.

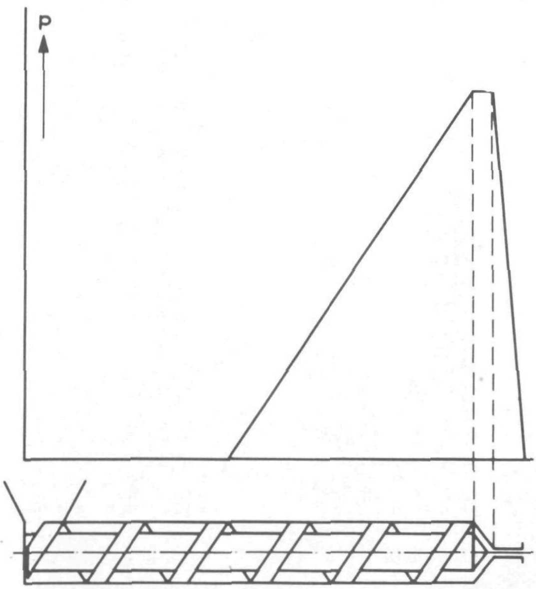

Fig. VI.2

Generalised pressure gradient in the extruder when working with isoviscous liquid and uniform screw profile.

When working with isoviscous fluids, the pressure gradient in this zone must therefore be constant as indicated in figure VI.2.

With this fixed pressure gradient present, there must be a point in the extruder where the over-pressure substantially falls to zero. Between this point and the die the chambers are completely filled, whereas from the hopper up to this point the chambers are only partially filled while there is no pressure gradient. In this region the leakage flows are therefore rather small.

It can be concluded that die pressure does not usually influence the feeding process, especially since polymer is usually prevented from melting by the use of a cooled entrance region.

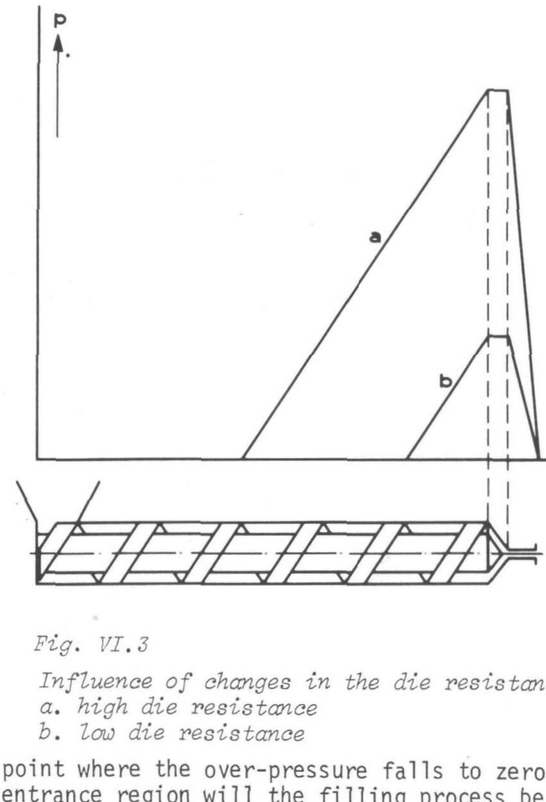

Fig. VI.3

Influence of changes in the die resistance
a. high die resistance
b. low die resistance

Only if the point where the over-pressure falls to zero lies as far back as the entrance region will the filling process be influenced and the output become dependent on the die pressure. Even if the die pressure changes, for example because of the use of another die, the throughput and the internal pressure gradient remains constant. Only the fully filled length in the extruder changes. This is indicated in figure VI.3. Two pressure lines for two different dies are given. When working with a small die, a large pressure is generated and the extruder is rather full (a). A large die will generate a low pressure and the extruder will remain relatively empty (b).

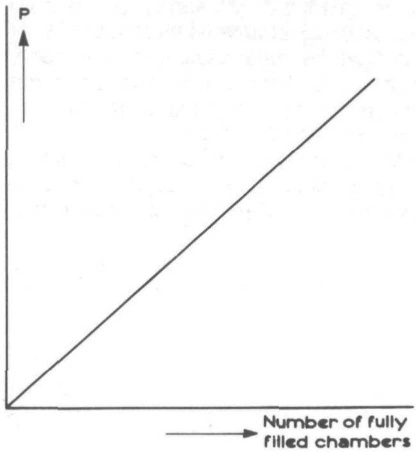

Fig. VI.4

Relation between the number of fully filled chambers and the die resistance when working with isoviscous liquid and a uniform screw profile.

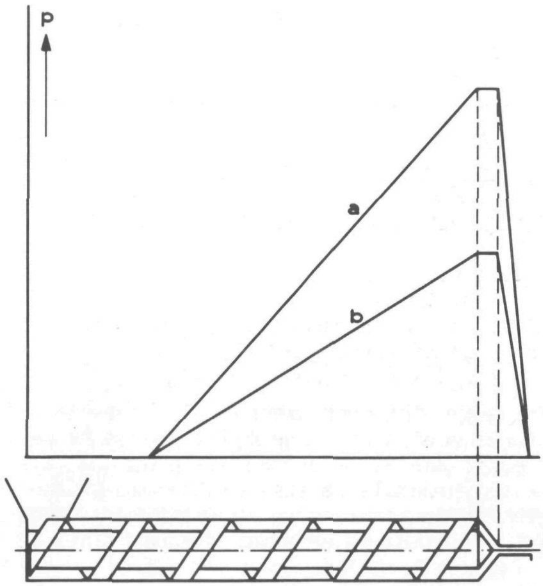

Fig. VI.5

Influence of changes in the screw speed.
a. high screw speed.
b. low screw speed.

In the Newtonian case the relationship between the die pressure and the number of fully filled chambers has the general form, shown in figure VI.4. It can also be concluded that in an ideal case the length of the extruder that is fully filled is independent of rotational speed if the filling process at the hopper is not affected. If the rotational speed is doubled the throughput will double too, according to formula VI.2, and under laminar flow conditions the die pressure generation will also become twice its original value. On the other hand the leakage flows will also become twice as large according to formula VI.4, which will in turn double the pressure gradient. For this reason the filled length does not change even if screw speed does, when working with a given die (figure VI.5).

VI.2 EXTENSION OF THE MODEL TO REAL SITUATIONS

The model presented in the previous paragraph deals with an isoviscous fluid and assumed the complete absence of solid material. When dealing with real polymers in a real extruder some modifications have to be made.

- The temperature of the polymer is not the same throughout the extruder. At the die end the temperature is higher than it is near the feed zone. The viscosity, and therefore the pressure gradient, is lowest at the die end of the extruder.
- The polymer can degrade in the extruder. This also leads to lower viscosity and a reduced pressure gradient at the die end.
- At high die pressures the fully filled zone can be longer than the zone in which the polymer is completely molten.
  In this case solid particles may partially block the gaps which will result in a greater pressure gradient near the feed end of the extruder.

All these corrections lead to a lower pressure gradient near the die than near the feed end of the extruder. Therefore the diagram of die pressure against the number of fully filled chambers will not be the straight line of figure VI.4, but look more like figure VI.6.

VI.3 MEASUREMENTS

To check the theories, measurements have been made with a Pasquetti laboratory twin screw extruder. The screw diameter is 47.7 mm. and the effective screw length is 360 mm. There is a 1-D feed section with four thread starts while the rest of the screw has a uniform profile with two thread starts. The compression ratio between the feed zone and the rest of the extruder screws is 1.5. The polymer used was granular polypropylene (Carlona P. from Shell). To investigate the stiffness of the extruder, i.e. the insensitivity of the throughput to the die pressure, various dies and screw speeds were used. The barrel temperatures were also varied, resulting in melt outlet temperatures between 200 and 230°C. Die pressures from 3 to 160 atmospheres could be achieved. The results are given in figure VI.7.

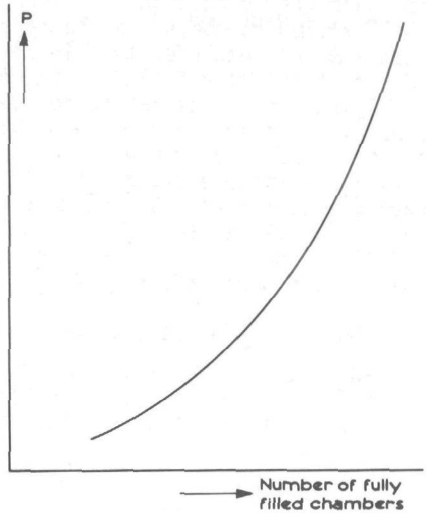

Fig. VI.6

Relation between the number of fully filled chambers and the die resistance when working with real polymers.

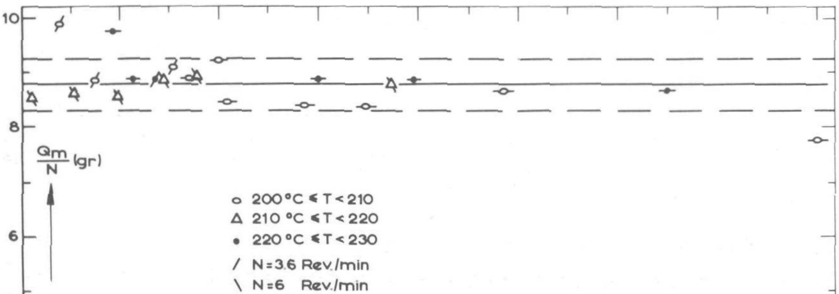

Fig. VI.7

Relation of die pressure and mass throughput divided by rotation rate for extrusion of polypropylene.

The abscissa shows the die pressure, the ordinate shows specific throughput i.e. throughput divided by rotation rate. It can be seen that the specific throughput is constant over the whole range of output pressures with no significant influence of temperature or rotational speed. The spread in these measurements is relatively small (about 5%).

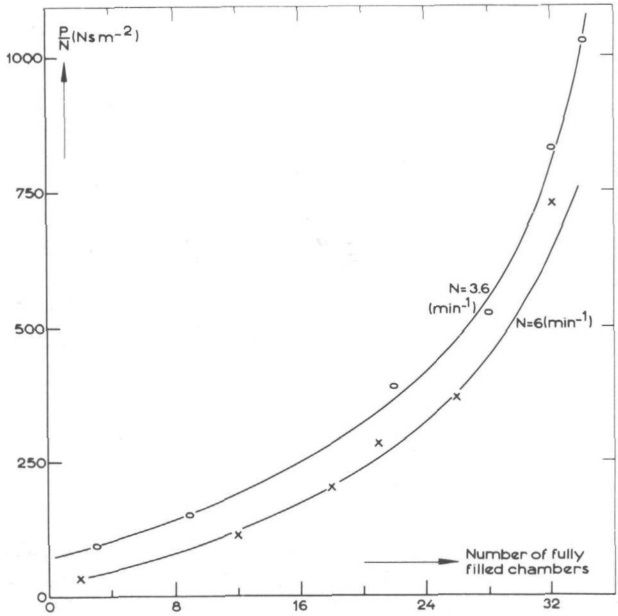

Fig. VI.8

Relation between the number of fully filled chambers and die pressure divided by rotation rate for extrusion of polypropylene at different screw speeds.

The number of completely filled chambers in the extruder was found by stopping the machine when operating under steady conditions cooling and then withdrawing the barrel. Figure VI.8 shows the results as a function of die pressure divided by screw speed. Two different screw speeds were used. The barrel temperatures from hopper to die were 172, 195, 195 and 200°C for each screw speed. The two curves lie close to each other, as predicted by the theory. The difference between the curves can be explained by temperature differences. At 3.6 revolutions per minute the polymer outlet temperatures at the die lay between 208 and 211.5°C. At 6 revolutions per minute these temperatures were between 213 and 219°C.

When working with a high rotational speed the viscosity is lower than when working with lower speeds. The pressure drop per chamber will be lower with given leakage flows and the completely filled length is longer at high rotational speeds.

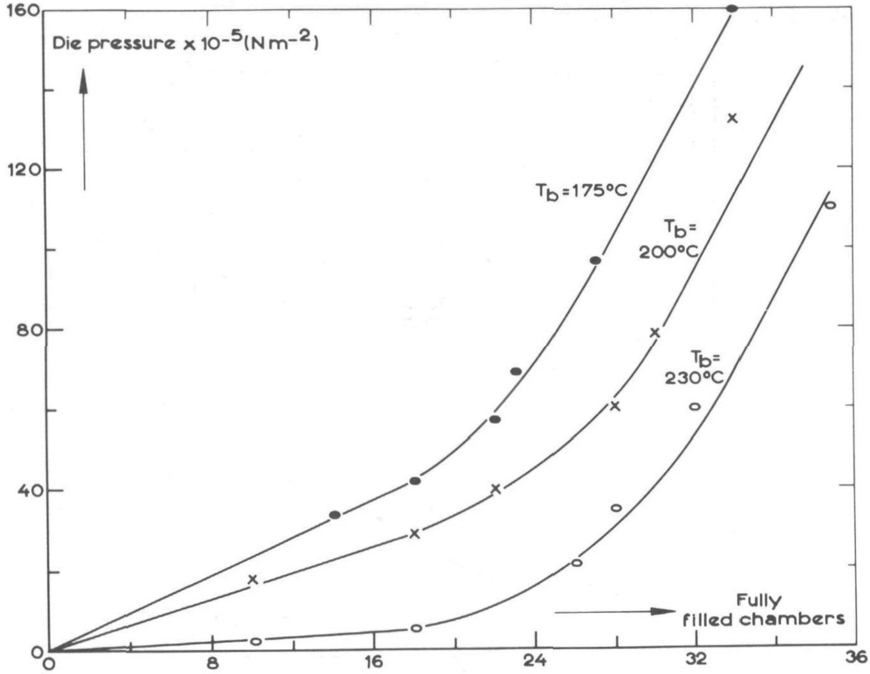

Fig. VI.9

Relation between the number of fully filled chambers and die pressure for extrusion of polypropylene at different barrel temperatures.

Figure VI.9 shows the influence of temperature. In these experiments the barrel temperatures were 175, 200 and 230°C. respectively and were constant along the whole length. The corresponding polymer temperatures at the die were 206, 222 and 252°C when the screws were operating at 9.8 r.p.m. Again the polymer at the lowest temperature has the shortest filled length for a given die pressure. When dealing with low die pressures only the last few chambers are fully filled. The curves tend to be straight and to go through the origin. This implies that the viscosity, and therefore temperature, is fairly uniform near the die end of the extruder. When there is a long fully filled zone in the extruder the curves again tend to straighten and become almost parallel.

This could be interpreted as meaning that in the first liquid filled chambers the contribution of each chamber to the pressure build-up is more or less independent of the barrel temperature, which implies a similar independence of the fluid viscosity in that region, i.e. that the fresh melt is always at approximately the same bulk temperature which is independent of the barrel temperature.

CHAPTER VII

THE MELTING MECHANISM

VII.1 INTRODUCTION

The melting mechanism in single screw extruders has been studied for a long time. Little is known however of the way polymer melts in twin screw extruders. In single screw extrusion the solid polymer bed melts as a result of both mechanical working and heat penetration from the barrel wall. Most investigations(e.g. VII.1, VII.2) indicate that a melt pool can be found at the pushing side of the screw flight (fig. VII.1). This implies that the molten polymer is wiped from the barrel wall by the moving flight and collected in the melt pool. On the other hand some articles report a melt pool on the other side of the channel (fig. VII.2) (VII.3). Recent investigations on large single screw extruders (VII.4) even report an absence of such a melt pool. In this case the solid bed was apparently located in the middel of the chamber while a layer of molten polymer was between this bed and the surfaces of the screw and barrel (fig. VII.3). In all cases the solid bed steadily diminishes along the length of the screw. This melting length forms a considerable part of the extruder.

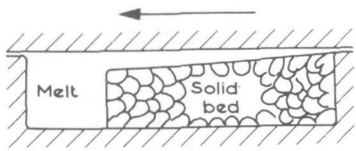

Fig. VII.1

Melt pool according to Maddock and Vermeulen e.a.

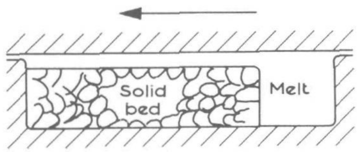

Fig. VII.2

Melt pool according to Menges and Klenk.

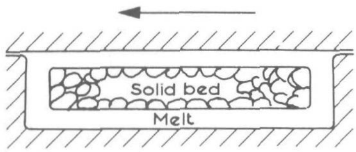

Fig. VII.3

Melt film according to Lindt.

VII.2 MELTING MECHANISM IN A TWIN SCREW EXTRUDER.

To investigate the mechanism that controls the melting of polymer in twin screw extruders, experiments have been done with polypropylene, both granules and powder. The extruder used was that described in the previous chapter. It was operated until the temperature of the melt at the die inlet was constant. Then the extruder was stopped, cooled, and the barrel removed. The polymer could be extracted from individual chambers and examined. There were great differences between the experiments with granules and powders. However, in all the experiments, the melting length was very short compared with similar situations in single screw extruders (e.g. VII.5).

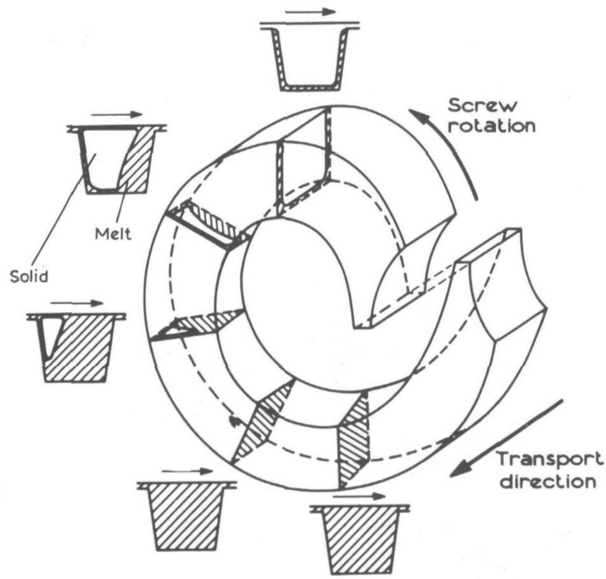

Fig. VII.4
Situation of the melt within one C shaped chamber when working with polypropylene granules (chamber 11).

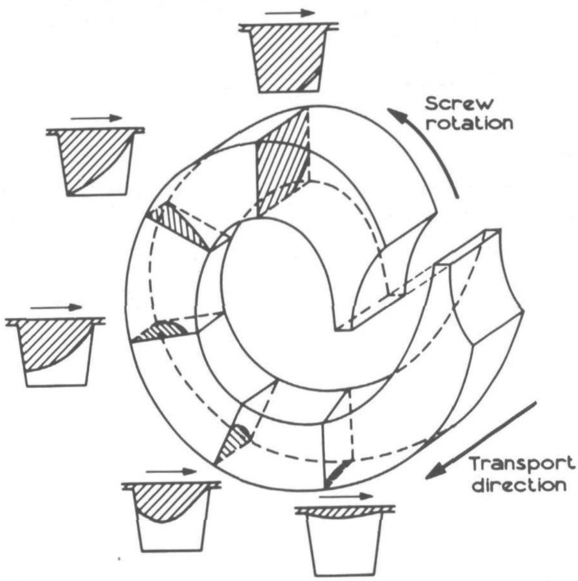

Fig. VII.5
Situation of the melt within one C shaped chember when working with polypropylene powder and low die pressure (chamber 13).

The transition from solid to melt was completed within 4 or 5 chambers.

In figures VII.4, VII.5, and VII.7 a C shaped chamber is drawn. In these chambers the position of the melt at several places is indicated by hatching. Cross sections are drawn separately, always with the barrel side up. The arrows above the cross sections indicate the direction of the axial wall velocity relative to the chamber.

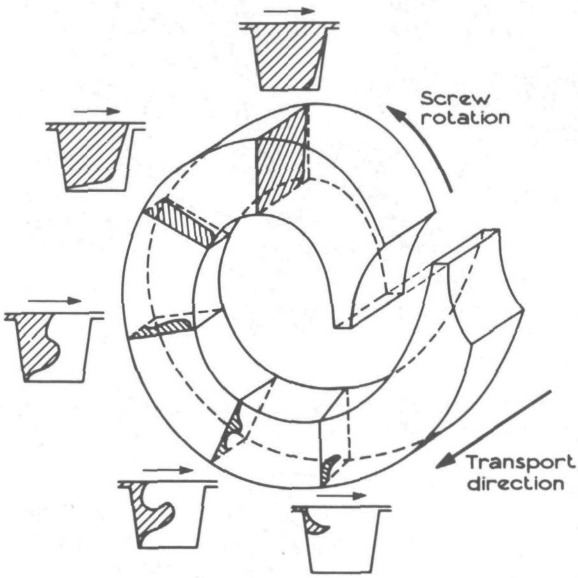

Fig. VII.7

Situation of the melt within one C shaped chamber when working with polypropylene powder and high die pressure (chamber 7).

Figure VII.4 shows the situation when working with granules. In this case the melt is found at the lower side of the chamber. In counter rotating twin screw extruders the rotation is such that the screws converge at this lower side of the channel. At the top of the channel some unmolten polymer and empty space was found. Only a thin layer of molten polymer that has leaked back through the calender gap and the side gaps is present on the screw surfaces. Along the screw channel between these regions the melting clearly spreads from the downstream end of the chamber to the leading end. The melt is generally found against the pushing flight.

When working with powder the situation is quite different. The melt was found mainly in the leading end of the chambers while the unmolten powder could be found mainly at the other end.

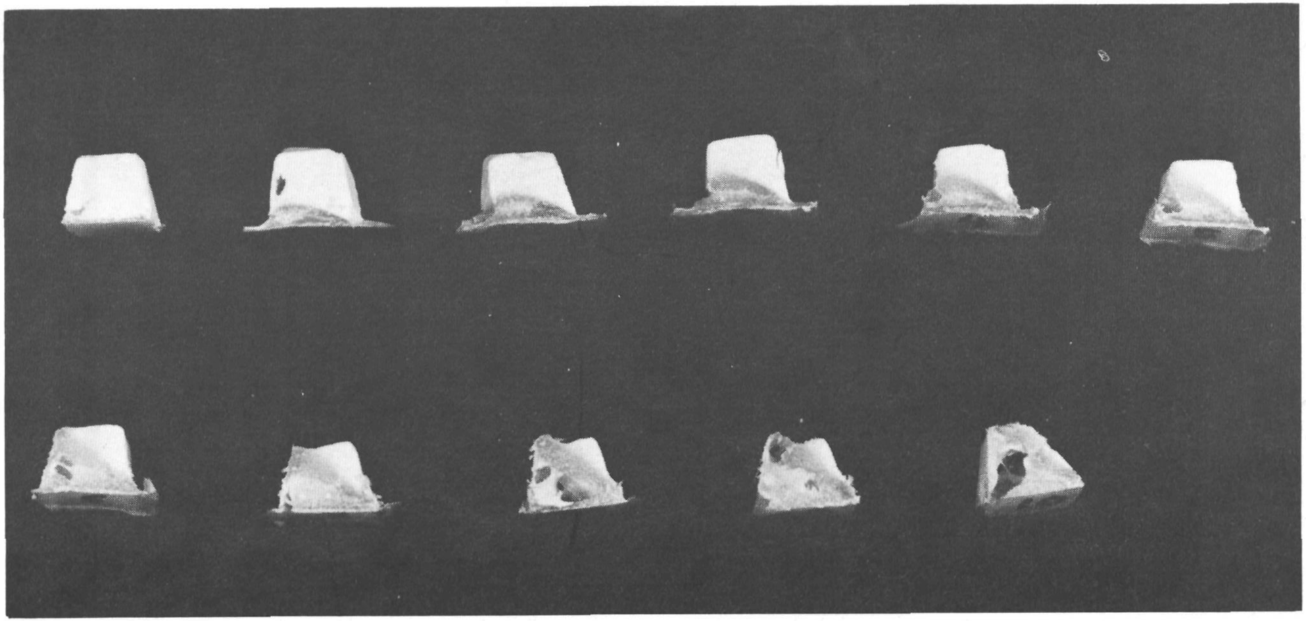

Fig. VII.6

Melting process, samples taken from the 13th chamber (compare with figure VII.5).

Unlike granules, which did not form a strong solid bed although deformed to a great extent, the powder was compressed into a dense coherent bed. A further distinction can also be drawn between the powder melting processes at high or low die pressures. In both cases the melting process is illustrated by a drawing of conditions in that particular C shaped chamber where at the top a cross section was present where no solid was left and at the bottom side a cross section could be found without melt.

Figures VII.5 and VII.6 indicate the place of the melt with a low die pressure (30×10^5 N/m^2). It can be seen that the melting at the downstream end of the chamber only takes place in a thin layer along the heated barrel wall and that the melt is collected behind the pulling side of the screw flight. All these samples were taken from the 13th chamber from the filling zone.

Figure VII.7 indicates the situation when working with a high die pressure (140×10^5 N/m^2). In this case the samples were taken from the 7th chamber from the filling zone while all variables except the die geometry were similar to those in the previous experiment. In this case the melting process is not induced by the heated barrel wall but seems to originate from the leakages through the gaps. Going from downstream to upstream in the chamber a penetration of the melt takes place. The shape of the melt boundary changes from an S shape near the pulling side of the screw flight to a thin straight layer of unmolten powder near the pushing side of the screw flight. It can clearly be seen that when applying high back pressures the heat penetration from the barrel wall is less important for the melting process than when only a low die pressure is used.

This chapter has indicated that the melting models commonly adopted for single screw extruders are not directly applicable to the different situations found in twin screw machines dependent on whether powder or granules are being processed. It is however clear that to establish a consistent melting model much more work has to be done (VII.6).

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CHAPTER VIII

CONCLUSIONS

VIII.1 THE FULLY FILLED PART OF THE EXTRUDER

It is clear that twin screw extrusion technology is still more of an art than a science. Although the development of these machines started nearly fifty years ago, little quantitative research into the phenomenology of the process has been done. Twin screw extruders with intermeshing screws are geometrically complicated. Unlike the channel in a single screw extruder, the screw channels are continuously interrupted by the flights of the other screw. Analytical calculations and computations based on a continuous channel are therefore erroneous. A better approach is the use of the physical model suggested in this thesis. This model is based on the fact that in a twin screw extruder C shaped chambers exist which move from hopper to die as the screws rotate. The interactions between these chambers within the extruder are due to leakage flows through the gaps. These gaps are always present, not only for reasons of shear and wear, but also because without them the extruder screws would not fit into each other.

The length of the fully filled zone in a twin screw extruder changes with die pressure while ceterus paribus the leakage flows remain constant. However, it helps the understanding of the process to consider the length of the fully filled zone to be constant while the leakage flows vary with die pressure.

The output from a fully filled pump zone of a twin screw extruder can be obtained from the theoretical positive displacement by subtraction of the various leakage flows. When the processing fluid is assumed to be Newtonian and isoviscous, the output pressure characteristics can be calculated. If this output to pressure relationship is presented in the dimensionless form $Q/2NmV$ and $\Delta P/N\eta$, it is completely independent of operating conditions (i.e. driving speed and viscosity). These dimensionless numbers also provide a useful basis for geometrical scale up. The output to pressure relationship is extremely dependent on details of the geometry of the screw. This explains the fact that in industry two "identical" twin screw extruders can have differences in performance. When using non-Newtonian liquids the performance also changes. A pseudoplastic process fluid has a lower throughput for the same pressure gradient in the extruder. This is obvious, since the shear rates in the leakage gaps can be very high. These high shear rates give low apparent viscosities in the gaps while the shear rates in the C shaped chambers are in general much lower. Therefore the internal pressure generation within the chambers themselves is not affected very much by the non-Newtonian properties of the liquid. It can be concluded that with a pseudoplastic process fluid instead of a Newtonian one the leakage flows will be larger and, if the same pressure gradient in the extruder is developed as in the Newtonian case, the net throughput will be less.

VIII.2 FLOW, MIXING AND RESIDENCE TIMES

Many quantitative articles report that twin screw extruders provide a better homogenisation on micro scale and assert that there is a narrower spread of residence time distributions than in single screw extruders. This statement, however frequently repeated, cannot be universally valid since, as has been pointed out, the dimensionless exit age distribution is not markedly different either from that for a single screw extruder or even that associated with laminar Poiseulle flow in a pipe. Also the mixing within the chambers is not to be overestimated. Both the work as described in this thesis and the work published by Kim et al confirm this. Whereas the papers of Kim et al deal mainly with the flow within the chambers the investigations described in chapter V of this thesis refer also to the interaction between the chambers, i.e. what happens with the fluid that leaks back from one chamber to another and how the mixing within the chamber is influenced by changes in the system. We have found that when the calender gap is small a closed and stable flow occurs at the bottom of the chamber that mixes very poorly with the bulk of the fluid in the rest of the chamber. Kim et al report that the mixing, described in terms of a variational coëfficient, is worse in the middle of the C shaped chamber than near the intermeshing zones. It is presumed that this is due to the unmixed zone at the bottom of the channel. Flow visualization experiments by means of colour injection into a transparent model extruder have shown that the tetrahedron leakage and the calender and side leakages interact with each other. This permits the leakage fluid in certain circumstances to bypass chambers completely. The flight leakage stands on its own. As in single screw extruder technology, when the flight gap is large enough, a liquid layer can form at the barrel wall which does not mix with the bulk of the liquid.

As far as the residence time distribution is concerned, it has been found that the usual response of output concentration to a step change in input concentration, when plotted with log-linear coordinates, shows a non-linearity which implies a limited amount of mixing within the extruder. Only when using a wide calender gap is this kink not significant. This implies that when using a counter rotating twin screw extruder for chemical reactions, where good homogenisation as well as a uniform residence time are important, the screws must not intermesh too closely.

VIII.3 RELATION BETWEEN THE MODELS AND A REAL TWIN SCREW EXTRUDER

A real twin screw extruder can be divided into four zones. Underneath the hopper a feed zone is generally present which is about one screw diameter long. In this zone the pitch of the screws is usually greater than the pitch in the rest of the extruder and often more thread starts are present. Because of the greater conveying property of the filling zone the polymer is compressed when going from the feed zone to the next zone, which is called the melting zone.

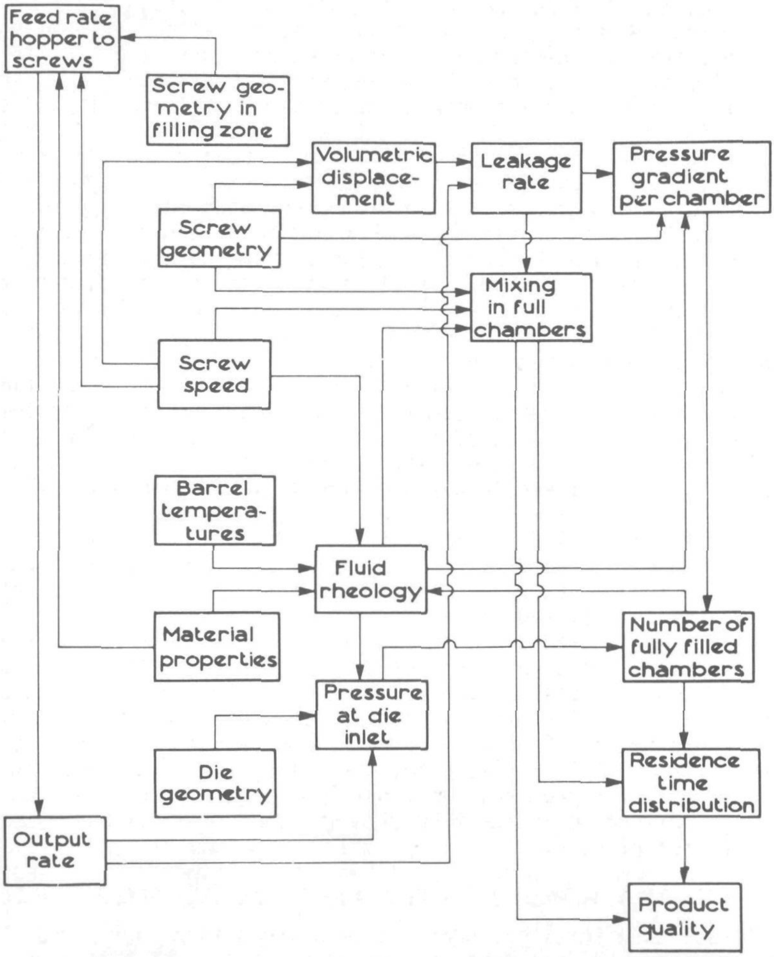

Fig. VIII.1

System diagram for the steady state in a polymer twin screw extruder.

In this region the polymer melts while the chambers are still partially empty. Another zone can be defined within which all the polymer is molten while the chambers are still partially empty. In this zone the leakage flows are relatively small compared to those in the fully filled region because the gaps are only partially used, the internal pressure generation is small and no back pressure gradient due to die pressure is present. We call this zone the conveying zone. In the last zone the chambers are fully filled and in this zone the leakages are as described in the first part of this thesis.

In a stationary process the output is completely determined by the input. In twin screw extruders in general the input is not affected by die pressure. Therefore the output is, at least when using polymer granules, independent of die pressure. As the positive conveying action of the screws is only determined by the geometry and the rotation rate of the screws, the leakage flows are also independent of die pressure, but determine the pressure gradient along this part of the extruder. The length of the fully filled zone changes with die pressure, and this may result in a conveying zone not being present.

VIII.4 DISCUSSION AND RECOMMENDATIONS

The interactions between the phenomena as they exist in a twin screw extruder can be summarised in a diagram representing the steady state (figure VIII.1). The most obvious feature is the connection between the feed rate from hopper to screw and the output rate. These two must indeed be the same. The feed rate is uniquely established by the screw geometry in the filling zone, the screw speed and such material properties as particle shape. The screw geometry in the rest of the extruder together with the screw speed determine the volumetric displacement, which in its turn controls, together with the output rate, the leakage rate. Because of degradation of the polymer and local changes in polymer temperature the fluid rheology is, apart from the material properties, also influenced by barrel temperature, screw speed and the number of fully filled chambers. This fluid rheology, the die geometry and the output rate determine the pressure at the die inlet. With a given screw geometry and a given fluid rheology the pressure gradient per chamber is completely determined by the leakage rate. From this pressure gradient together with the pressure at the die inlet the number of fully filled chambers is determined by the point in the extruder where the pressure falls substantially to zero. Since whether the chambers are fully filled or not affects the leakage flows, the number of fully filled chambers together with the mixing within one chamber gives an estimation of the residence time distribution. This residence time distribution is important because it can be assumed to determine amongst other things the product quality, and in particular, the homogeneity of the end product.

One important block remains vaque; the degree of mixing in full chambers. This will be determined by the leakage rate, the screw speed, the screw geometry and the fluid rheology.

The only definite statement one can make is that this mixing is, in contrast to what is frequently stated in literature, far from ideal as can be seen from the non linearity in the exit age distribution. The mixing becomes better when using a fairly wide calender gap. However when the calender gap becomes too wide the twin screw extruder acts as two parallel single screw extruders and all the advantages of the positive conveying action are lost. As the mixing in the fully filled chambers affects the product quality directly, as well as via the residence time distribution, it might be an interesting topic for future research.

The absence of accurate measurements of temperature profiles in twin screw extruders is also a notable gap in present knowledge. Such data could lead to a model for the heat transfer process, which is completely lacking at the moment. An understanding of the heat transfer process can also provide a lead for the application of twin screw extruders to chemical reaction engineering technology where there appear to be attractive fields for exploitation.

The point must also be made that almost without exception the effects of polymer rheology have been neglected. In view of the unsteady recirculating nature of the flow through the leakage gaps it may well be that elastic effects as well as shear dependent viscosities should not be ignored. In general the greatest difficulty that arises in the analysis of the process is the complicated geometry, especially when considering non-Newtonian effects. Since an extruder with closely intermeshing twin screws has no continuous channel, this implies at once that the analytical treatment of the pseudo fully developed flow and temperature fields of the single screw machine have no counterpart when twin screws are considered. It appears inevitable that the description of the conditions in the more complex geometry will have to be based on a semi-empirical model of the extruder as a chain of seperate chambers with the leakage flows providing the interaction between them. The suggestion that, particularly for the near future, an experimental basis for the analysis is likely to be the most productive is an admission that the author believes that analogue experiments using real fluids to solve the Navier-Stokes equations in such complicated geometries as in twin screw extruders are still likely to be both cheaper and more valuable than the alternative necessarily three dimensional computer simulations.

LIST OF SYMBOLS

b	axial width of the flight	m
B	axial width of the flight at outher radius	m
c	concentration	-
C	concentration	-
d	correction factor	-
D	screw diameter	m
E	axial distance between the screw flanks	m
F	variational coefficient	-
h	distance of the centre line to screw surfaces in the calender gap	m
H	chamber height	m
i	counting variable	-
I	modified Bessel function of the first kind	-
J	correction factor	m^3/s
k	consistency	Ns^n/m^2
K	modified Bessel function of the second kind	-
l	length of the overlapping area	m
m	number of thread starts	-
M	number of samples	-
n	power law index	-
N	rotation rate	s^{-1}
p	pressure	N/m^2
P	die pressure	N/m^2
q	volumetric flow rate	m^3/s
Q	volumetric flow rate	m^3/s
r	radius	m
R	outer radius of the screw	m
s	number of screws with single thread start	-
S	pitch	m
t	time	s
T	temperature	^oC
u	filling degree	-
v	velocity	m/s

V	chamber volume	m^3
w	axial wall velocity	m/s
x	coordinate	m
y	coordinate	m
z	coordinate	m
α	angle of the overlapping area	rad
β	coordinate angle	rad
γ	coordinate	m
δ	flight gap width	m
Δ	difference	-
ε	side gap width	m
η	viscosity	Ns/m^2
Θ	time constant	s
μ	compression factor	-
ξ	dimensionless variable	-
σ	calender gap width	m
τ	shear stress	N/m^2
Φ	screw angle	rad
Ψ	flight wall angle	rad
ΔP	pressure drop between two consecutive chambers due to die pressure	N/m^2

subscripts

a	apparent
b	barrel
c	calender gap
f	flight gap
i	counting variable
ℓ	total leakage
l	left screw
o	initial value
pr	practical
q	concerning flow rate q
Q	concerning flow rate Q

r	right screw
s	side gap
t	tetrahedron gap
th	theoretical
x	x direction
y	y direction
z	z direction
γ	γ direction (tangential)
ϕ	feed zone
1,2,3	

super scripts

$\overline{}$	average value
'	per unit length

STELLINGEN

1. De benadering van een dubbelschroefextruder door twee series afzonderlijke kamers, die de polymeer van de vultrechter naar de spuitopening transporteren en waarbij de lekstromen zorgen voor de interactie tussen deze kamers, biedt de meeste kans voor het opstellen van een handzaam model.

   Dit proefschrift.

2. In tegenstelling tot wat vaak in de literatuur wordt beweerd, kan niet zonder meer worden aangenomen, dat een dubbelschroefextruder een kleinere spreiding in de verblijftijd heeft, dan een enkelschroefextruder.

   Dit proefschrift.

3. Warmteoverdrachtmodellen in enkelschroefextruders, die gebaseerd zijn op het zogenaamde Jepson-effect, dienen met grote voorzichtigheid gehanteerd te worden.

   Jepson.C.H., Ind. Eng. Chem. <u>45</u> *(1953) 992.*
   Janssen.L.P.B.M.,Noomen.G.H.,Smith.J.M., Plastics and Polymers. <u>43</u> *(1975) 135.*

4. De temperatuurverschillen in de smelt in enkelschroefextruders zijn veel kleiner in tangentieële richting dan in radieële richting.

   Schläffer.W.,Schijf.J.,Janeschitz-Kriegl.H., Plastics and Polymers. <u>39</u> *(1971) 193.*
   Janssen.L.P.B.M.,Noomen.G.H.,Smith.J.M., Ibid. <u>43</u> *(1975) 135.*

5. Dat in pseudo-wetenschappelijke kwalitatieve artikelen over dubbelschroefextruders vaak veel meer wordt beloofd dan in kwantitatieve artikelen kan worden waargemaakt, kan geweten worden aan het feit, dat bij te veel auteurs de verkoopresultaten van hun broodheren prevaleren boven een kritische wetenschappelijke onbevangenheid.

6. Het zou aan te bevelen zijn, dat de inspecteur der belastingen voor het waarderen van afschrijvingen op investeringen in plaats van het historische kostprijsbeginsel het vervangingswaardestelsel zou hanteren.

   Slot.R.,Kostenberekening en Prijspolitiek, Leiden (1972).

7. Als in de economische en sociale wetenschappen aandacht zou worden besteed aan de thermodynamica, zou in deze wetenschappen het entropiebegrip zeer verhelderend kunnen werken.

8. Gezien het werkterrein van veel natuurkundig ingenieurs verdient het aanbeveling in de studie meer aandacht te besteden aan eenheidswerkwijzen.

9. In verband met het grote verschil in brekingsindices van lucht en water verdient het aanbeveling om bij waterpolowedstrijden twee scheidsrechters in plaats van één aan te stellen.

10. Dat enkele grote kruideniersbedrijven papieren zakken trachten te slijten door te appeleren aan het millieubewustzijn van de kopers versterkt de volledig foutieve algemene opinie, dat papier minder schadelijk voor het millieu zou zijn dan plastic.