gear, β = 190/12 = 15.8 and the second, incorporating the next module in the standard list, leads to a narrow gear β = 99.6/16 = 6.2. The final solution is therefore chosen to be m =16 mm, with $\beta = \beta_{min}$ = 9 > 6.2, so the facewidth is f = 9*16 = 144 mm. Obviously this is over-designed and long-lived, but the cost of the extra width could be expected to be small. Other candidates with different profile shifts, materials etc. might be tried.

```
problem 13a wide
power, kW     75.0   PINION, WHEEL - speeds, rpm         90.0         48.6
appl'n factor 1.50   tooth number, profile shift       20  0.42     37  0.21
rel'y factor  1.00   all. contact, bending stresses  1300   180   1250   175 MPa
dist'n factor 1.38   bending geom & max life fctrs  0.455  1.04  0.416  1.04
vel'y factor  1.20   contact, bending life factors  0.803 0.847  0.835 0.954
velocity, m/s 1.18   contact, bending lives, khr    92.47 313.0  84.91 14.99
module, mm   12.00   pitting geom factor 0.1098    contact ratio 1.500
width, mm    190.0   commercial, 6 accuracy level gears
```

```
problem 13b narrow
power, kW     75.0   PINION, WHEEL - speeds, rpm         90.0         48.6
appl'n factor 1.50   tooth number, profile shift       20  0.42     37  0.21
rel'y factor  1.00   all. contact, bending stresses  1300   180   1250   175 MPa
dist'n factor 1.25   bending geom & max life fctrs  0.455  1.04  0.416  1.04
vel'y factor  1.24   contact, bending life factors  0.803 0.847  0.835 0.953
velocity, m/s 1.57   contact, bending lives, khr    92.62 314.8  85.05 15.08
module, mm   16.00   pitting geom factor 0.1098    contact ratio 1.500
width, mm     99.6   commercial, 6 accuracy level gears
```

```
problem 13c min width, excessive life
power, kW     75.0   PINION, WHEEL - speeds, rpm         90.0         48.6
appl'n factor 1.50   tooth number, profile shift       20  0.42     37  0.21
rel'y factor  1.00   all. contact, bending stresses  1300   180   1250   175 MPa
dist'n factor 1.30   bending geom & max life fctrs  0.455  1.04  0.416  1.04
vel'y factor  1.24   contact, bending life factors  0.681 0.608  0.708 0.685
velocity, m/s 1.57   contact, bending lives, khr    large large  large large
module, mm   16.00   pitting geom factor 0.1098    contact ratio 1.500
width, mm    144.0   commercial, 6 accuracy level gears
```

PROBLEM 14

```
problem 14 - essentially another solution candidate for worked synthesis example
power, kW    125.0   PINION, WHEEL - speeds, rpm        200.0         55.6
appl'n factor 1.00   tooth number, profile shift       10  0.55     36  0.22
rel'y factor  1.00   all. contact, bending stresses  1320   380   1100   360 MPa
dist'n factor 1.41   bending geom & max life fctrs  0.434  1.04  0.382  1.04
vel'y factor  1.16   contact, bending life factors  0.759 0.223  0.911 0.268
velocity, m/s 1.86   contact, bending lives, khr    115.3 large  16.00 large
module, mm   16.00   pitting geom factor 0.1190    contact ratio 1.346
width, mm    186.8   commercial, 8 accuracy level gears
```

PROBLEM 15

Trial pinion tooth numbers were input to the program, which output the corresponding module and facewidth necessary to achieve the design life. Only the solutions corresponding to the smallest tooth number for each module are shown below. As noted in the extension to the worked synthesis example, there’s not much to choose between the candidates, given that fatigue failure only is considered.

```
problem 15a   pinion volume = 0.25p*104.7*sq( 8*9) =  426 cc
power, kW    100.0   PINION, WHEEL - speeds, rpm        1450.0        466.1
appl'n factor 1.00   tooth number, profile shift        9  0.56     28  0.33
rel'y factor  1.00   all. contact, bending stresses  1450   400   1300   350 MPa
dist'n factor 1.35   bending geom & max life fctrs  0.426  1.04  0.391  1.04
vel'y factor  1.28   contact, bending life factors  0.716 0.198  0.798 0.246
```