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MATCHING ENGINE AND PROPELLER

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ABSTRACT

The torque and shaft speed (rpm) of an engine will be the same as the torque and rpm of the propeller it drives. The marine designer attempts to place this torque-rpm coincidence at a point that will be best by some criterion for the total propulsion plant, and that will be satisfactory for the individual components under all operating conditions. This is the "matching" problem. Basic principles of driver-load relationships, the fundamental problem of choosing the matching point, and allowances for deteriorations in service, are developed here. Effects of towing loads and of auxiliary loads are also discussed. The use of a controllable-pitch propeller complicates the matching, since propeller pitch variations constitute a degree of freedom in addition to that provided by engine fuel control. The marine designer's task with propulsion engines driving this type of propeller is outlined.

CHAPTER 5

MATCHING ENGINE AND PROPELLER

5.1 INTRODUCTION

The matching of an engine and its propeller is a design process that seeks to establish the optimal fuel-to-thrust conversion under rated operating conditions, while ensuring that all possible operating conditions are acceptable to each component.

The process is basically one of working with the power-rpm or torque-rpm characteristics of engine and of propeller. Conservation of energy demands that the power produced by the engine (minus any loss in transmission) equal that absorbed by the propeller. Engine torque, multiplied by the reduction gear ratio (if any), likewise must equal propeller torque, and must do so at the common rpm. Since the respective characteristics are always represented graphically, the matching technique involves finding the intersections (i.e. points where torque, power, rpm, are equal) of these curves, then adjusting engine or propeller parameters to place these intersections at the desired locations.

The process is somewhat complicated by changes that occur in service. Among these are minor changes in hull resistance because of fouling, weather variation, or draft changes; changes in number of engines running (in multi-engine installations); and possibly the gross change in resistance that occurs if a tow is taken on.

In some instances the propulsion engine also drives auxiliary loads, such as electric generator or hydraulic system pump. The effect of such loads on matching to the propeller needs to be considered, as well as the problem of matching the auxiliary itself.

If a controllable-pitch propeller is used, some aspects of the matching problem are eased, since pitch can be adjusted in service. But this second degree of freedom also allows operation of the propulsion plant at unfavorable points, so that the designer usually removes the second degree of freedom by linking pitch control and throttle control in a definite program. Construction of this program is considered here as a major adjunct of the matching problem.

5.2 POWER SYSTEM PRINCIPLES

The matching of engine and propeller is an application of the principle of conservation of energy. In the present context, the principle is this: power produced by the engine must equal the power absorbed by the load. In a simple application, this statement comes close to being trivial, and may be intuitively obvious. Things are a bit less obvious in complex cases, as when several engines drive the same load, or when an engine simultaneously drives several diverse loads. It is therefore worth the effort to pay heed to some generalities before looking specifically at engines and propellers.

Consider first that however power is transmitted, it is characterized by two factors. In the case of transmission by a rotating shaft, these factors are speed (rpm) and torque. Electrical transmission involves voltage and current; hydraulic involves pressure and flow rate; etc. Always, one factor must be the same at both ends of the transmission link (barring leakage). In the mechanical case, rpm is the factor that must be the same at both ends. Torque may be less at the output end than at the input end, if bearings or other devices absorb energy along the shaft. If there are no such absorbers, then torque also must be the same at both ends.

Consider then a simple case: a driver (engine) linked to a load (propeller) by a shaft without bearings or other intermediate absorbers of energy. Shaft rpm must be the same for both, and since conservation of energy requires that power (i.e. energy rate) be the same for both, torque must be the same for both also. Each of the two units will have a torque-rpm *characteristic*, i.e. a relationship between its rpm and the torque that it can produce, or absorb, at that speed. If these characteristics are plotted on the same graph, their intersection shows the torque-rpm coordinates at which both factors are equal for both units, and therefore shows the operating point for the load-driver combination. See Figure 5.1. This is the point of equilibrium between load and driver, and is their only possible steady-state operating point.

Usually this pair of curves intersects in only one point, and usually the intersection is such that the equilibrium is a stable one. Stability can be inferred from the characteristic curves. Note in Figure 5.1 that if a momentary aberration of some kind caused speed to increase above the point of intersection, torque demanded by the load to maintain that rpm would

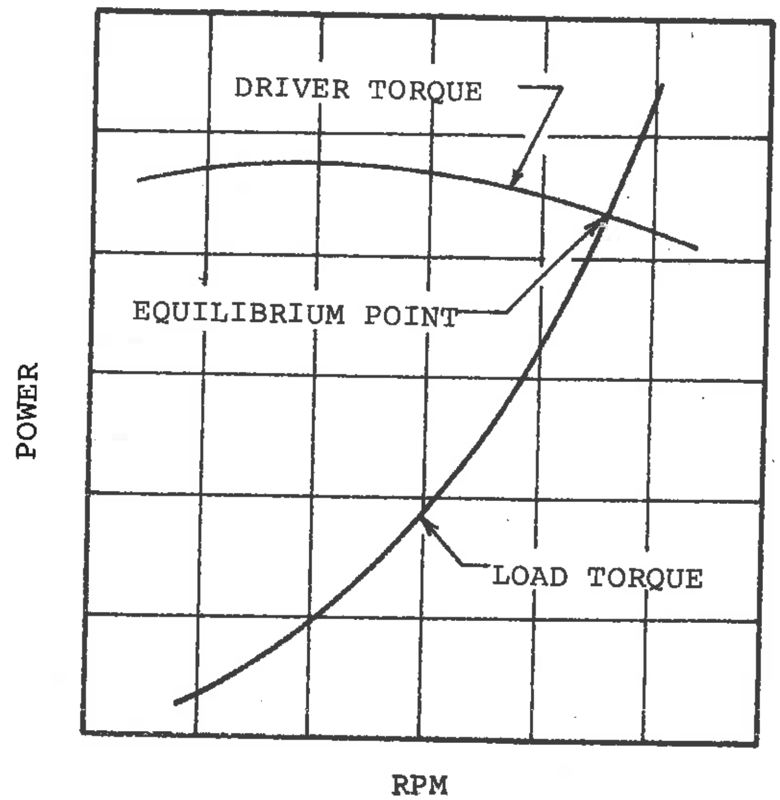


FIGURE 5.1 Driver and Load Equilibrium

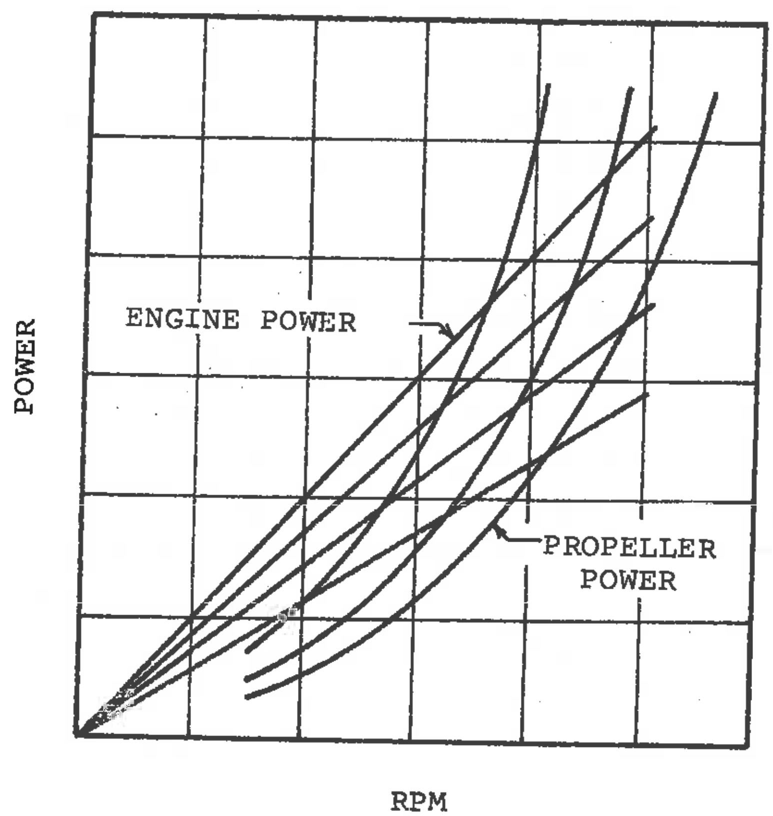


FIGURE 5.2 Engine and Propeller Equilibrium

exceed that available from the driver, hence the machine would accelerate toward equilibrium. Since deviations are automatically reduced to zero, the equilibrium is stable.

Changes in torque due to bearing losses, and in rpm due to reduction gearing, occur in practice, and complicate this discussion--but only slightly. If a reduction gear is used to change rpm between driver and load, lump the gear with the driver, i.e. take driver rpm (and hence load rpm) to be engine rpm divided by gear ratio; take driver torque to be engine torque multiplied by the gear ratio, less any torque losses in the gear set. Torque losses in the shaft bearings can be subtracted from the driver torque, or added to the load torque, before the plot of Figure 5.1 is made. In this fashion, the problem is reduced easily to the simple one of driver-and-load.

The preceding paragraph gives a hint of the technique to follow in analysis of yet more complicated cases; in a word: lump. This means to combine (lump) all load characteristics into a single characteristic; then find the intersection as in Figure 5.1. An example will illustrate: a diesel engine and a gas turbine drive a propeller through a common reduction gear, with an electric generator being attached to the same gear set; the problem is to find propeller rpm under a specified set of operating conditions. Let's suppose that torque-rpm curves are available for propeller, generator, diesel, and gas turbine, and that reduction gear power loss is known, for the specified condition. Reduction gear ratios for the diesel, gas turbine, and generator (all with repect to propeller rpm) are r_d:1 , r_{gt}:1 , and r_{gen}:1 , respectively. Proceed as follows:

1. Change the scales on the diesel torque-rpm plot by dividing rpm scale by r_d , and multiplying torque scale by r_d .
2. Do same for the gas turbine, using r_{gt} , of course.
3. Observe that conservation of energy requires that total drive torque equal the sum of diesel torque and gas turbine torque. Since steps 1. and 2. have adjusted their rpm to a common basis, simply add their individual torques to get a total driver torque.
4. Divide power lost in the reduction gear by rpm (using suitable constants for consistent units) to get the lost torque.
5. Subtract the gear lost torque from the curve found in 3. This now

represents the torque-rpm behavior of the driver--the lumped combination of diesel, gas turbine, and reduction gear.

6. Now work on the loads, lumping them in similar fashion. First, change the scales on the generator torque-rpm plot by dividing rpm by r_{gen}, multiplying torque by r_{gen}.
7. Conservation of energy requires that total load torque equal sum of generator and propeller torques. Add their torque curves to produce the total load torque-rpm characteristic.
8. The problem is now reduced to one of a single driver and a single load. Torque-rpm characteristics of both are on the same scale. So simply intersect the two curves, as done in Figure 5.1.

An example illustrating this technique more specifically is given in Section 5.5.

It should be observed that a power-rpm characteristic conveys the same information as torque-rpm, since power is proportional to the product of torque and rpm. The analysis discussed above could therefore be done as well with power-rpm plots, and you will find in following sections that power-rpm is used frequently in lieu of torque-rpm. It's mostly a matter of convenience, of using whichever form of the data is furnished in the instance at hand.

5.3 MATCHING ENGINE AND PROPELLER AT THE DESIGN POINT (FIXED PITCH PROPELLER)

5.3.1 The Designer's Fundamental Problem

If propeller and engine torque-rpm curves, or power-rpm curves, are plotted together, the intersection shows the point of operation. The problem facing the designer is one of choosing which engine curve and which propeller curve to offer for intersection--there can be many of each. See Figure 5.2. There, on the power-rpm plane, four engine curves and three propeller curves are shown. At first glance it would appear that suddenly there are twelve intersections to contend with rather than the one promised in Section 5.2. Not so; the figure is to illustrate that the designer must choose which of many engine curves he wishes to employ, and which of the many propeller curves, so that the two chosen will intersect at the desired spot.

The engine curves in Figure 5.2 represent engine power at different values of its brake mean effective pressure (bmep). Presumably one such curve represents the rated* bmep of the engine, and tentatively would be the one to use, although operating at higher or lower bmep is possible, and may be desireable in some circumstances. Alternatively, these curves might each represent the rated condition for four different engines being considered.

The propeller curves of Figure 5.2 most likely represent propellers that differ only in pitch. Normally, diameter is made as large as the space behind the hull allows, number of blades is chosen to minimize vibration excitation, area ratio is chosen for satisfactory loading, etc. Matching to the engine typically involves only choice of pitch, and such is the situation assumed here.

An important consideration, not evident in Figure 5.2, is the difference in propeller efficiency among the propeller curves. One such curve will produce the highest propeller efficiency, and the immediate--naïve--solution to the matching problem is to select that curve. To see the potential defect in this approach, look at Figure 5.3.

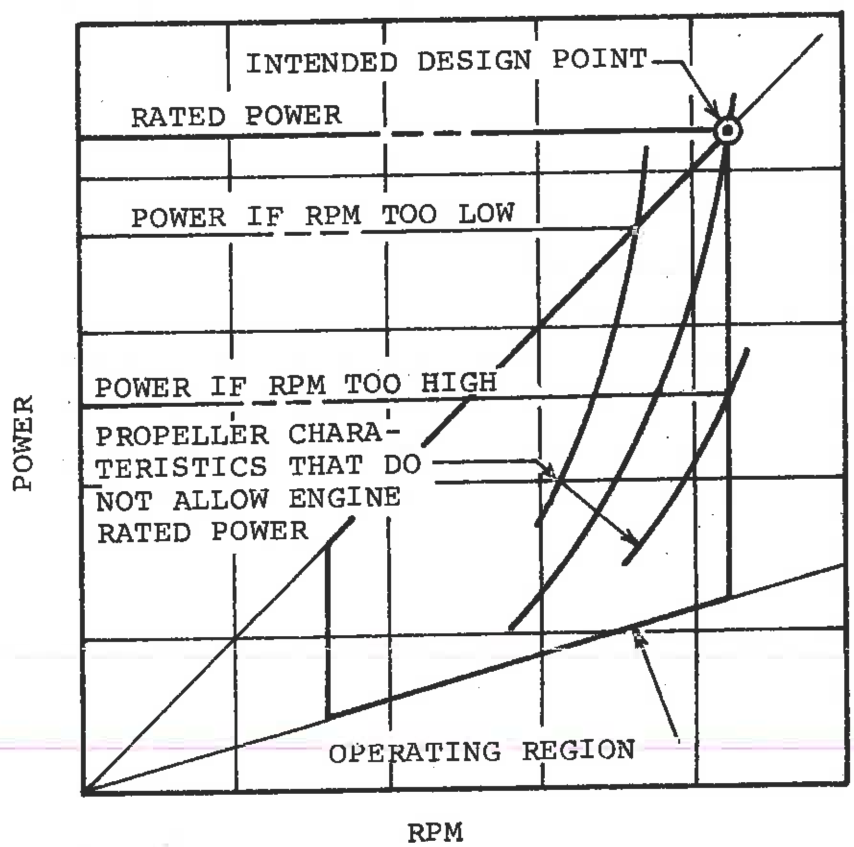


FIGURE 5.3 Loss of Power Capability if Rated RPM Is Incorrect

* Implications of the term "rated" are discussed in a subsequent chapter.

Figure 5.3 shows in simplified form, on the power-rpm plane, the allowable operating area for the engine. Note that there are upper and lower limits to rpm, and upper and lower limits to mean effective pressure. If the upper limits are the rated rpm and the rated bmep, then rated power of the engine is developed only at the upper right-hand corner of this area --only when the engine can simultaneously develop both rated rpm and rated bmep. If, then, the engine is to develop rated power, the propeller curve must pass through that point. Otherwise, the rated rpm will be reached before the rated bmep (pitch too low), or rated bmep will be reached before rate rpm (pitch too high). In both cases, as the figure shows, rated power cannot be reached. The designer's matching problem can therefore be called one of "hitting the corner" via his choice of propeller pitch.

If fortunate circumstances allow the propeller curve of highest efficiency to be the one that hits the corner, fine and dandy. If not so lucky, however, the designer must adjust the pitch to move the propeller curve, or must accept some overspeeding or overloading of the engine to reach rated power, or must select another engine whose "corner" is situated on the desired propeller curve. An example following may illustrate the dilemma somewhat.

5.3.2 An Example

A small research vessel requires 375 shp to reach its design speed of 10 knots. The largest practicable propeller diameter is six feet. Figure 5.4 shows the efficiency-rpm relation for the six-foot propeller absorbing 375 shp, pitch being chosen to suit at each point. Strictly on the basis of this figure, one would choose the pitch ratio that gives 250 rpm, since there the efficiency is highest. But now for the engine.

To keep the example simple, assume the choice to be limited to offerings of the Cummins Engine Co. A search through the data sheets for their marine engines reveals two possible choices, as summarized here:

Engine	V12-500M	V12-635M
Rated shp	376	458
Rated rpm	1800	1800

(medium commercial ratings used)

At first glance, the V12-500M looks to be the choice since its power is just right; the remaining problem is to pick a reduction gear for 250 rpm at the propeller. The ratio required is 7.2:1 . But the highest ratio available from suitable stock gears is 5.86:1 . With the latter gear, the V12-500M engine can't develop 376 shp with a propeller pitched for 250 rpm, unless the bmep is increased 23% above rated (1800/5.86 x 250 = 1.23).

Figure 5.5 shows that the V12-635M engine with 5.86:1 ratio reduction gear could produce the 375 shp at 250 propeller rpm, 1465 engine rpm, but its capability of producing 458 shp if allowed to turn at its rated speed would then be wasted. In other words, the owner of the vessel would pay for unused engine capability in order to have his highest possible propeller efficiency.

Suppose that the propeller is instead pitched to turn 307 rpm at the design point, thereby giving 1800 engine rpm with the 5.86:1 gear. Figure 5.4 indicates that the propeller efficiency is about 1.5 percent less than before. The V12-500M engine can be used, however, running at its rating.

The designer appears to have three principal choices. To summarize, they are

1. Pick the most efficient propeller and the smaller engine, overloading the engine by 23 percent in order to obtain the necessary shaft power. He must decide if the consequent shortened engine life is an acceptable price to pay for the benefits of having the highest possible propeller efficiency.
2. Pick the most efficient propeller and the larger engine. He must decide if the added cost and weight of engine is an acceptable price to pay for the benefits of having the highest possible propeller efficiency.
3. Pick the smaller engine, and pitch the propeller to allow this engine to develop its rated power. He must decide if the consequent loss of 1.5 percent in propeller efficiency is an acceptable price to pay for having the best engine choice.

Other alternatives may indeed exist. For example, if the propeller diameter is decreased, the rpm of highest efficiency is shifted upwards. Figure 5.4 shows this by incorporating curves for several smaller diameters. It also shows that the highest efficiency decreases, so that decreasing the diameter appears to be a poor way out of the dilemma posed by the three choices above. Another solution would be to have that 7.3:1 reduction gear designed and manufactured especially for this application. The high cost of this solution probably places it beyond consideration.

5.3.3 The Choice to Make

The preceding example has illustrated the engine vs propeller dilemma that may face a designer, but has not given the answer. A firm general answer cannot be given, because other circumstances of a particular case may be heavily significant. For example, that 1.5 percent difference in propeller efficiency may be significant if the vessel spends most of its time underway (i.e. burning fuel) but not if it spends most of its time in port. But in the absence of careful analyses of all such factors, the general recommendation is this: *pitch the propeller to let the engine develop its rating*. The basis for this is the rather small change in propeller efficiency that occurs for modest changes in propeller design rpm (e.g. 1.5 percent loss for a 23 percent increase in rpm in the example). The third alternative in the example would therefore be recommended in the absence of strong contrary evidence.

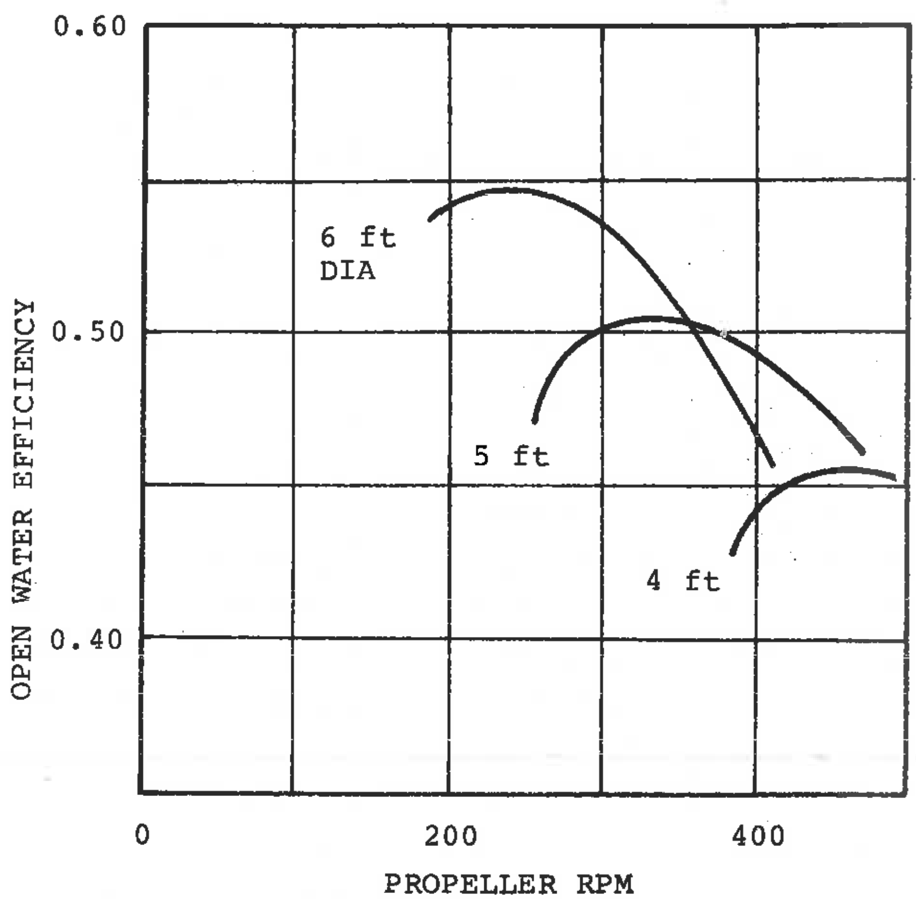


FIGURE 5.4 Illustrating Effect of Propeller Efficiency on RPM Choice

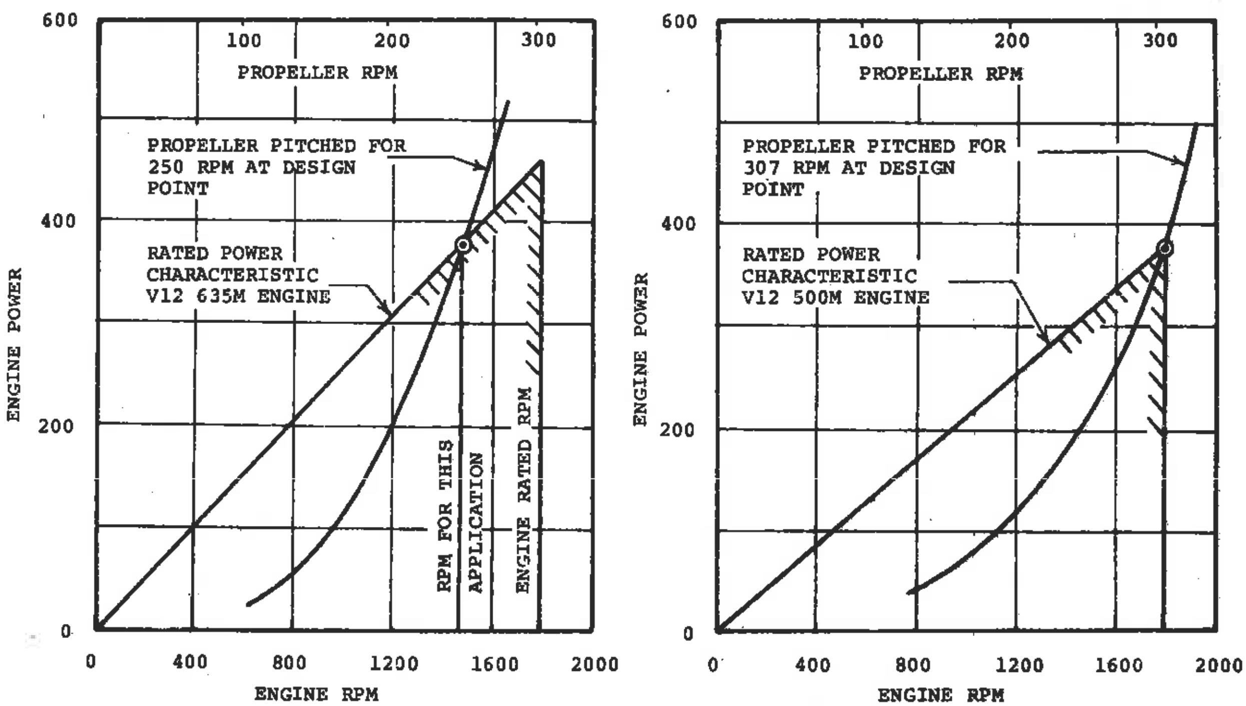


FIGURE 5.5 Illustrating Power Capability as Affected by RPM Choice

5.3.4 Uncertainty

Let's recognize that there are always *uncertainties* (finite probabilities of error) in design and manufacturing processes. For example, the propeller will not have exactly the pitch ratio specified, nor if it did, would it produce exactly the efficiency and rpm that the design charts predict. Analogous statements can be made for the engine. As a result, "hitting the corner" precisely can only happen fortuitously. Fortunately, the consequences of this handicap are absorbed by the modification to the design process discussed next.

5.3.5 Modification to Allow for Service Conditions

To summarize Section 5.3 to this point, it has been said that the engine develops its rated power when it produces its rated bmep at the rated rpm, and that the propeller pitch ratio should be chosen so that this condition is attained. Now this simple precept must be modified to accomodate certain changes that inevitably occur in service.

Sea state, wind, hull roughness, propeller roughness, and draft all affect the speed-power relation of the vessel. In most instances, the changes are unfavorable, e.g. bottom is rougher and weather is worse than under design conditons. In terms of the power-rpm plot, this means that the propeller characteristic shifts to the left. The consequences to the engine are readily seen in Figure 5.6: either the engine must slow down, thus losing power capability, or if rpm is to be maintained, the engine must be overloaded. There is little published data on the magnitude of this effect, but an eventual 20 percent increase in resistance at design speed appears to be a reasonable estimate of the hull roughening factor for an ocean-going steel ship [1]. This increase was used in constructing Figure 5.6, and this figure shows an 18 percent overload required to maintain rated rpm.

The solution is to select the engine and choose the propeller pitch so that design speed is attained on trial trip with a bmep less than rated. Then when deteriorations occur in service, bmep can be increased up to the rating without overload. On the trial trip, the engine will not reach its full power, and the vessel its maximum possible speed, because it will reach maximum rpm before reaching rated bmep. A partial remedy for this handicap is to allow a margin in rpm also, or (with engine builder's acquiescence) simply to overspeed the engine for a brief time on trials. These

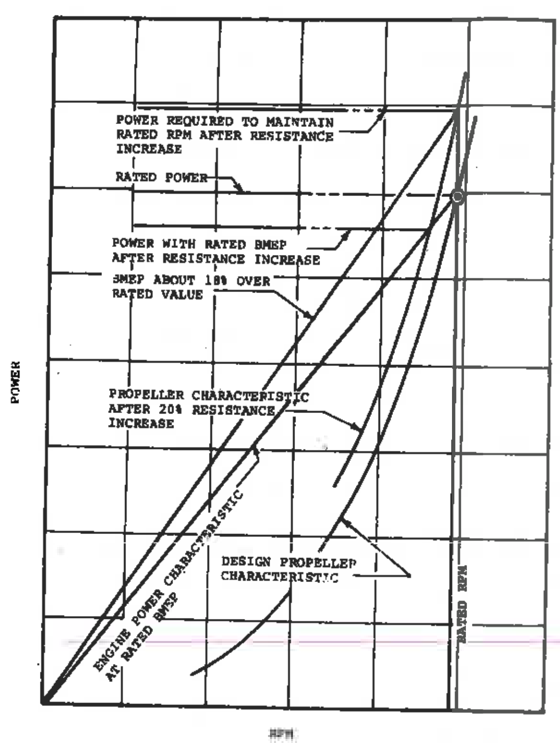


FIGURE 5.6 Power Loss Caused by Service Deteriorations

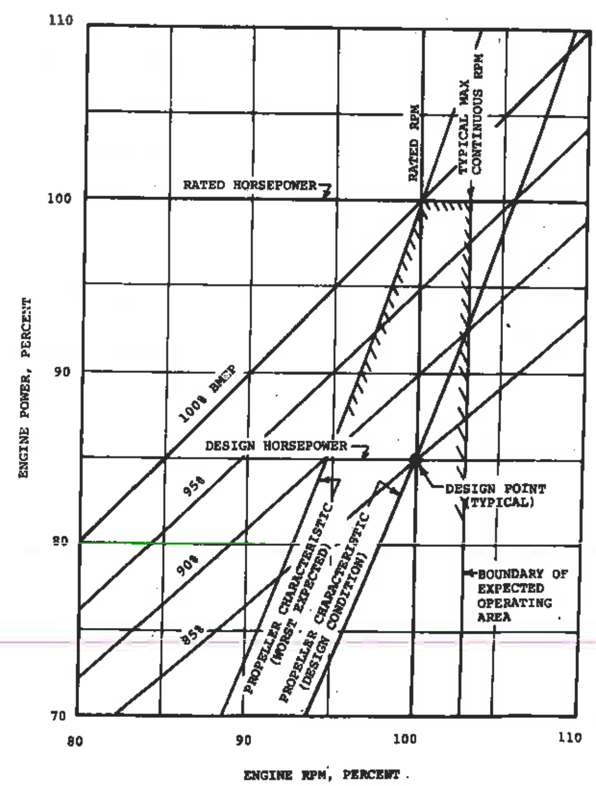


FIGURE 5.7 Setting a Margin to Offset Service Deteriorations

points are illustrated by Figure 5.7. This figure shows the design bmep set at 85 percent of the rated. This is a typical figure for ocean-going ships, but the designer should modify it as he will to suit the weather, surface coating, hull material, etc., appropriate to his vessel.

To summarize, the "corner" that must be hit to obtain the full capability of the engine should be moved slightly in the design process so that it will end up in the desired place after service deteriorations have occurred.

Returning for the moment to 5.3.4, observe that the effects discussed here are also highly uncertain in magnitude. The margin allowed is thus a crude estimate, and so covers the effects of design and building uncertainties mentioned in that section.

5.4 OFF-DESIGN OPERATION

As noted in Section 5.3, changes occur in propeller-hull conditions during service, and these are (or should be) compensated for by allowing a margin in selecting the design point. This section discusses changes of larger magnitude, generally ones that mere design margins cannot accommodate.

5.4.1 Multiple Engines

The use of more than one engine (as many as four per shaft have been used) connected to a single propulsion shaft is common. The connection is accomplished by means of a reduction gear pinion (input gear) for each engine. Clutches are nearly always provided so that an engine can be disconnected from the shaft while the vessel is underway with the remaining engine (or engines) furnishing power.

A common problem is a multi-engine plant is to predict the power available with one or more engines disconnected. Suppose, for example, that one engine of a pair is run at full throttle (rated bmep) while the other is disconnected. Figure 5.8, left sketch, shows the power-rpm situation. The operating point lies at the intersection of the propeller curve and the curve representing the remaining engine. Note that the on-line engine cannot develop its rated power because it cannot reach rated rpm. Thus the power availbable from one engine of a pair is distinctly less than one-half of the total.

Gas turbines and diesel engines are sometimes combined to drive the same shaft, the diesel to furnish the cruising power at its high efficiency, and the gas turbine to furnish boost power for occasional high-speed operation. If the two run together, the arrangement is known as CODAG (combined diesel *and* gas turbine). If one or the other carries the propulsion load, the arrangement is known as CODOG (combined diesel *or* gas turbine). In the former, the one-engine situation is the same as with two diesel engines, i.e. the cruising engine (diesel) cannot develop rated power because it alone cannot turn the propeller ar full rpm. See the middle sketch in Figure 5.8. It is essentially the same as the left sketch; only the shape of the gas turbine curve differs slightly from the diesel shape.

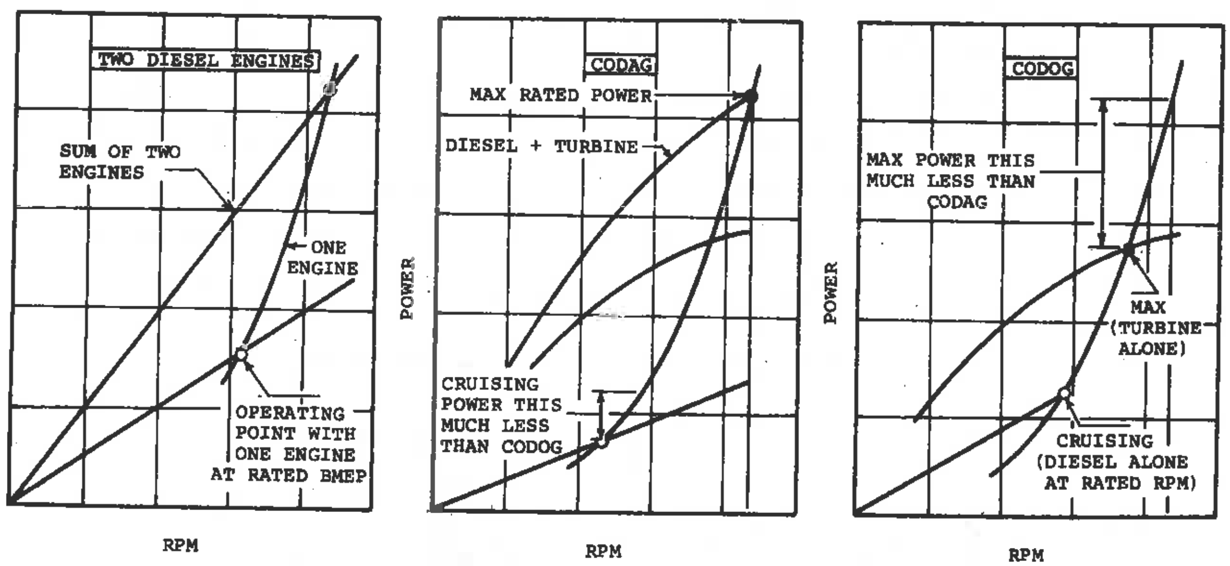


FIGURE 5.8 Combinations of Propulsion Engines

The defect in CODAG leads to consideration of CODOG. If only one *or* the other engine is run, then each can be matched independently to the propeller so that each can develop its full capability. The defect here is that the two engines cannot run together, since they would overspeed, and so maximum power is less than with CODAG. The controllable-pitch propeller offers a solution to this dilemma, as will be discussed in a later section.

Note from Figure 5.8 that the power-rpm (or torque-rpm also) characteristic of a combined plant is the sum of the characteristics of the individual components. Recall discussion of Section 5.2.

5.4.2 Low-Speed Operation

With a fixed-pitch propeller the only controlled variable is fuel flow to the engine. To slow down, you close the throttle. Obvious. Figure 5.9 illustrates, showing several power-rpm lines that result from reduced fuel flows. As always, the operating point is at the intersection of the engine curve and the propeller curve.

The diesel engine has a minimum speed at which it can furnish power. This speed is typically 20-30 percent of the rated rpm. The limit is indicated by the vertical dashed line in Figure 5.9. The implication is that the vessel in consequence cannot operate steadily below a certain speed, often a handicap in maneuvering. Many maneuver adequately by short bursts of ahead and astern rpm. Where this expedient is not satisfactory, there are several solutions. The simplest is to arrange clutch controls (usually pneumatic) for partial engagement; the clutch slips, thus allowing propeller rpm down to zero while the engine continues to run at its minimum rpm. Another is the controllable pitch propeller. With this device, thrust, and hence vessel speed, can be reduced to zero by reducing pitch while rpm remains high.

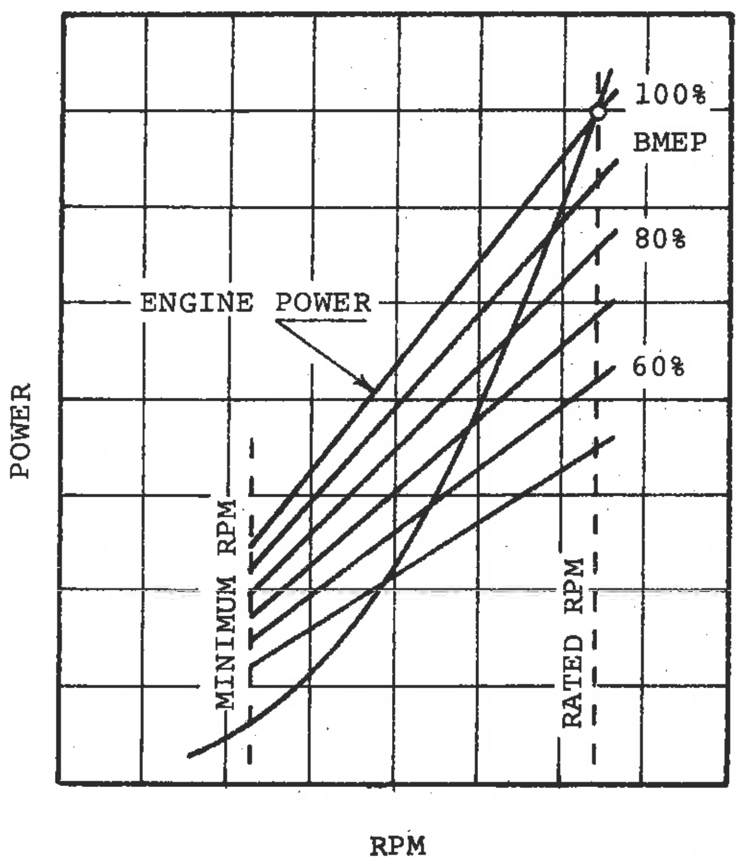


FIGURE 5.9 Low Speed Operation

5.4.3 Towing Loads

Towing loads represent an increase in resistance at a given speed, evident on a power-rpm or torque-rpm plot as a shift of the propeller curve to the left. Qualitatively, this is the same effect as that due to deterioration in service discussed in Section 5.3, but quantitatively may be so much greater that a separate discussion is warranted. Significant losses in propeller efficiency and vessel speed may demand special measures to compensate for them.

Figure 5.10 shows what happens, on both ehp-speed and shp-rpm planes. Look first at point A, assumed to be the design point, free-running condition. If the propeller curve shifts to the left as shown, and if the engine torque remains constant, the operating point shifts to the intersection at B. Because the engine rpm has fallen, the engine cannot produce its rated power. Speed of the vessel naturally is less because of the tow resistance, and because of the decrease in propeller efficiency (this loss is evident in the greater drop in ehp than in shp), and because of this engine power loss. However, there are several ways of keeping engine rpm, and hence power, constant. The line A-C on the left sketch (a point in the right sketch) in Figure 5.10 is a constant-shp contour, and indicates by its intersection with the propeller curve that ehp and vessel speed are significantly higher than otherwise.

The several ways of maintaining constant engine power and rpm in the face of such major changes in resistance are

1. A variable reduction gear ratio. A shift is made to a higher ratio, or a controlled slip is introduced.
2. A controllable-pitch propeller. Pitch is reduced so that rpm stays the same.
3. DC electric drive. In this drive system, speed of the propeller can vary independently of the engine speed.

Another type of remedy, suitable for a vessel specifically designed for towing, is to design the propeller for the towing condition, i.e. select its pitch ratio so that the engine turns at rated rpm under towing conditions. On Figure 5.10 this implies that point C, rather than point A, is the design point on the ehp-speed plane. There is a disadvantage, however, when the vessel is running free. Under this condition, the engine would tend to overspeed because of the light load, and thus the throttle must be

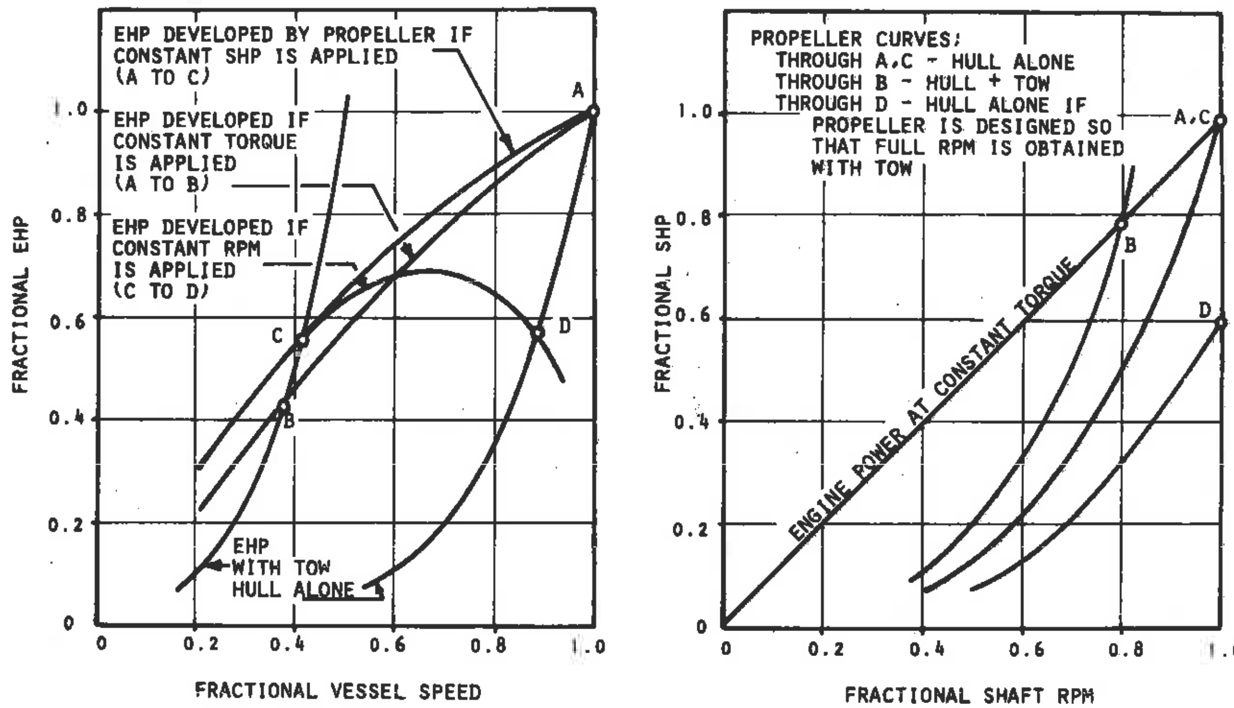


FIGURE 5.10 Loss of Power Capability Caused by a Towing Load

cut back to maintain rated rpm. The line C-D on the left sketch is a constant-rpm contour, and by intersecting the free-running propeller curve at D shows how much less power and speed must be than with the free-running design (point A).

It is appropriate here to explain how the contours in the left sketch of Figure 5.10 (lines A-B, A-C, C-D) are constructed. It is essentially a matter of manipulating the definitions of torque coefficient, thrust coefficient, and advance coefficient for the propeller, aided by definitions of ehp and shp, and by the assumption that wake fraction and thrust deduction are not affected by the change in resistance. First consider the constant-torque contour; it is found from these relations that

$$\frac{J}{J_0} = \frac{V}{V_0}\,\frac{N_0}{N} \tag{5.1}$$

$$\frac{K_Q}{K_{Q_0}} = \frac{N_0{}^2}{N^2} \tag{5.2}$$

$$\frac{K_T}{K_{T_0}} = \frac{T}{T_0}\ \frac{N_0{}^2}{N^2} \qquad (5.3)$$

$$K_Q = K_Q\ (J) \qquad (5.4)$$

$$K_T = K_T\ (J) \qquad (5.5)$$

$$\frac{P_E}{P_{E_0}} = \frac{T\ V}{T_0 V_0} \qquad (5.6)$$

where V = vessel speed
N = rpm
T = thrust
P_E = ehp

The first three of these are the relations for advance coefficient (J), torque coefficient (K_Q), and thrust coefficient (K_T), formed into ratios. The zero subscript implies the design point (i.e. point A); its absence implies some general point. Note that torque terms have cancelled out of the K_Q ratio, as a result of the constant-torque stipulation. The next two denote the graphical relations of K_Q and K_T to J for the particular propeller, such as shown in Chapter 2. (An empirical equation can be used for each of these relations if preferred to graphical work. A straight-line approximation is usually adequate.) The last is a ratio formed from the definition of ehp.

There are here six equations with seven unknowns (J, K_Q , K_T , P_E , V, N, T). If a value is fixed for any one of the unknowns, a solution is possible for any or all of the others. By choosing a set of V, you define the corresponding set of P_E, for example, and thereby obtain the contour on the ehp-V plane, as required. My favorite technique is to start with J, i.e. I choose a $J \neq J_0$, then proceed as follows:

(1) Read K_T and K_Q from the curves thereof.
(2) Solve for N/N_0 by equation (5.2).
(3) Solve for V/V_0 by equation (5.1).
(4) Solve for T/T_0 by equation (5.3).
(5) Solve for P_E/P_{E_0} by equation (5.6).

Repetition produces enough points to define P_E/P_{E_0} versus V/V_0.

If the contour of constant shp is wanted, a preliminary step is to multiply numerator and denominator of the K_Q formula by N, giving

$$K_Q = \frac{Q\ N}{D^5 N^3} \propto \frac{P_S}{N^3} \tag{5.7}$$

Then in place of equation (5.2)

$$\frac{K_Q}{K_{Q_0}} = \frac{P_S}{P_{S_0}} \frac{N_0^{\ 3}}{N^3} = \frac{N_0^{\ 3}}{N^3} \tag{5.8}$$

As indicated earlier, shp might be kept constant by keeping constant both rpm and bmep. In such a case $N/N_0 = 1$ also, and it is the pitch that must change. Equations (5.1) through (5.6) become, including the (5.8) modification

$$\frac{J}{J_0} = \frac{V}{V_0} \tag{5.9}$$

$$\frac{K_Q}{K_{Q_0}} = 1 \tag{5.10}$$

$$\frac{K_T}{K_{Q_0}} = \frac{T}{T_0} \tag{5.11}$$

$$K_Q = K_Q\ (J,P) \quad \text{(P symbolizes pitch)} \tag{5.12}$$

$$K_T = K_T\ (J,P) \tag{5.13}$$

$$\frac{P_E}{P_{E_0}} = \frac{T\ V}{T_0 V_0} \tag{5.14}$$

To plot a point on the constant-shp contour, choose a $J \neq J_0$. Read the pitch ratio from the K_Q - K_T - J chart to satisfy equation (5.10) at this J. Then read the K_T for this point. Remaining steps should be obvious.

5.5 AUXILIARY LOADS

An engine may drive more than one load. Two propellers driven by a single engine is an uncommon design, but sometimes seen. More common is the use of the propulsion engine to drive a generator, or the pump for a hydraulic system. Sometimes the power needed for these devices is so small compared to propulsion power that the designer may neglect its influence. But on the other hand, it may be necessary to predict the engine rpm under both loads, or perhaps how propeller rpm will change when the auxiliary load is cut in or out. The principle involved in the analysis has been stated in Section 5.2, and illustrated with a generalized example. In this section, a more specific example is used to demonstrate the solution technique in more detail.

A 100-hp engine drives the propeller of a small fishing vessel. A hydraulic system powers certain fishing gear; its pump is attached via a clutch to the forward end of the engine. The problem is to find the rpm and power available to propulsion when the pump clutch is engaged.

Recall from Section 5.2 that the recommended technique is to lump all loads and all drivers into a single load and a single driver. Only one driver is present here, but there are two loads. The power-rpm curve of one load (and likewise of the driver) is given in the bottom sketch of Figure 5.11 (curve labelled PROPELLER POWER). It is thus appropriate to combine loads on the power-rpm plane; first, however, the power-rpm curve of the second load (the pump) must be found from the head-flow curves given. The pump and the hydraulic system comprise a driver-load pair operating at the intersection of their respective head-flow characteristics. Power at the intersection is calculated from the product of head, flow, and pump efficiency, 50 percent being the value of the efficiency assumed in this example. This is done for the intersections corresponding to 100, 200, 300, 400, and 500 rpm. The results are plotted in the bottom sketch as the curve labelled PUMP POWER.

Pump and propeller might, of course, turn at different rpm because of different speed ratios between each and the engine, but here it is assumed that if this be the case, pump rpm has already been corrected to propeller rpm in the manner of Section 5.2. Total load as a function of rpm is therefore obtained by directly adding the powers along each of the two load curves to produce the curve TOTAL POWER in the figure.

The operating point is now found by intersecting the load curve with

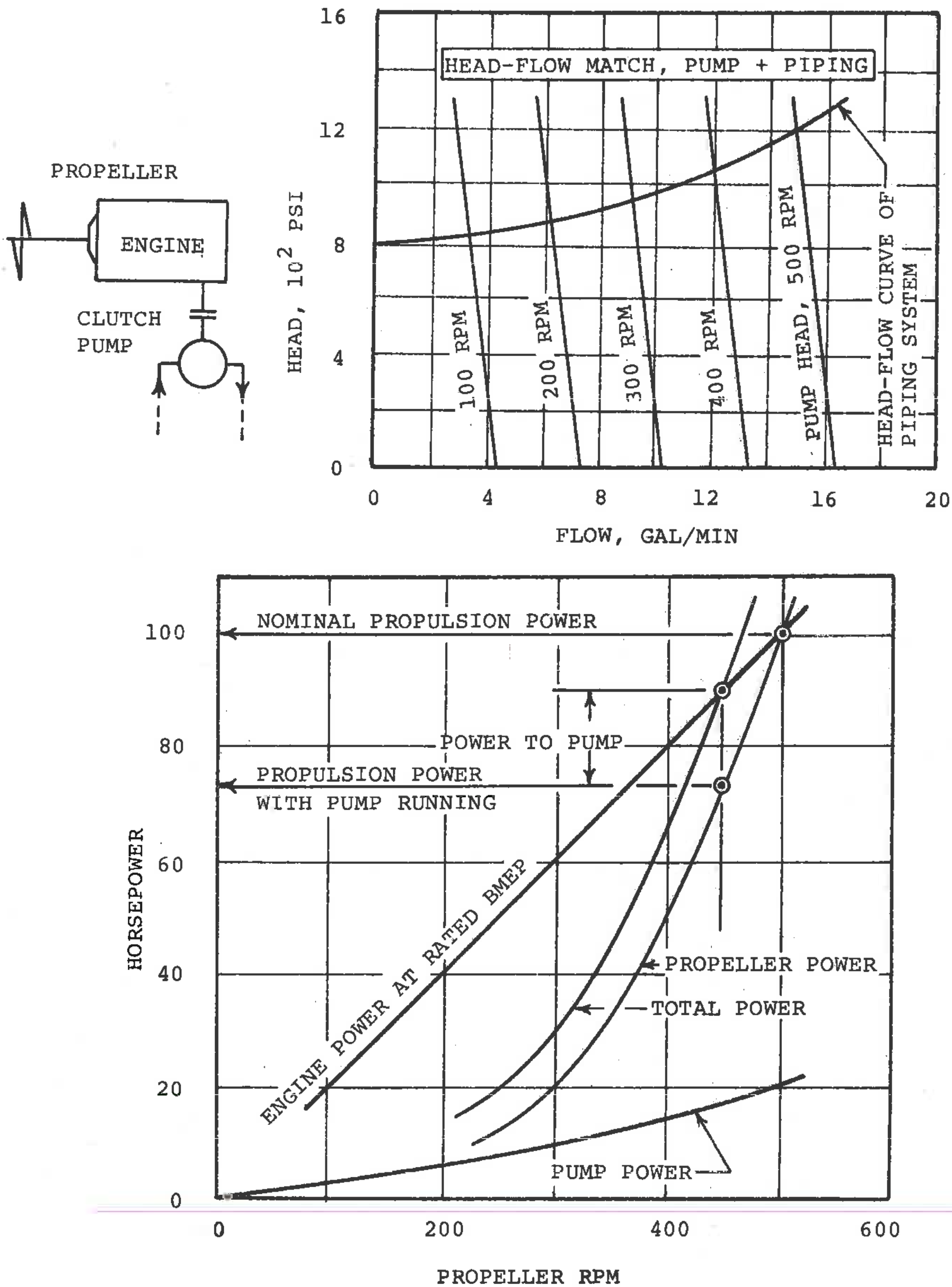


FIGURE 5.11 Illustrating Power Absorbed by Auxiliary Loads

the engine curve; the point lies at 450 rpm, 90 hp. To find propulsion power merely requires reading the power absorbed by the propeller at 450 rpm, i.e. 72 hp. Pump power is read to be 18 hp.

Note that propulsion power is 100 in the absence of the pump. Borrowing 18 hp to drive the pump reduces propulsion power to 72 hp, a seeming paradox (or error in arithmetic) that is explained by the graphical construction of Figure 5.11.

5.6 THE ENGINE AND THE CONTROLLABLE PITCH PROPELLER

The ability to change the pitch of the propeller gives the vessel operator a second degree of freedom (in addition to fuel control) in control of the propulsion plant. It allows him to compensate, at least in part, for poor operating conditions at part load, and for loss in output due to increases in hull resistance. Basically, the advantage conferred by this additional degree of freedom is the ability to adjust engine rpm to its best value, no matter what the conditions of vessel speed or resistance. This is not a matter to be left solely to the discretion of the vessel operator, however, for if he can adjust the rpm to the best value, he can also go the other way and adjust it to the worst value. He can do this through ignorance or laziness, but even with the best of knowledge and ambition, can't really do too well because the parameters he might judge by, such as fuel rate, bmep, and propeller efficiency, aren't readily discernible in practical operation. So that advantage will be taken of the pitch-change ability, the designer must usurp the second degree of freedom to his own domain. He extends his matching process to broader-scope endeavor: the construction of a pitch + fuel *program* that covers most operating conditions. This program forms the basis for design of a control system. The operator establishes the input to the system via a *single* handle or knob that may be calibrated in horsepower, vessel speed, thrust, or whatever. The system hardware carries out the wisdom of the designer by setting simultaneously the pitch and fuel controls according to the program. Design of the fuel-pitch program is the major topic of this section. First, however, a summary of the reasons--particularly from the viewpoint of the engine--of using a controllable pitch propeller (to be abbreviated CPP frequently hereafter).

5.6.1 Merits of the Controllable-Pitch Propeller

5.6.1.1 Part Load Efficiency

Figure 5.12 shows a fixed pitch propeller characteristic on the power-rpm plane, with typical engine fuel rate contours superimposed. All operating points must fall on the propeller line; it is obvious in the figure that the part-load points do not pass through the regions of best fuel rate. A part-load path that does pass through the best possible fuel rates is also shown, and operating points could be made to follow this line by pitch adjustment. If a vessel is to operate extensively at reduced speed, this potential fuel saving may justify the use of the CPP.

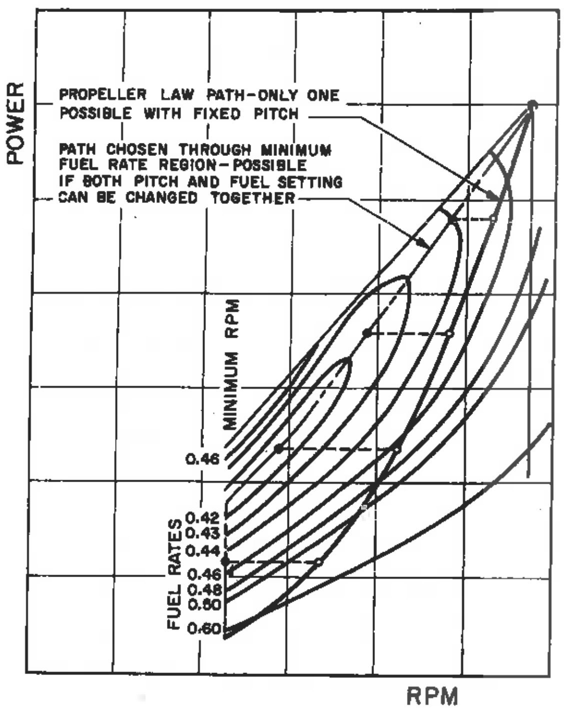
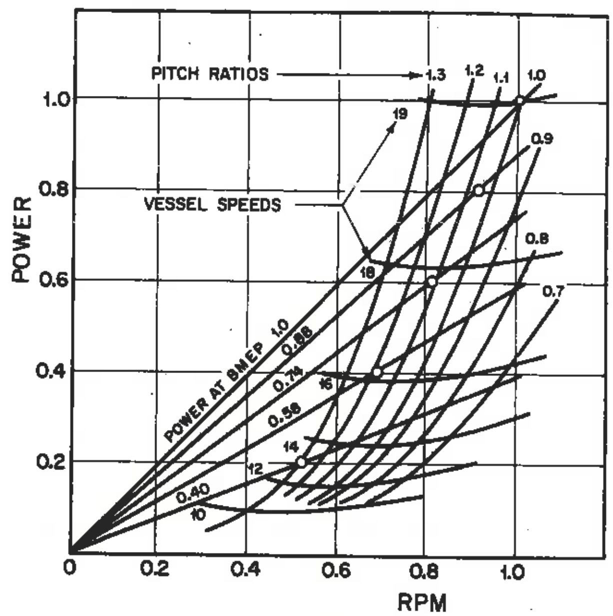


FIGURE 5.12 Minimum Fuel Rate Path on the Power-RPM Plane



POWER	RPM	PITCH	BMEP	SPEED
1.00	1.00	1.00	1.00	19.0
0.80	0.91	1.05	0.88	18.5
0.60	0.81	1.11	0.74	17.7
0.40	0.68	1.19	0.58	16.2
0.20	0.52	1.30	0.40	13.0

FIGURE 5.13 Constructing a Fuel and Pitch Program

5.6.1.2 Changes in Resistance

Earlier mention has been made of resistance increases and how they prevent the engine from developing rated power because of the shift in the power-rpm intersection point. This handicap afflicts all vessels because of weather, seastate, and hull surface deterioration. Towing vessels are additionally afflicted by the added resistance of the towed object. Tugs probably are the most severe cases, since they must occasionally operate against essentially infinite resistance (the *bollard pull* condition). By permitting an increase in rpm via a pitch reduction, the CPP erases the difficulty; although it cannot restore propeller efficiency to its best value, it can permit the engine to develop rated power. This benefit is discussed at many places in the literature; several references [2,3,4] are cited here.

5.6.1.3 Reversing

The CPP provides means of reversing thrust without reversing rotation. It thereby eliminates the need for a reverse train in reduction gears, or for reverse-running features with a direct-connected engine. Concerning the latter, the avoidance of many stops and starts during maneuvering is claimed by some authorities to be a significant factor in reducing engine maintenance.

5.6.1.4 Low-Speed Maneuvering

Diesel engines labor under the handicap of a minimum self-sustaining speed below which they cannot operate. Thus there is a minimum propeller rpm, and minimum steady vessel speed, which may be 20 to 30 percent of rated speed. This can be an obvious handicap to the vessel if it must maintain a low speed for a lengthy period (i.e. cannot rely on kicks ahead and astern for momentary speeds). The CPP offers a solution since its pitch can be reduced to any thrust from maximum down to zero, at rpm satisfactory to the engine. If it is used for other reasons, this feature is essentially a bonus, since it is automatically available, save perhaps for minor modifications to pitch-control hardware.

5.6.1.5 Combinations with Auxiliary Loads

If there is reason to want to maintain a fixed engine rpm, the CPP makes this type of operation feasible, since the full range of vessel speeds can be obtained by pitch change. For example, if the propulsion engine runs at a constant speed, it can also drive an AC generator. There is a cost saving because an auxiliary engine to drive the generator is not needed. (There may be need for a port-use generator, however, so that savings are often not as large as a first thought might suggest.)

Constant-rpm is unattractive from the part-load fuel economy standpoint, as can be seen simply by following a constant rpm path to low power in Figure 5.12.

A DC generator does not require constant rpm, but the use of a CPP also facilitates driving this type of machine from the main engine. This is because there is a limited range of speeds over which the voltage regulator would not be feasible (in vessels with small electrical loads, a battery takes care of the problem--as in the automobile).

Several references [3,5] have further discussion of points mentioned here.

5.6.1.6 Stopping Time and Distance

The time and distance required to stop a vessel depends on the astern thrust that the propeller can develop, and how quickly it can be developed following the decision to stop. The CPP can develop more thrust than its fixed pitch counterpart over most of the stopping maneuver, simply because its pitch can be adjusted to suit the highly off-design flow conditions that exist transiently. Thus it is that stopping time and distance are expected to be shorter when a CPP is used. Table 5.1, taken verbatim from a paper by Ridley and Midttun [6] lists data that tends to confirm the shorter stopping time of vessels with the CPP.

TABLE 5.1 Some Comparative Stopping Times, FPP and CPP

*B—Ballast. F—Full Load.

Ship	TDW	Power BHP	Starting Speed Knots	Stopping Time Seconds	Type	Condition*
M/S Andorra	12000	12000	19.8	183	CPP	B
M/S Azuma	13150	15000	22.0	160	CPP	B
M/T Esso Fawley	16700	10080	17.0	199	CPP	B
Tanker	18000	8000	15.0	534	FPP	F
M/S Columbialand	24850	11400	15.8	240	CPP	B
M/S Holtefjell	35500	12600	15.5	366	CPP	B
Tanker	33000	11000	15.0	558	FPP	F
Tanker	35000	12500	16.3	582	FPP	F
M/T Sinclair Venezuela	51300	2 x 8400	16.5	286	2 CPP	B
Tanker	47000	15000	16.6	560	FPP	F
Tanker	48500	16000	15.8	630	FPP	F
M/S Nuolja	72500	17600	17.0	420	CPP	B
M/S Nikkala	72500	17600	16.3	426	CPP	F
Tanker	65000	17500	17.0	690	FPP	F
Tanker	79000	22000	15.9	750	FPP	F
SS Fort Henry	12000	6000	20 MPH	293	FPP	B
M/V Fort Chambly	12000	6000	20 MPH	163	CPP	B
SS Murray Bay	25000	10000	17.75 MPH	296	FPP	B
M/V Saguenay	25000	9500	17.75 MPH	285	CPP	B

5.6.2 Matching the Propeller and Engine

Section 5.3 has discussed the matching of a diesel engine and a fixed propeller. To review, the message there is essentially this: the pitch of the propeller should be chosen so that the engine can develop its rated output, but usually with some margin incorporated so that the rated output will be obtained after service deteriorations have taken place.

With a CPP, the matching process is essentially the same, except that the margin need not be allowed. The change in pitch that is readily accomplished in service permits the engine to turn at the same rpm under both design and deteriorated conditions, and it is this constancy of rpm that is the benefit of margin with fixed pitch. The CPP therefore eliminates the need for the margin, and allows the vessel to develop full power under both trial and service conditions.

There is an additional feature of design with the CPP that merits inclusion in a discussion of the matching process; this is the selection of combinations of pitch, rpm, bmep that constitute the program for part-load operation, guided by a single input from the operator. The construction of this program usually must be modified to accommodate very low speed operation, and to respond to changes in external operating conditions.

5.6.2.1 Path of Operating Points

Figure 5.12 has shown that a path of best fuel rate exists on the power-rpm plane, and that it can be followed only if pitch can be changed. Typically, best possible fuel rate is the designer's objective, and the discussions here assume this to be the case. Other objectives are possible, however. For example, if an AC generator were attached to the propulsion engine, a constant-rpm path would be wanted. The reader should thus keep in mind that the theme at this point is the technique of composing the fuel-pitch program, and that a best-fuel rate criterion is being used for illustration.

To construct the program, three items of information are necessary, namely these:

1. Power-rpm (or torque-rpm) characteristics of the propeller.
2. Power-rpm (or torque-rpm) characteristics of the engine, with contours of constant fuel rate included, as in Figure 5.12. The scale must be the same as that of the propeller sheet.

3. A specification of the input calibration desired.

For the last of these items, let's assume that the operator's control handle is to be calibrated in engine horsepower, i.e. each position of this handle is to correspond to a particular bhp. The process of constructing the program can now proceed as follows:

1. Draw the desired path of part-load points on the engine power-rpm plane, as in Figure 5.12.
2. Mark on this path several points of bhp, likewise as in Figure 5.12. These should be at the powers specified to correspond to particular control handle positions, as mentioned just above.
3. Lay the propeller curves over the engine power-rpm sheet.
4. From the superimposed curves, record the bhp, bmep, pitch, and rpm at each of the points selected in 2.

Figure 5.13 illustrates this construction (fuel rate contours are omitted to avoid excess of lines). Note that several bhp points are marked, and that the information listed above (plus vessel speed, which isn't really needed) has been tabulated below the figure for each of these points. This tabulation is the program. It will subsequently serve as the key input data in the design of the hardware that will actually carry out the designer's intentions.

This example program has been based on the announced objective of obtaining the best fuel rate at part loads. But just as this chapter has earlier asserted that highest propeller efficiency is not necessarily the best criterion in matching propeller and engine, it should here point out that lowest fuel rate (quantity of fuel burned per hp-hr) is not necessarily the best criterion for high-efficiency part-load operation with the CPP. Again it involves a discrepancy between what is best for the engine and what is best for the propeller. Look at Figure 5.14. Here a highest-efficiency path has been drawn across a CPP power-rpm chart. A comparative glance back at Figure 5.12 shows that this path could not coincide with the lowest fuel rate path in that figure. Lowest fuel rate at part loads is therefore likely to involve a small sacrifice in propeller efficiency. A compromise between the two seems appropriate. Least fuel consumed per mile is perhaps a good compromise criterion to pursue, this involving both engine fuel rate and propeller efficiency. The designer may therefore wish to adjust his program points slightly in the direction of better propeller efficiency in order to

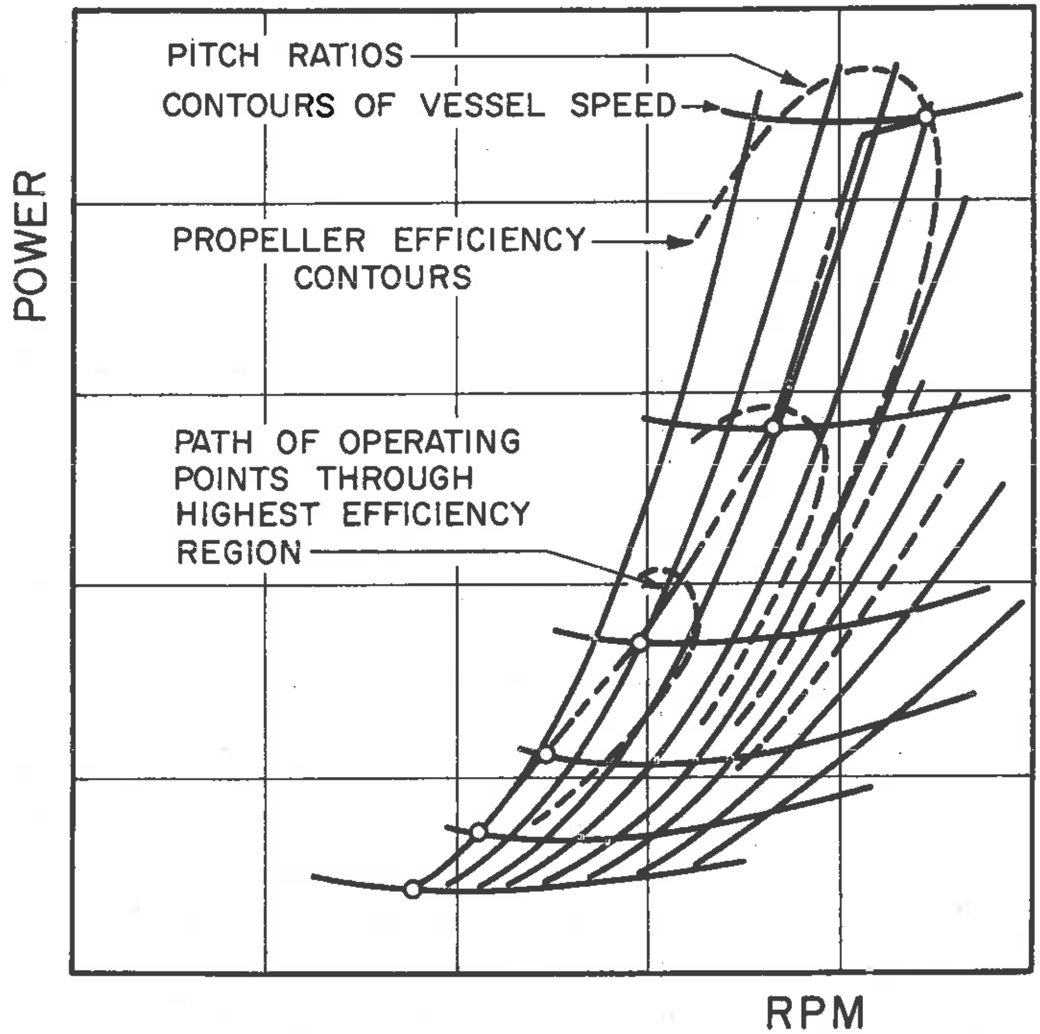


FIGURE 5.14 **Best Propeller Efficiency Path for Part Loads on CPP**

reach positions at which he calculates consumption per mile to be the least.

For use in constructing a good-fuel program when engine fuel rate contours are not known, Schanz [5] offers the following formula for the path across the power-rpm plane:

$$\frac{N}{N_0} = 0.5\left[\frac{P}{P_0}\right]^{\frac{1}{2}} + K\left[\frac{P}{P_0}\right]^{\frac{1}{3}} \qquad (5.15)$$

where

$$K = \frac{35n}{D\,N_0}\left[\frac{P_0}{a_d D^2}\right]^{\frac{1}{3}} \qquad (5.16)$$

and N = rpm D = propeller diameter, meters

P = power a_d = propeller developed area ratio

n = reduction gear ratio

$_0$ subscript denotes rated conditions

This formula may spot points outside the maximum bmep, rpm, or pitch, permissible or available with a particular engine-propeller combination, in which case the designer would move the point to the nearest boundary.

The points in Figure 5.13 are found by Schanz's formula.

5.6.2.2 Response to Changes in External Conditions

As noted earlier in several places, operating conditions are never precisely as predicted in design, and also undergo frequent variations as weather, hull surface roughness, etc., change. It was also noted in these earlier discussions that a partial remedy for the consequent losses in power and speed is the controllable pitch propeller, since it permits the engine to develop rated power under a wide range of conditions. To take advantage of this feature, the control scheme must have built into it a capability for automatically responding to a load change. Typically, the response is pitch change, under command of a governor, to maintain rpm constant. The engine, being governed for a particular rpm, thus sees no change, and remains at its previous operating condition--its power, rpm, torque, and efficiency are unaffected.

The engine program point is not moved, but the propeller point is. Using Figure 5.13 as an example, we might say that the propeller characteristics plot slides to the left or right as load changes, while the engine plot remains fixed. Constant torque and constant rpm means constant K_Q . Constant K_Q does not mean constant efficiency, as you can easily see by laying out a horizontal path on a K_T , K_Q , η_0 plot for any typical CPP. For large load increases, as occur in taking on a heavy tow, a significant decrease in η_0 is unavoidable. For small increases, the efficiency loss is usually insignificant. Schanz [5] discusses this point at some length, and indicates that a loss of less than one percent in efficiency can be expected over a moderate range of load changes.

5.6.2.3 Maneuvering Requirements

There are difficulties in both analysis and in hardware if the part-load program is extended to low vessel speeds. Basically, they are brought about by the cubic (or greater) variation in the propeller power-rpm characteristic; you can see by looking at several of the figures (e.g. 5.13, 5.14) how difficult it is to represent things accurately in the lower left corners. This difficulty might be assuaged simply by an enlarged plot, but it has its counterpart in the hardware; pilot valves, ports, etc., are sized for high power range, and so cannot be precise when handling trifling powers. The problem is aggravated by the lack of definition in propeller characteristics in that lower left corner, even after their scale is enlarged. For example, the input power to a CPP becomes virtually constant over the range of low speeds (very roughly, about the lowest ¼ of the vessel speed range) because its efficiency falls off about as fast as ehp does. All in all, the simplest solution is just let the operator set the pitch directly to give whatever speed through the water pleases him, while the engine is kept at a constant rpm by its governor.

5.6.3 Examples of CPP-Engine Control Systems

Most propulsion plants combining the diesel engine with the controllable-pitch propeller incorporate the three features that have been discussed above, namely (1) a part-load fuel-pitch program, (2) automatic response to changes in external conditions, and (3) direct setting of pitch, with engine rpm constant, at low vessels speeds. It may be quite interesting to the reader to see how these features are carried out in practice, i.e. the combinations of hardware that are employed. Delving into hardware is inappropriate to the theme of this chapter, but examples of CPP-engine control arrangements may be seen in a later chapter devoted to the control of marine engines.

5.7 THE ENGINE-PROPELLER DURING TRANSIENTS

During transient operations, conditions for both propeller and engine are likely to depart far from steady-state values. A crash stop, for example, requires the propeller to develop reverse thrust while the vessel is moving ahead. A braking torque is required from engine or shaft brake to arrest the ahead rotation of the shaft, and then an engine torque is required to accelerate the shaft in the reverse direction. If the controllable-pitch propeller is used, reverse rotation is not required, but torque similarly fluctuates from its steady-state value during the transient period.

A designer may be concerned with estimating how much time and distance are required to stop from a specified speed. The key data in the analysis is the thrust developed by the propeller. This depends upon its speed of advance and its rotational speed; the latter depends upon the torque that can be applied to the shaft. From the engine standpoint, consequently, the concern is with its ability to supply the torque required, or perhaps to avoid being stalled by excessive torque when attempting to reverse shaft rotation.

Analysis of transient operation is simple in principle. The basic equations are [7]

$$M \frac{dV(t)}{dt} + R(V) + T(N,V) = 0 \tag{5.17}$$

$$I \frac{dV(t)}{dt} + Q_e(N,F) + Q_m(N,V) + Q_f = 0 \tag{5.18}$$

where M = mass of vessel and added water mass

V = vessel speed

R = resistance of vessel

T = propeller thrust

N = propeller rpm

I = polar moment of inertia of rotating components, including that of added water mass

Q_e = torque developed by engine

Q_m = torque imposed on propeller by motion of ship

Q_f = fictional torque, including torque developed by a brake, if there is one

F = engine fuel setting

t = time

These equations are usually solved by step-wise integration because of non-linearities, and because the individual terms may represent information that is available only in graphical form (e.g. $T(N,V)$). The solution is beyond the intended scope of this text, and indeed, so little has been published on diesel propulsion transients that a complete exposition cannot be attempted. The present discussion is subsequently limited to qualitative remarks on the problems facing the engine during transients, particularly the crash reversal transients, plus a few example figures.

First consider the situation of a reversing diesel engine during a crash reversal. Upon receipt of a stop signal, its throttle is closed, but the engine continues to turn ahead because the propeller will windmill due to its continued forward motion (with some contribution from inertia). The friction torque of the engine, plus that of bearings and stern tube slows the shaft, and the windmilling torque decreases as the vessel slows. When the vessel has slowed to a point that the engine starter can run engine, shaft, and propeller up to firing rpm in the reverse direction, the engine is started in reverse, and the engine then accelerates the propeller in the reverse direction. The starting and firing torques must be sufficient for the task or the operator must wait until ahead rpm falls to a safe level before attempting a reverse start.

Figure 5.15 illustrates the stopping situation. It is a Robinson diagram, or plot of propeller torques vs rpm during a reversal. A dashed

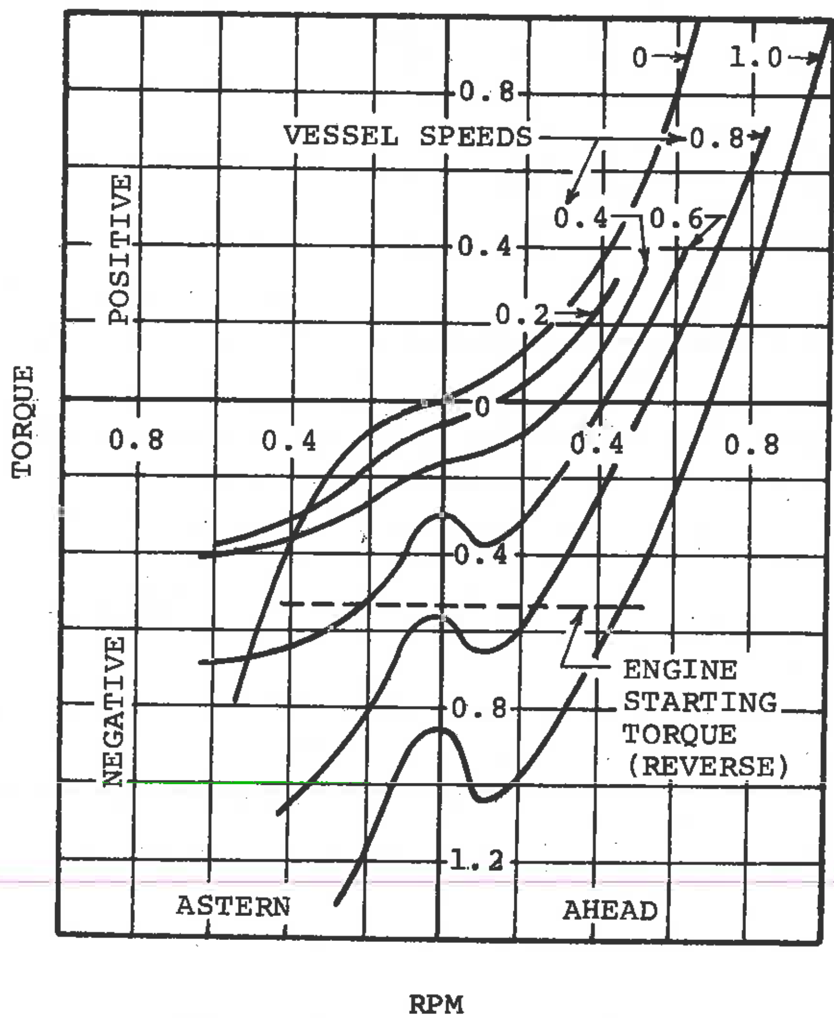


FIGURE 5.15 Propeller Torque During Deceleration and Reversing

line represents an assumed starting torque of a diesel engine. Note that vessel speed must fall to about 70 percent of full ahead before this torque is equal to the torque required to stop the shaft. If the minimum firing rpm of the engine is, say, 0.2, then the figure shows that vessel speed must fall to about 60 percent before the starting torque is sufficient to bring the engine up to firing speed.

Stopping can be accomplished in simpler fashion by stopping the engine, then coasting down to zero speed. However, stopping times and distances are so much shorter when reverse thrust is applied, that emergencies, and even most routine maneuvering, will demand use of the engine. Figure 5.16 illustrates this point by showing comparative stopping times for a large tanker, with and without use of the engine.

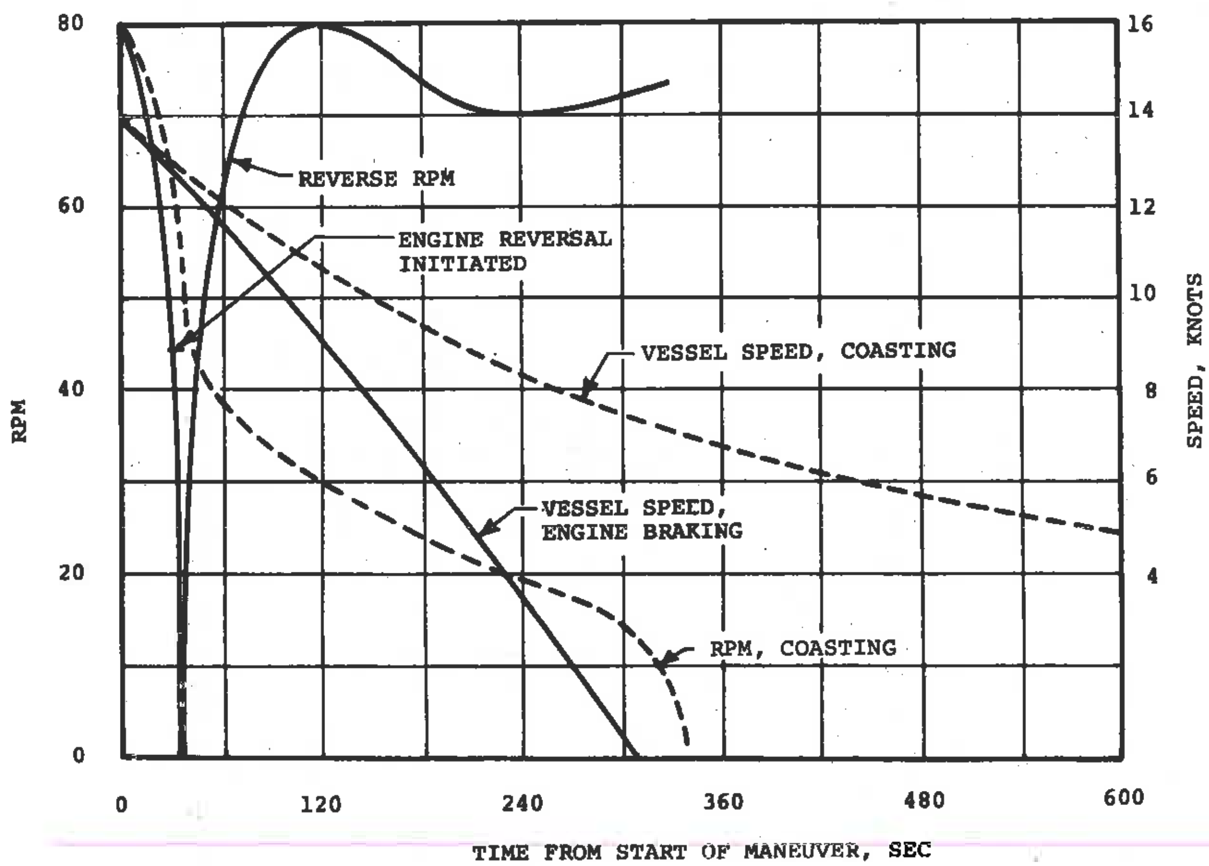


FIGURE 5.16 Crash Stop with Reversing Engine, Sulzer RD Engine (from Sulzer Sales Literature)

A more common installation among smaller vessels is the reverse-reduction gear; the engine continues in ahead rotation during the maneuver, but is declutched from the ahead gear train and reclutched into the astern train. Problems are to simultaneously reduce engine fuel to avoid over-speeding during the disengaged period and to reach an rpm at which astern clutch can be safely engaged, and to avoid stalling of the engine by the windmilling torque after engagement is effected. Usually the throttle and clutches are controlled by a common system, with interlocks and time delays for the necessary protection.

Figure 5.17 illustrates the reverse-reduction gear crash stop by plotting the variation of several machinery variables for a harbor tug. This vessel has a shaft brake that is automatically applied on receipt of the stop signal. Its torque is the chief factor in stopping the shaft, but the engine still must bear the burden of accelerating the shaft in the reverse direction.

The CPP reverses by reversing pitch, so that there is no stopping and starting of the engine, nor disengagement or engagement of clutches. Typically, the engine rpm is brought to idle before the pitch change begins, and is maintained there by its governor until full reverse pitch is reached. Thereafter, if required, it will provide the torque necessary to accelerate the shaft back to full rpm. Since the vessel may still be going ahead at high speed when reverse pitch is reached, torque may be quite high. The engine must, at the very least, supply sufficient torque to keep the shaft speed from falling below idle rpm. Better, from the standpoint of quick stopping of the vessel, is sufficient torque to accelerate the shaft.

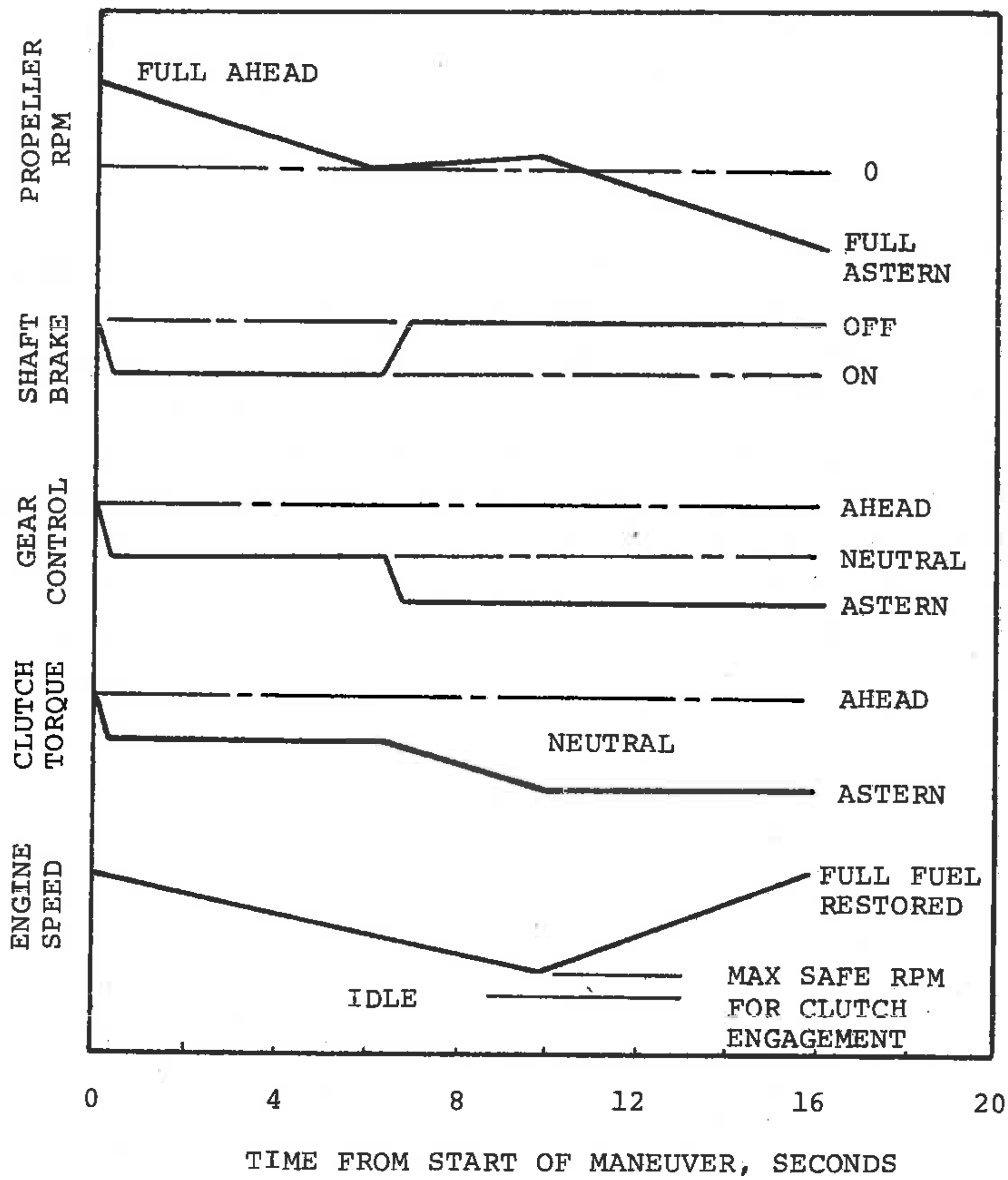


FIGURE 5.17 Change of Several Parameters During Reversal of a Tug with Reversing Transmission (from [4])

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