The roller die (B.F. Goodrich, 1933) is a combination of a standard sheet die and a calender. It allows high throughput by reducing the diehead pressure; it reduces air entrapment and provides good gauge control.

2.2 The Multiscrew Extruder

2.2.1 The Twin Screw Extruder

A twin screw extruder is a machine with two Archimedean screws. Admittedly, this is a very general definition. However, as soon as the definition is made more specific, it is limited to a specific class of twin screw extruders. There is a tremendous variety of twin screw extruders, with vast differences in design, principle of operation, and field of application. It is, therefore, difficult to make general comments about twin screw extruders. The differences between the various twin screw extruders are much larger than the differences between single screw extruders. This is to be expected, since the twin screw construction substantially increases the number of design variables, such as direction of rotation, degree of intermeshing, etc. A classification of twin screw extruders is shown in Table 2.2. This classification is primarily based on the geometrical configuration of the twin screw extruder. Some twin screw extruders function in much the same fashion as single screw extruders. Other twin screw extruders operate quite differently from single screw extruders and are used in very different applications. The design of the various twin screw extruders with their operational and functional aspects will be covered in more detail in Chapter 10.

<table>
<thead>
<tr>
<th>Intermeshing extruders</th>
<th>Co-rotating extruders</th>
<th>Low speed extruders for profile extrusion</th>
<th>High speed extruders for compounding</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Conical extruders</td>
<td>Conical extruders for profile extrusion</td>
<td>Conical extruders for profile extrusion</td>
</tr>
<tr>
<td></td>
<td>Parallel extruders</td>
<td>Parallel extruders for profile extrusion</td>
<td>Parallel extruders for profile extrusion</td>
</tr>
<tr>
<td></td>
<td>High speed extruders</td>
<td>High speed extruders for compounding</td>
<td>High speed extruders for compounding</td>
</tr>
</tbody>
</table>

| Non-intermeshing extruders | Counter-rotating extruders | Equal screw length |
|                           |                         | Unequal screw length |
|                           | Co-rotating extruders   | Not used in practice |
|                           | Co-axial extruders      | Inner melt transport forward |
|                           |                         | Inner melt transport rearward |
|                           |                         | Inner solids transport rearward |
|                           |                         | Inner plasticating with rearward transport |
2.2.2 The Multiscrew Extruder With More Than Two Screws

There are several types of extruders, which incorporate more than two screws. One relatively well-known example is the planetary roller extruder, see Fig. 2.10.

This extruder looks similar to a single screw extruder. The feed section is, in fact, the same as on a standard single screw extruder. However, the mixing section of the extruder looks considerably different. In the planetary roller section of the extruder, six or more planetary screws, evenly spaced, revolve around the circumference of the main screw. In the planetary screw section, the main screw is referred to as the sun screw. The planetary screws intermesh with the sun screw and the barrel. The planetary barrel section, therefore, must have helical grooves corresponding to the helical flights on the planetary screws. This planetary barrel section is generally a separate barrel section with a flange-type connection to the feed barrel section.

In the first part of the machine, before the planetary screws, the material moves forward as in a regular single screw extruder. As the material reaches the planetary section, being largely plasticized at this point, it is exposed to intensive mixing by the rolling action between the planetary screws, the sun screw, and the barrel. The helical design of the barrel, sun screw, and planetary screws result in a large surface area relative to the barrel length. The small clearance between the planetary screws and the mating surfaces, about $\frac{1}{4}$ mm, allows thin layers of compound to be exposed to large surface areas, resulting in effective devolatilization, heat exchange, and temperature control. Thus, heat-sensitive compounds can be processed with a minimum of degradation. For this reason, the planetary gear extruder is frequently used for extrusion/compounding of PVC formulations, both rigid and plasticized [21,22]. Planetary roller sections are also used as add-ons to regular extruders to improve mixing performance [97,98]. Another multiscrew extruder is the four-screw extruder, shown in Fig. 2.11.
This machine is used primarily for devolatilization of solvents from 40% to as low as 0.3% [23]. Flash devolatilization occurs in a flash dome attached to the barrel. The polymer solution is delivered under pressure and at temperatures above the boiling point of the solvent. The solution is then expanded through a nozzle into the flash dome. The foamy material resulting from the flash devolatilization is then transported away by the four screws. In many cases, downstream vent sections will be incorporated to further reduce the solvent level.

2.2.3 The Gear Pump Extruder

Gear pumps are used in some extrusion operations at the end of a plasticating extruder, either single screw or twin screw [99–106]. Strictly speaking, the gear pump is a closely intermeshing counterrotating twin screw extruder. However, since gear pumps are solely used to generate pressure, they are generally not referred to as an extruder although the gear pump is an extruder. One of the main advantages of the gear pump is its good pressure-generating capability and its ability to maintain a relatively constant outlet pressure even if the inlet pressure fluctuates considerably. Some fluctuation in the outlet pressure will result from the intermeshing of the gear teeth. This fluctuation can be reduced by a helical orientation of the gear teeth instead of an axial orientation.

Gear pumps are sometimes referred to as positive displacement devices. This is not completely correct because there must be mechanical clearances between the gears and the housing, which causes leakage. Therefore, the gear pump output is dependent on pressure, although the pressure sensitivity will generally be less than that of a single screw extruder. The actual pressure sensitivity will be determined by the design clearances, the polymer melt viscosity, and the rotational speed of the gears. A good method to obtain constant throughput is to maintain a constant pressure differential across the pump. This can be done by a relatively simple pressure feedback control on the extruder feeding into the gear pump [102]. The non-zero clearances in the gear pump will cause a transformation of mechanical energy into heat by viscous heat generation, see Section 5.3.4. Thus, the energy efficiency of actual gear pumps is considerably below 100%; the pumping efficiency generally ranges from 15 to 35%. The other 65 to 85% goes into mechanical losses and viscous heat generation. Mechanical losses usually range from about 20 to 40% and viscous heating from about 40 to 50%. As a result, the polymer melt going through the gear pump will experience a considerable temperature rise, typically 5 to 10 °C. However, in some cases the temperature rise can be as much as 20 to 30 °C. Since the gear pump has limited energy efficiency, the combination extruder-gear pump is not necessarily more energy-efficient than the extruder without the gear pump. Only if the extruder feeding into the gear pump is very inefficient in its pressure development will the addition of a gear pump allow a reduction in energy consumption. This could be the case with co-rotating twin screw extruders or single screw extruders with inefficient screw design.

The mixing capacity of gear pumps is very limited. This was clearly demonstrated by Kramer [106] by comparing melt temperature fluctuation before and after the gear pump, which showed no distinguishable improvement in melt temperature uniformity. Gear pumps are often added to extruders with unacceptable output fluctuations. In many cases, this constitutes treating the symptoms but not curing the actual problem. Most single screw extruders, if properly designed, can maintain their output to within ± 1%. If the output fluctuation is considerably larger than 1%, there is probably something wrong with the machine; very often incorrect screw design. In these cases, solving the actual problem will generally be more efficient than adding a gear pump. For an efficient extruder-gear pump system, the extruder screw has to be modified to reduce the pressure-generating capacity of the screw.
Gear pumps can be used advantageously:

1. On extruders with poor pressure-generating capability (e.g., co-rotating twin screws, multi-stage vented extruders, etc.)
2. When output stability is required better than 1%, i.e., in close tolerance extrusion (e.g., fiber spinning, cable extrusion, medical tubing, coextrusion, etc.)

Gear pumps can cause problems when:

1. The polymer contains abrasive components; because of the small clearances, the gear pump is very susceptible to wear.
2. When the polymer is susceptible to degradation; gear pumps are not self-cleaning and combined with the exposure to high temperatures this will result in degraded product.

2.3 Disk Extruders

There are a number of extruders, which do not utilize an Archimedean screw for transport of the material, but still fall in the class of continuous extruders. Sometimes these machines are referred to as screwless extruders. These machines employ some kind of disk or drum to extrude the material. One can classify the disk extruders according to their conveying mechanism (see Table 2.1). Most of the disk extruders are based on viscous drag transport. One special disk extruder utilizes the elasticity of polymer melts to convey the material and to develop the necessary diehead pressure.

Disk extruders have been around for a long time, at least since 1950. However, at this point in time the industrial significance of disk extruders is still relatively small compared to screw extruders.

2.3.1 Viscous Drag Disk Extruders

2.3.1.1 Stepped Disk Extruder

One of the first disk extruders was developed by Westover at Bell Telephone Laboratories: it is often referred to as a stepped disk extruder or slider pad extruder [24]. A schematic picture of the extruder is shown in Fig. 2.12.

The heart of the machine is the stepped disk positioned a small distance from a flat disk. When one of the disks is rotated with a polymer melt in the axial gap, a pressure build-up will occur at the transition of one gap size to another, smaller gap size, see Fig. 2.13.

If exit channels are incorporated into the stepped disk, the polymer can be extruded in a continuous fashion. The design of this extruder is based on Rayleigh’s [25] analysis of hydrodynamic lubrication in various geometries. He concluded that the parallel stepped pad was capable of supporting the greatest load. The stepped disk extruder has also been designed in a different configuration using a gradual change in gap size. This extruder has a wedge-shaped disk with a gradual increase in pressure with radial distance.

A practical disadvantage of the stepped disk extruder is the fact that the machine is difficult to clean because of the intricate design of the flow channels in the stepped disk.
2.3.1.2 Drum Extruder

Another rather old concept is the drum extruder. A schematic picture of a machine manufactured by Schmid & Kocher in Switzerland is shown in Fig. 2.14.

Material is fed by a feed hopper into an annular space between rotor and barrel. By the rotational motion of the rotor, the material is carried along the circumference of the barrel. Just before the material reaches the feed hopper, it encounters a wiper bar. This wiper bar scrapes the polymer from the rotor and deflects the polymer flow into a channel that leads to the extruder die. Several patents [26,27] were issued on this design; however, these patents have long since expired.

A very similar extruder (see Fig. 2.15) was developed by Askco Engineering and Cosden Oil & Chemical in a joint venture; later this became Permian Research. Two patents have been issued on this design [28,29], even though the concept is very similar to the Schmid & Kocher design.

One special feature of this design is the capability to adjust the local gap by means of a choker bar, similar to the gap adjustment in a flat sheet die, see Section 9.2. The choker bar in this drum extruder is activated by adjustable hydraulic oil pressure. Drum extruders have not been able to be a serious competition to the single screw extruder over the last 50 years.

2.3.1.3 Spiral Disk Extruder

The spiral disk extruder is another type of disk extruder that has been known for many years. Several patented designs were described by Schenkel (Chapter 1, [3]). Similar to the stepped disk extruder, the development of the spiral disk extruder is closely connected to spiral groove bearings. It has long been known that spiral groove bearings are capable of supporting substan-
tial loads. Ingen Housz [30] has analyzed the melt conveying in a spiral disk extruder with logarithmic grooves in the disk, based on Newtonian flow behavior of the polymer melt. In terms of melt conveying capability, the spiral disk extruder seems comparable to the screw extruder; however, the solids conveying capability is questionable.

2.3.1.4 Diskpack Extruder

Another development in disk extruders is the diskpack extruder. Tadmor originated the idea of the diskpack machine, which is covered under several patents [31–33]. The development of the machine was undertaken by the Farrel Machinery Group of Emgant Corporation in cooperation with Tadmor [34–39]. The basic concept of the machine is shown in Fig. 2.16.
optimum channel depth for power consumption when both the channel depth and the helix angle are optimized simultaneously. It can further be seen in Fig. 8.22 that the higher pressure gradient causes a substantially higher specific energy consumption. In the normal channel depth range, \( H = 0.01 \, D - 0.05 \, D \), the difference in SEC is about 50%. This will improve the mixing efficiency of the metering section, but it will also cause more viscous heat generation in the melt and may overheat the polymer.

The optimum channel depth \( H^* \) can be calculated from:

\[
\frac{\partial Z}{\partial H} \dot{V} = \frac{\partial \dot{V}}{\partial H} Z
\]

(8.75)

This results in another lengthy expression similar to Eq. 8.74. This expression is difficult to solve analytically and is generally solved numerically or graphically, as shown in Fig. 8.22.

### 8.3.2 Effect of Flight Clearance

The effect of the radial clearance on the optimum helix angle is shown in Fig. 8.23 for two values of the pressure gradient.

It can be seen that the radial clearance has relatively little effect on the optimum helix angle. The corresponding SEC as a function of clearance is shown in Fig. 8.24.

It can be seen that there is an optimum value of the clearance \( \delta^* \) for which the SEC reaches a minimum value. For the large pressure gradient, the optimum clearance \( \delta^* = 20E-5 \) m and for the small pressure gradient, \( \delta^* > 30E-5 \) m. Thus, the standard radial clearance is not necessarily the best clearance in the metering section of the extruder. However, as discussed in Section 8.2.2, the radial clearance in the plasticating zone should be as small as possible to enhance the melting capacity. This indicates that a varying clearance along the length of the extruder may not be a bad idea. This can be achieved by varying the barrel inside diameter and/or the screw outside diameter. Unfortunately, wear usually occurs right at a location where it should not happen: in the compression section. If wear would occur in the metering section, it could actually have a beneficial effect.

![Figure 8.23. Optimum helix angle versus radial clearance](image-url)
The optimum radial clearance can be determined from:

\[
\frac{\partial Z}{\partial \delta} \dot{V} = \frac{\partial \dot{V}}{\partial \delta} Z \quad (8.76)
\]

Again, the resulting equation cannot be easily solved analytically. Thus, the optimum clearance can be found by solving Eq. 8.76 numerically or graphically.

### 8.3.3 Effect of Flight Width

If the various contributions to power consumption are carefully examined, it can be seen that a substantial portion is consumed in the clearance between the flight tip and the barrel; see, for instance, Eq. 8.71(d). The power consumption in the clearance is inversely proportional to the radial clearance and directly proportional to the total flight width \(w\). However, the viscosity in the clearance will generally be lower than the viscosity in the channel since the polymer melt is pseudo-plastic. Figure 8.25 shows how the ratio of power consumption in the clearance \(Z_{cl}\) to the total power consumption \(Z_t\) depends on the ratio of flight width \(w\) to channel width plus flight width \(W + w\) for a polymer with a power law index \(n = 0.5\).

Figure 8.25 shows that the relative contribution of the power consumption in the clearance increases strongly when the flight width increases. A typical ratio of \(w/(W + w)\) is about 0.1; in the example shown in Fig. 8.25 this corresponds to a power consumption in the clearance of about 40% of the total power consumption! The power consumed in the clearance does not serve any useful purpose. It does not aid in transporting the polymer forward, but causes a viscous heating of the polymer. Therefore, one would like to make the flight width as narrow as possible to reduce the power consumption in the clearance. The power consumption in the clearance will be more pronounced when the material is more Newtonian in flow behavior, i.e., less shear-thinning. This is shown in Fig. 8.26, where \(Z_{cl}/Z_t\) is plotted against the power law index of the polymer melt. The viscosity in the clearance is determined from:

\[
\mu_{cl} = m \left( \frac{v_1}{\delta} \right)^{n-1} \quad (8.77)
\]
The viscosity in the channel is determined from:

\[ \mu = m \left( \frac{v_b}{H} \right)^{n-1} \]  

(8.78)

The results shown in Figs. 8.25 and 8.26 are for a standard radial clearance of 0.001 D.

The contribution of the power consumption in the clearance rises dramatically when the power law index increases. When the fluid is Newtonian, around 80% of the total power is consumed in the clearance! Thus, problems with excessive power consumption are more likely to occur when the material of a certain melt index is less shear thinning. This is the main reason behind the extrusion problems encountered with materials such as linear low-density polyethylene, LLDPE [4,5] and metallocenes.

The most logical way to reduce power consumption in the metering section is to reduce the ratio \( w/(W+w) \). This can be achieved in two ways. One is to increase the channel width \( W \) by increasing the helix angle, as discussed in Section 8.3.1. The other approach is to reduce the flight width itself. The combination of increasing the helix angle and decreasing the flight width is obviously most effective. This allows a reduction of the \( w/(W+w) \) ratio from a typical value of 0.1 down to around 0.03. The minimum flight width is not determined by functional considerations but by mechanical considerations. Functionally, one would like the flight width to be almost infinitely thin. However, there must obviously be sufficient mechanical strength in the flight to withstand the forces acting on it. These mechanical considerations are discussed in detail in Section 8.1.2.

The flight width in the metering section \( w_m \) can be considerably narrower than the flight width in the feed section \( w_f \) because the flight height or channel depth is much smaller in the metering section. In order to keep the mechanical stresses in the flight approximately the same, the following rule can be used to determine the flight width in the metering section.

\[ w_m = w_f \left( \frac{H_m}{H_f} \right)^{0.5} \]  

(8.79)
Considering that the flight width in the feed section is generally about 0.1 D, Eq. 8.79 can be written as:

$$W_w = \frac{W_f}{X_c} = \frac{D}{10\sqrt{X_c}}$$  \hspace{1cm} (8.80)

where $X_c$ is the channel depth ratio $H_f / H_m$.

Based on these considerations, a new screw design was recently developed to reduce power consumption in materials like LLDPE; the screw is often referred to as the LL-screw [4,5].

Figure 8.27 shows a standard screw geometry and the LL-screw geometry. Both screws have a 38 mm (1.5 in) diameter and a 24 L/D ratio.

Essentially, the two screws differ in the helix angle and the flight width. Thus, the differences in performance between the two screws can be attributed solely to these two geometrical factors. Figure 8.28 shows the predicted output versus hp curves for the two screws shown in Fig. 8.27.

Figure 8.29 shows the actual output versus hp curves for the same two screws.

It is clear that the optimized geometry of the LL-screw results in considerably lower power consumption compared to the standard geometry. The drop in power consumption is around 35 to 40%! It is also interesting to note that the predicted power consumption, Fig. 8.28, agrees quite well with the actual power consumption, Fig. 8.29. The difference in the ordinates is caused by the fact that Fig. 8.28 is predicted power for the metering section only, while Fig. 8.29 is actual total power consumption. A patent was issued on the LL-screw [65]; Migrandy Corporation obtained a license to supply extruder screws using this technology.

It should be noted that the benefits of a reduced flight width and increased helix angle are valid for the plasticating zone as well. The power consumption in the plasticating zone of the extruder is also reduced when the flight width is reduced. In Section 8.2.2, it was discussed that increasing the helix angle improves melting performance. Thus, the combination of increased helix
8.3 Optimizing for Power Consumption

Figure 8.27. The LL-extruder screw versus a standard extruder screw

Figure 8.28. Predicted output versus power consumption

Figure 8.29. Actual output versus power consumption
angle and reduced flight width should have a beneficial effect not only on the melt conveying zone, but also on the melting zone of the extruder.

8.4 Single Flighted Extruder Screws

In the previous sections of this chapter, screw design was analyzed by functional performance. By using the extrusion theory developed in Chapter 7, it was shown how the screw design can be determined quantitatively for optimum performance. In this section, screw design will be approached from another angle. Screw designs in use today will be described and their advantages and disadvantages will be discussed and analyzed.

8.4.1 The Standard Extruder Screw

In many discussions on extrusion, reference is made to a so-called standard or conventional extruder screw. In order to define this term more quantitatively, the general characteristics of the standard extruder will be listed; see also Fig. 8.30:

- Total length 20–30 D
- Length of feed section 4–8 D
- Length of metering section 6–10 D
- Number of parallel flights 1
- Flight pitch 1 D (helix angle 17.66°)
- Flight width 0.1 D
- Channel depth in feed section 0.15–0.20 D
- Channel depth ratio 2–4

These dimensions are approximate, but it is interesting that the majority of the extruder screws in use today have the general characteristics listed above. For profile extrusion of PA, PC, and PBTB, Brinkschroeder and Johannaber [46] recommend a channel depth in the feed section of \( H_f \approx 0.11 (D + 25) \) and a channel depth in the metering section of \( H_m \approx 0.04 (D + 25) \), where channel depth \( H \) and diameter \( D \) are expressed in mm. Based on these guidelines, the geometry of a standard extruder screw can be determined easily.

Based on the design methodology developed in Sections 8.2 and 8.3, it should be clear that the standard screw design is by no means an optimum screw design. It has developed over the last several decades mostly in an empirical fashion and works reasonably well with many polymers. However, significant improvements in performance can be made by functional optimization using extrusion theory. In this light, it is somewhat surprising that the standard extruder screw is still so popular today. It probably indicates a lack of awareness of the implications of extrusion theory on screw design and the improvements that can be realized from functional opti-
mization of the screw geometry. Another interesting note is that several manufacturers of extruder screws claim to use sophisticated computer programs to optimize the screw geometry, but often still end up with a standard square pitch screw. It can be shown from an elementary analysis that the square pitch geometry is not optimum for melting or melt conveying. Thus, if the result of the screw optimization by computer is a square-pitch geometry, this indicates that either the computer program is incorrect or the person using the program is not using it correctly.

8.4.2 Modifications of the Standard Extruder Screw

There are a large number of modifications of the standard extruder screw in use today. It will not be possible to mention all of them, but an effort will be made to discuss the more significant ones. Figure 8.31 shows the standard screw with an additional flight in the feed section.

The additional flight is intended to smooth out the pressure fluctuation caused by the flight interrupting the in-flow of material from the feed hopper every revolution of the screw. An additional benefit of the double-flighted geometry is that the forces acting on the screw are balanced; thus, screw deflection is less likely to occur. On the negative side, the additional flight reduces the open cross-sectional channel area and increases the contact area between solid bed and screw. Thus, pressure surges may be reduced, but the actual solids conveying rate will be reduced as well. As a result, a double-flighted feed section in smooth bore extruders often results in reduced performance.

Figure 8.32(a) shows a variable pitch extruder screw.

The varying pitch allows the use of the locally optimum helix angle, i.e., optimum helix angle for solids conveying in the feed section and optimum helix angle for melt conveying in the metering section of the screw. This design is covered by a U.S. patent [6] and is described in a 1980 ANTEC paper [7]. Figure 8.32(b) shows a variable pitch extruder screw as often used for rubber extrusion, see also Section 2.1.4.

In this design, the pitch decreases with axial distance as opposed to the screw shown in Fig. 8.32(a). The reducing pitch causes a lateral compression of the material in the screw channel; as a result, the normal compression from the reducing channel depth can be reduced or eliminated.

---

Figure 8.31. Standard screw with additional flight in the feed section

Figure 8.32(a). Variable pitch extruder screw with increasing pitch

Figure 8.32(b). Variable pitch extruder screw with reducing pitch